Heating Ventilating
Air Conditioning
Guide 1949

HEATING VENTILATING AIR CONDITIONING GUIDE 1949

An Instrument of Service
Prepared for the Profession
containing

A TECHNICAL DATA SECTION OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS BASED ON—THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND COOPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY; A MANUFACTURERS' CATALOG DATA SECTION CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT; COMPLETE INDEXES TO TECHNICAL AND CATALOG DATA SECTIONS.

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PREFACE TO THE 27th EDITION

IN THE revision of The Heating Ventilating Air Conditioning Guide for the 1949 edition, all chapters have been examined and improvements have been made wherever current engineering practice or recent published data have indicated progress or change in the presentation of the subject. The Technical Data Section has been enlarged by 80 pages, and the Catalog Data Section has been increased by the addition of up-to-date products of many additional manufacturers.

Particular attention is called to the revision of the following chapters:

Chapter 1—Terminology. Many definitions have been added to the previous list.

Chapter 4—Fluid Flow. The theory and mathematical background for fluid flow computations have been enlarged upon and a section has been added on variable-area flow meters.

Chapter 6—Heat Transmission Coefficients of Building Materials. The section on vapor transmission has been enlarged and the method of computing the overall heat transfer coefficient of walls having high conductivity members has been expanded and improved. Federal specifications for water-proofed building papers have been added.

Chapter 10—Air Contaminants. Revisions have been made in tables of permissible limits of contaminants wherever changes have been found in accepted values.

Chapter 11—Instruments and Measurements. Descriptions and comments on use of several additional instruments have been added. The chapter has been extensively revised and portions rewritten to clarify principles of operation and to indicate precision of various measurements.

Chapter 12—Physiological Principles. Revisions in sections relating to human reactions to extremes of hot and cold environments have brought data on these effects up to date with most recent research, and also with findings of the Armed Services.

Chapter 13—Air Conditioning in the Prevention and Treatment of Disease. The text has been coordinated with present medical conclusions in regard to environmental conditions found most beneficial in treatment of disease.

Chapter 14—Heating Load. Data on tabulated winter climatic conditions have been extended by addition of new information received from the U. S. Weather Bureau.

Chapter 15—Cooling Load. The calculation of solar heat gain through walls and roofs has been simplified by means of tables providing equivalent temperature differentials which can be used with the heat transfer coefficients of the structure to obtain the heat transfer resulting from solar radiation, as well as air temperature difference. The tables also contain correction factors to permit application to other than tabulated conditions. Tabulated climatic conditions have been brought up to date by new data furnished by the U. S. Weather Bureau.

Chapter 17—Automatic Fuel Burning Equipment. The section on Vaporizing Oil Burners has been enlarged. Tables and data on sizing of gas piping to appliances have been added.

Chapter 18—Heating Boilers, Furnaces, Space Heaters. A section has been added to cover space heaters, their design, rating, installation and operation.

Chapter 22—Mechanical Warm Air Systems. The chapter has been completely revised in conformity with latest recommendations of the *NWAH&ACA*. A section, Design Procedure for Large Systems, has been added.

Chapter 23—Steam Heating Systems and Piping. The section referring to traps has been rewritten and illustrations of various types of traps added. The range of steam main and riser capacities has been extended for two-pipe systems. Many new diagrams showing piping connections to heating coils have been included.

Chapter 26—Unit Heaters, Unit Ventilators, Unit Humidifiers. Improved air vent piping connections are shown in unit heater piping diagram.

Chapter 30—Electric Heating. Sections pertaining to Resistors and Heating Elements, Unit Heaters, and Electric Heating by Induction and Dielectric means, have been rewritten.

Chapter 33—Air Cleaning Devices. Dust Collectors are discussed in a new section which covers types, application and selection of collectors for industrial use.

Chapter 35—Motors and Motor Controls. Revisions have been made wherever necessary to insure compliance with present motor and motor control practice and specifications. Descriptions of types of motors have been improved.

Chapter 37—Spray Apparatus. The entire sections dealing with cooling towers and spray ponds have been revised and rewritten to include description of current equipment and to present up-to-date design information. Factors affecting location, selection and performance of cooling towers are given. Illustrations of various types of towers are included.

Chapter 39—Refrigeration. A thorough revision improving the presentation of the subject includes a new section on the heat pump, and added pressure-enthalpy charts for ammonia and "Freon-12". Revised tables are supplied for sizing of refrigerant piping. Emphasis is placed upon application of refrigeration equipment.

Chapter 41—Air Duct Design. New tables giving circular equivalents of rectangular ducts are based upon formulas developed in latest ASHVE research. The treatment of losses occurring in systems has been enlarged and includes a detailed discussion of losses due to area changes. Under the subject of sizing ducts, three methods are outlined and examples are included.

Chapter 46—Industrial Exhaust Systems. Numerous revisions in fundamental design data have been added to keep this chapter in agreement with the latest knowledge on the subject.

Chapter 47—Drying Systems. New illustrations of dryers and new diagrams of drying operations are included in the rewritten and enlarged chapter. Problems in drying have been amplified. An example is given in the solution of a typical drying problem and the selection of a belt-type of dryer.

Chapter 48—Transportation Air Conditioning. This chapter has been rewritten in order to present current practice in design and control of heating, cooling and air conditioning systems in modern railway passenger cars, street cars, trolley coaches, passenger buses and aircraft. Special requirements for various types of planes are discussed.

Chapter 52—Codes and Standards. The record of codes and standards applicable to heating, ventilating and air conditioning has been reviewed, and references to the latest editions of the codes have been brought up to date. In addition, a reference column indicates where the codes may be obtained, and a list of addresses of the sponsoring organizations has been added.

The cross index of the Technical Data Section has been enlarged to facilitate reference to the technical text.

The high standard established by The Guide has been made possible by the contributions of many members, organizations and individual authorities in the past, as well as those who have devoted considerable time and effort to the improvement of the present edition. It is a pleasure to acknowledge this valuable service of the following contributors to the 27th edition:

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The reader's attention is directed to the Catalog Data Section which provides concise information on the up-to-date products of 249 manufacturers. A cross index of manufactured products facilitates the finding of sources of supply for equipment needed in heating, ventilating and air conditioning installations.

The 1949 edition of The Guide is presented with the hope that the many improvements incorporated will make this volume increasingly valuable to the manufacturer, student, teacher and practicing engineer.

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CODE OF ETHICS FOR ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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CHAPTER 1

TERMINOLOGY

Glossary of Physical and Heating, Ventilating, Refrigerating and Air Conditioning Terms Used in the Text

Absolute Zero: The zero from which absolute temperature is reckoned. Approximately -273.2 C or -459.8 F.

Absorbent: A sorbent which changes physically or chemically, or both, during the sorption process.

Absorption: The action of a material in extracting one or more substances present in an atmosphere or mixture of gases or liquids; accompanied by physical change, chemical change, or both, of the sorbent.

Acceleration: The time rate of change of velocity *i.e.*, the derivative of velocity with respect to time. In the cgs system the unit of acceleration is the centimeter per (second) (second), in the fps system the unit is the foot per (second) (second) $a = \frac{dv}{dt}$.

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body, the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per (sec) (sec) or 32.174 ft per (sec) (sec) which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process such that no heat is added to or taken from a substance or system undergoing the process.

Adsorbent: A sorbent which does not change physically or chemically during the sorption process.

Adsorption: The action, associated with surface adherence, of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids unaccompanied by physical or chemical change. Commercial adsorbent materials have enormous internal surfaces.

Aerosol: An assemblage of small particles, solid or liquid, suspended in air. The diameters of the particles may vary from 100 microns down to 0.01 micron or less, *e.g.* dust, fog, smoke.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, gases, vapors, fumes and smokes. (Air cleaners include air washers, air filters, electrostatic precipitators and charcoal filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (*See Comfort Air Conditioning.*)

Air, Dry: In psychrometry, air unmixed with, or containing no, water vapor.

Air, Saturated: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature.

Air, Standard: Air with a density of 0.075 lb per cu ft and an absolute viscosity of 1.22×10^{-5} lb mass per (ft) (sec). This is substantially equivalent to dry air at 70 F and 29.92 in. (Hg) barometer.

Air Washer: An enclosure in which air is drawn or forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of a fluid.

Aspect Ratio: In air distribution outlets the ratio of the length of the core of a grille, face or register to the width.

In rectangular ducts the ratio of the width to the depth.

Atmospheric Pressure: The pressure due to the weight of the atmosphere. It is the pressure indicated by a barometer. *Standard Atmospheric Pressure* or *Standard Atmosphere* is the pressure of 76 cm of mercury having a density of 13.5951 grams per cu cm, under standard gravity of 980.665 cm per (sec) (sec). It is equivalent to 14.696 lb psi or 29.921 in. of mercury at 32 F.

Baffle: A surface used for deflecting fluids, usually in the form of a plate or wall.

Blast Heater: A set of heat transfer coils or sections used to heat air which is drawn or forced through it by a fan.

Blow (throw): In air distribution, the distance an air stream travels from an outlet to a position at which air motion along the axis reduces to a velocity of 50 fpm.

For unit heaters, the distance an air stream travels from a heater without a perceptible rise due to temperature difference and loss of velocity.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.3 \times 34.5 = 33{,}475$ Btu per hr.

British Thermal Unit: Classically the Btu is defined as the quantity of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree. By this definition the exact value depends upon the initial temperature of the water. Several values of the Btu are in more or less common use, each differing from the others by a slight amount. One of the more common of these is the *mean Btu* which is defined as 1/180 of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F at a constant atmospheric pressure of 14.696 lb per sq in absolute.

For most accurate work the *International Table* (I.T.) Btu is usually used. This is defined by the relation: 1 (I.T.) Btu per (pound) (Fahrenheit degree) = 1 (I.T.) calorie per (gram) (Centigrade degree). This value corresponds to the amount of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree at 58 F and also at 149 F. The *mean Btu* corresponds to 1.0008 (I.T.) Btu.

By-Pass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around an element of a system.

Calorie (Gram Calorie): Classically the calorie is defined as the quantity of heat required to raise the temperature of 1 gram of water 1 Centigrade degree. By this definition the exact value depends upon the initial temperature of the water. Several values of the calorie are in more or less common use, each differing from the others by a slight amount. Among these are the *15 C calorie* and the *17½ C calorie*. The *mean calorie*, *i.e.* 1/100 the quantity of heat required to raise the temperature of 1 gram of water from 0 C to 100 C, is also extensively used.

For the most accurate work the International Table (I.T.) calorie, defined in terms of the international electrical units, is usually used: 1 (I.T.) calorie = 1/860 international watt-hour = 3,600/860 international watt-seconds or international joules. The kilocalorie = 1,000 cal.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distributing ducts. (See Chapter 43.)

Chimney Effect: The tendency of air or gas in a duct or other vertical passage to rise when heated due to its lower density compared with that of the surrounding air or gas. In buildings, the tendency toward displacement (caused by the difference in temperature) of internal heated air by unheated outside air due to the difference in density of outside and inside air.

Comfort Air-Conditioning: The process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. (See *Air Conditioning*.)

Comfort Line: The effective temperature at which the largest percentage of adults feels comfortable.

Comfort Zone (*Average*): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. (See Chapter 12.)

Condensate: The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condensation: The process of changing a vapor into liquid by the extraction of heat. Condensation of steam or water vapor is effected in either steam condensers or in dehumidifying coils and the resulting water is called *condensate*.

Conductance, Surface (*Unit*): The amount of heat transferred by radiation, conduction, and convection from unit area of a surface to the air or other fluid in contact

with it, or vice versa, in unit time for a unit difference in temperature between the surface and the fluid. The common unit is: Btu per (hour) (square foot) (Fahrenheit degree). Symbol *f*. The temperature of the fluid should be taken in a plane sufficiently far from the surface that it will not be affected by the temperature of the surface.

Conductance, Thermal: The time rate of heat flow through unit area of a body, of given size and shape, per unit temperature difference. Common unit is: Btu per (hour) (square foot) (Fahrenheit degree). Symbol *C*.

Conduction, Thermal: The process of heat transfer through a material medium in which kinetic energy is transmitted by the particles of the material from particle to particle without gross displacement of the particles.

Conductivity, Thermal: The time rate of heat flow through unit area of a homogeneous substance under the influence of a unit temperature gradient. Common units are: Btu per (hour) (square foot) (Fahrenheit degree per inch). Symbol *k*

Conductor, Thermal: A material which readily transmits heat by means of conduction.

Convection: The motion resulting in a fluid from the differences in density and the action of gravity. In heat transmission this meaning has been extended to include both *forced* and *natural* motion or circulation.

Convective Heat Transfer: The transmission of heat by either natural or forced motion of a fluid (liquid or gas).

Convector: An agency of convection. In heat transfer, a surface designed to transfer its heat to a surrounding fluid largely or wholly by convection. The heated fluid may be removed mechanically or by gravity (Gravity Convector). Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator*.) (See also Chapter 25.)

Decibel: A unit used to express the relation between two amounts of power. By definition the difference in decibels between two powers P_1 and P_2, P_2 being the larger, is: db difference $= 10 \log_{10} P_2/P_1$.

In acoustics the threshold of hearing at 1,000 cycles per sec has been standardized at 10^{-16} watts per sq cm. If P_2 is the power in watts per square centimeter of a measured sound, then $10 \log_{10} P_2/10^{-16}$ is the db difference above the threshold and is known as the ***intensity level***. This is a definite recognized way of describing the intensity of a sound.

Declination of Sun: The angle above or below equatorial plane. It is plus if north of the plane, and minus if below. Celestral objects are located by declination.

Degree-Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65 F, there exists as many degree-days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65 F.

Dehumidify: To reduce by any process, the quantity of water vapor within a given space.

Dehydrate: To remove water in all forms from matter. Liquid water, hygroscopic water, and water of crystallization or water of hydration are included.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight per unit volume.

Dew-Point: See *Temperature, Dew-Point*.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct-Return System (*Hot Water*)**:** A hot water system in which the water, after it has passed through a heating unit is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)**:** A pipe which carries steam downward to the heating units and into which the condensate from the heating units drains.

Down-Feed System (*Steam*)**:** A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft: A current of air. Usually refers to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater or space.

Draft Head (*Side Outlet Enclosure*): The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*): The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry: To separate or remove a liquid or vapor from another substance. The liquid may be water but the term is also used for the removal of liquid or vapor forms of other substances.

Dust: An air suspension (aerosol) of solid particles of any material. (See also Chapter 10.)

Element Electric Heating. A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Enthalpy: A term used in lieu of *total heat* or *heat content*. Expressible in Btu per pound. Mathematically defined as $h = u + pv/J$. When a change occurs at constant pressure, as when water is boiled, the change in enthalpy is equal to the heat added, in this case latent heat.

Enthalpy, Free: A thermodynamic property which serves as a measure of the available energy of a system with respect to surroundings at the same temperature and same pressure as that of the system. No process involving an increase in available energy can occur spontaneously.

Enthalpy, Specific: A term sometimes applied to enthalpy per unit weight, the English unit being Btu per pound.

Entropy: The ratio of the heat added to a substance to the absolute temperature at which it is added. Mathematically, for a reversible process, $dS = dQ/T$ or $S = \int dQ/T$.

These formulas are applicable when temperature is not constant. During a reversible adiabatic change entropy is constant. During a reversible isothermal change the heat absorbed by the substance is equal to the product of the absolute temperature of the substance and its change of entropy.

Entropy, Specific: A term sometimes applied to entropy per unit weight, the English unit being Btu per (Fahrenheit degree, absolute) (pound).

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and coresponding atmospheric pressure.

Fan Furnace System: See *Warm Air Heating System*.

Fog: Suspended liquid droplets generated by condensation from the gaseous to the liquid state or by breaking up a liquid into a dispersed state, such as by splashing, foaming, and atomizing. (See also Chapter 10.)

Force: The action on a body which tends to change its relative condition as to rest or motion.

Fumes: Smoke; aromatic smoke; odor emitted, as of flowers; a smoky or vaporous exhalation, usually odorous, as that from concentrated nitric acid. The word *fumes* is so broad and inclusive that its usefulness as a technical term is very limited. Its principal definitive characteristic is that it implies an odor. The terms vapor, smoke, fog, etc., which can be more strictly defined, should be used whenever possible.

Also defined as solid particles generated by condensation from the gaseous state, generally, after volatilization from molten metals, etc., and often accompanied by a chemical reaction such as oxidation. Fumes flocculate and sometimes coalesce. (See also Chapter 10.)

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also a complete heating unit for transferring heat from fuel being burned to the air supplied to a heating system.

Furnace Volume (*Total*): The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as previously defined.

Gravity, Specific: The ratio of the mass of a unit volume of a substance to the mass of the same volume of a standard substance at a standard temperature. Water at 39.2 F is the standard substance usually referred to. For gases, dry air, at the same temperature and pressure as the gas, is often taken as the standard substance.

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Head, Dynamic: Same as Total Pressure expressed in height of liquid.

Heat: The form of energy that is transferred by virtue of a temperature difference. At constant pressure heat added is equal to enthalpy change.

Heat, Humid: Ratio of increase of enthalpy per pound of dry air to rise of temperature under conditions of constant pressure and constant humidity ratio.

Heat, Latent: A term used to express the energy involved in a change of state.

Heat, Sensible: A term used in heating and cooling to indicate any portion of heat which changes only the temperature of the substances involved.

Heat of the Liquid: The increase in enthalpy per unit weight of a saturated liquid as its temperature increases from a chosen base temperature. For water the base temperature is usually taken as 32 F.

Heat, Specific: The heat absorbed (or given up) by a unit mass of a substance when its temperature is increased (or decreased) by 1 deg. Common Units: Btu per (pound) (Fahrenheit degree), calories per (gram) (Centigrade degree). For gases, both specific heat at constant pressure (C_p) and specific heat at constant volume (C_v) are frequently used. In air-conditioning, C_p is usually used.

Heat, Total: See Enthalpy.

Heat Transmission, Coefficient: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures. (See thermal conductance, thermal conductivity, thermal resistance, thermal resistivity, thermal transmittance, etc.).

Heater, Electric: A complete assembly of heating elments with their enclosure ready for installation in service.

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried from the boiler to the heating units.

Humidify: To increase, by any process, the density of water vapor within a given space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor within a given space.

Humidity, Absolute: The weight of water vapor per unit volume, pounds per cubic foot or grams per cubic centimeter.

Humidity, Relative: The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature. (See discussion in Chapter 3.)

Humidity Ratio: In a mixture of water vapor and air, the weight of water vapor per pound of dry air. Also called *Specific Humidity*.

Humidity Specific: See Humidity Ratio.

Hygrostat: Same as Humidistat.

Inch of Water: A unit of pressure equal to the pressure exerted by a column of liquid water 1 in. high at a standard temperature. The standard temperature is sometimes taken as 0 C and sometimes as 62 F. One inch of water at 62 F = 5.197 lb per sq ft.

Insulation (*Thermal*): A material having a relatively high resistance to heat flow, and used principally to retard the flow of heat.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Load, Estimated Design: In a heating or cooling system, the sum of the useful heat transfer plus heat transfer from or to the connected piping plus heat transfer occurring in any auxiliary apparatus connected to the system. The units are Btu per hour or, in heating, equivalent direct radiation (EDR) which is becoming obsolete.

Load, Estimated Maximum: In a heating or cooling system, the calculated maximum heat transfer that the system will be called upon to provide.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: A measure of the inertia of a body. It also measures the quantity of matter in a body. Since the only general property of a given portion of matter that cannot be changed is its inertia, it is this property by which quantities of matter are defined. Two bodies which have equal inertias are said to have equal masses or to contain equal quantities of matter. (This definition fails at velocities approaching the velocity of light.) The mass of a body is numerically equal to the ratio of the force required to give the body a given acceleration to the acceleration. $m = F/a$. The common units of mass are the gram and the pound.

Mechanical Equivalent of Heat: The quantity of mechanical energy equal to one unit of heat. $J = 778.3$ ft-lb per Btu $= 4.187 \times 10^7$ ergs per gram-calorie.

Medium, Heating: A substance such as water, steam, air or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Micron: A unit of length, the thousandth part of 1 mm or the millionth of a meter.

Millimeter of Mercury: A unit of pressure equal to the pressure exerted by a column of mercury 1 mm high at a temperature of 0 C. One millimeter of mercury at 0 C $= 1.934 \times 10^{-2}$ psi.

Mol: A weight of a substance numerically equal to its molecular weight. If the weight is in pounds the unit is a *Pound Mol*, in grams the unit is a *Gram Mol*. For perfect gases the volume of 1 mol is constant for all gases at the same temperature and pressure. For real gases this is approximately true at moderate pressures. At 32 F and zero-pressure the value of the product, pressure times specific volume, is 359.045 $\pm$ 0.006 atmosphere cubic feet (atm ft^3), *for 1 mol of any gas*. For dry air at 32 F and standard atmospheric pressure, the specific volume is 358.83 cu ft per mol (ft^3 per mol).

One-Pipe Supply Riser—(*Steam*): A pipe which carries steam vertically to a heating unit and which also carries the condensate from the heating unit. In an upfeed system steam and condensate flow in opposite directions; in an overhead or down-feed system they flow in the same direction.

One-Pipe System—(*Steam*): A steam heating system in which a single main serves the dual purpose of supplying steam to the heating unit and conveying condensate from it. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used. (Hot Water)—A hot water system in which the cooled water from the heating units is returned to the supply main. Consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

Overhead System: Any steam or hot water system in which the supply main is above the heating unit. In a steam system the return must be below the heating units; in a water system the return may be above or below the heating units.

Panel Heating: A heating system in which heat is transmitted by both radiation and convection from panel surfaces to both air and surrounding surfaces.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for comparing small electromotive forces or for measuring small electromotive forces by comparison with a known electromotive force. Its principal advantage is that during the measurement no current flows through the source of electromotive force. .

Power: The rate of performing work. Common units are horsepower, Btu per hour, and watts.

Pressure: Force per unit area. Common units are pounds per square inch, gram per square centimeter, inch of water, millimeter of mercury.

Pressure, Absolute: The sum of the gage pressure and the barometric pressure.

Pressure, Gage: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Pressure, Dynamic: Same as Total Pressure.

Pressure, Saturation: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid or vapor and solid can co-exist in stable equilibrium.

Pressure, Static: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances created by inserting the tube cancel. It is supposed that the thermo-dynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Pressure, Total: In the theory of the flow of fluids; the sum of the static pressure and the velocity pressure at the point of measurement.

Pressure, Vapor: The pressure exerted by a vapor. If a vapor is kept in confinement over its liquid so that the vapor can accumulate above the liquid, the temperature being held constant, the vapor pressure approaches a fixed limit called the maximum, or saturated, vapor pressure, dependent only on the temperature and the liquid. The term *vapor pressure* is sometimes used as synonymous with *saturated vapor pressure.*

Pressure, Velocity: In a moving fluid, the pressure capable of causing an equivalent velocity if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

Psychrometric: Pertaining to psychrometry or the state of the atmosphere with reference to moisture.

Psychrometry: The branch of physics relating to the measurement or determination of atmospheric conditions, particularly regarding the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiant Heating: A heating system in which only the heat radiated from panels is effective in providing the heating requirements. The term *Radiant Heating* is frequently used to include both *Panel and Radiant Heating.*

Radiation: The transmission of energy by means of electromagnetic waves.

Radiation, Thermal (Heat) Radiation: The transmission of energy by means of electromagnetic waves of very long wave length. Radiant energy of any wave length may, when absorbed, become thermal energy and result in an increase in the temperature of the absorbing body.

Radiation, Equivalent Direct (EDR): A unit of heat delivery of 240 Btu (*Steam*) or 150 Btu (*Water*) per hour. It does not imply 144 sq in. of surface.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects within visible range and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called radiator is also a convector but the term radiator has been established by long usage.

Radiator, Concealed: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, is not visible from the room. Such a device transfers its heat to the room largely by convection air currents.

Radiator, Direct: Same as Radiator.

Radiator, Recessed: A heating unit set back into a wall recess but not enclosed.

Radiator, Tube or Tubular: A heating unit used as a radiator in which the heat transfer surfaces are principally tubes.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Refrigeration, Ton of: The removal of heat at a rate of 200 Btu per min, 12,000 Btu per hr, or 288,000 Btu per 24 hr.

Resistance, Thermal: The reciprocal of thermal conductance. Symbol R.

Resistor, Electric: A material used to produce heat by passing an electric current through it.

Resistivity, Thermal: The reciprocal of thermal conductivity. Symbol *r*.

Return, Dry: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Return, Wet.*)

Return, Wet: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Return, Dry.*)

Return Mains: Pipes or conduits which return the heating or cooling medium from the heat transfer unit to the source of heat or refrigeration.

Reversed-Return System: A system in which the heating or cooling medium from several heat transfer units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of it are of practically equal length.

Sabin: A unit of equivalent sound absorption equal to the equivalent absorption of one square foot of a surface of unit absorptivity (*i.e.*, of one square foot of surface which absorbs all incident sound energy).

Saturation: The condition for co-existence in stable equilibrium of a vapor and liquid or a vapor and solid phase of the same substance. Example: Steam over the water from which it is being generated.

Saturation, Degree of: The ratio of the weight of water vapor associated with a pound of dry air to the weight of water vapor associated with a pound of dry air saturated at the same temperature.

Smoke: An air suspension (aerosol) of particles, usually but not necessarily solid, often originating in a solid nucleus, formed from combustion or sublimation. Also defined as carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Solar Constant: The solar intensity incident on a normal surface located outside the earth's atmosphere at a distance from the sun equal to the mean distance between the earth and the sun. Its value is 415, 445, or 430 Btu per (hr) (sq foot) as the July, January, or mean value respectively. At sea level in July the solar intensity value is about 300 Btu per (sq ft) (hr) since about 28 per cent is absorbed in the earth's atmosphere.

Sorbent: A material which extracts one or more substances present in an atmosphere or mixture of gases or liquids with which it is in contact, due to an affinity for such substances.

Sorption: Adsorption or absorption.

Split System: A system in which the heating is accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point. Ventilation may be provided by the same system.

Square Foot of Heating Surface (*Equivalent*)**:** This term is synonymous with Equivalent Direct Radiation (EDR).

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of heat to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate, or of air and condensate and preventing the passage of steam.

Supply Mains: The pipes through which the heating medium flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface, Heating: The exterior surface of a heating unit. *Extended heating surface* (*or extended surface*): Heating surface consisting of fins, pins or ribs which receive heat by conduction from the prime surface. *Prime Surface*: Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also Boiler Heating Surface.)

Temperature: The thermal state of matter with reference to its tendency to communicate heat to matter in contact with it. If no heat flows upon contact, there is no difference in temperature.

Temperature, Absolute: Temperature expressed in degrees above absolute zero.

Temperature, Dry-Bulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Temperature, Dew-Point: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 per cent relative humidity) for a given absolute humidity at constant pressure.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Wet-Bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*A.S.M.E.* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways. *The First Law*: (1) When work is expended in generating heat, the quantity of heat produced is proportional to the work expended; and conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done. (Joule)[a] (G.P.)[b]; (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states. (Zemansky); (3) In any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. *The Second Law* (1) It is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower to one of higher temperature. (Clausius) (G.P.); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin) (G.P.); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls temperature.

Transmittance, Thermal: The time rate of heat flow, from the fluid on the warm side to the fluid on the cold side, per (square foot) (degree temperature difference between the two fluids). Sometimes called *Over-all Coefficient of Heat Transfer*.

Common unit is Btu per (hour) (square foot) (Fahrenheit degree). Symbol U.

Two-Pipe System (*Steam or Water*): A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Up-Feed System: A heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vane Ratio: In air distributing devices the ratio of depth of vane to shortest opening width between two adjacent grille bars.

Vapor: The gaseous form of substances which are normally in the solid or liquid state and which can be changed to these states either by increasing the pressure or decreasing the temperature. Vapors diffuse. (*A.S.A.* definition.)

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensate to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the

[a] Names of authors who first stated law are given in parentheses.
[b] From Glossary of Physics, by LeRoy Dougherty Weld, (McGraw-Hill, 1937)

return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return.

Velocity: A vector quantity which denotes at once the time rate and the direction of a linear motion. $V = \frac{ds}{dt}$. For uniform linear motion $V = \frac{s}{t}$. Common units are: feet per second.

Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See Air Conditioning.)

Volume, Specific: The volume of a substance per unit mass; the reciprocal of density. Units: cubic feet per pound, cubic centimeters per gram, etc.

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts.

Warm Air Heating System, Gravity: A warm air heating system in which the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing.

Warm Air Heating System, Mechanical: A warm air heating system in which circulation of air is effected by a fan. Such a system may include air cleaning devices.

CHAPTER 2

ABBREVIATIONS AND SYMBOLS

Standard Abbreviations; Standard Symbols; Greek Alphabet; Conversion Equations; Graphical Symbols for Piping, Ductwork, Heating and Ventilating, Refrigerating; Identification of Piping by Color; Specific Heat Table

THIS chapter contains information regarding abbreviations, symbols, and conversion equations, which are of particular interest to the engineer engaged in heating, ventilating, and air conditioning.

ABBREVIATIONS

Abbreviations are shortened forms of names and expressions employed in texts and tabulations and should not generally be used as symbols in equations. Most of the following abbreviations have been compiled from a list of approved standards[1]. In general the period has been omitted in all abbreviations except where the omission results in the formation of an English word. Additional abbreviations applying to individual chapters will be found at the end of Chapters 3, 4, 5, 7, 15, 16, 39, and 41.

Absolute	abs
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
British thermal units per hour	Btuh
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Cubic	cu
Cubic centimeter	cu cm or cc
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[2]	deg or °
Degree, Centigrade	C
Degree, Fahrenheit	F
Degree, Kelvin	K
Degree, Réaumur	R
Diameter	diam

[1] Abbreviations for Scientific and Engineering Terms, Z10 1-1941 (*American Standards Association*).
[2] It is recommended that the abbreviation for the temperature scale, F, C, K, R, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Thermal conductance per unit area, Unit conductance: heat transferred per (unit time) (unit area) (degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity: heat transferred per (unit time) (unit area) (degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer: heat transferred per (unit time) (unit area) (degree) .... f

$$= \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area: heat transferred per (unit time) (unit area) (degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degree per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* $_{fg}$
Velocity V
Viscosity, absolute .... μ
Viscosity, kinematic .... μ/ρ

Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight. q
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

THE GREEK ALPHABET

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ κ	Kappa	Σ σ ς	Sigma
Γ γ	Gamma	Λ λ	Lamdba	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	Υ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ ϕ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ ϑ θ	Theta	Π π	Pi	Ω ω	Omega

CONVERSION EQUATIONS[5]

Heat, Power and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.4 Btu per pound
1 Btu	= 778.3 ft-lb 0.2930 Int. whr 252.0 I.T. calorie
1 Int. watthour	= 2656 ft-lb 3.413 Btu 3600 Int. joules 860 I.T. calories
1 Int. kilowatthour	= 3,413 Btu 3.517 lb water evaporated from and at 212 F
1 Int. kilowatt (1000 watts)	= 1.341 hp 56.88 Btu per minute 44,267 ft-lb per minute
1000 I.T. calories 1 I.T. Kilocalorie	= 3.968 Btu 3088 ft-lb 1.1628 Int. whr
1 horsepower	= 0.7455 Int. kw 42.40 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,475 Btu per hour 9.809 Int. kw

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.1337 cu ft
1 British or Imperial gallon	= 277.42 cu in.
1 cu ft	= 7.481 gal 1728 cu in.
1 cu ft water at 60 F (in vacuo)	= 62.37 lb
1 cu ft water at 212 F (" ")	= 59.83 lb
1 gal water at 60 F (" ")	= 8.338 lb
1 gal water at 212 F (" ")	= 7.998 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb

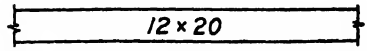
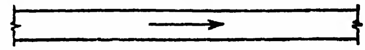
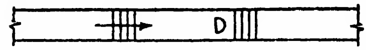
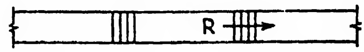
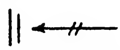
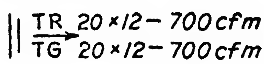
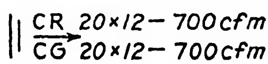
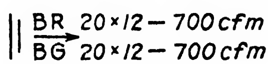
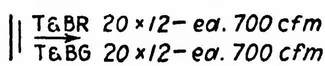
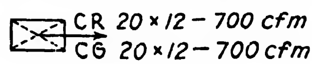
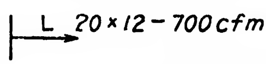
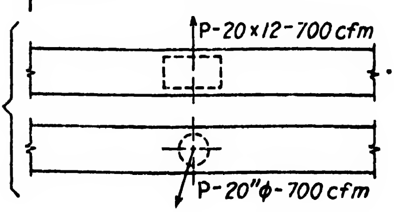
Pressure

1 lb per square inch	= 144 lb per square foot 2.0360 in. mercury at 32 F 2.0422 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F 1.732 in. water at 62 F

[5] Checked in 1944 by *National Bureau of Standards*. Abbreviations *Int.* and *I.T.* refer to *International* and *International (Steam) Table* respectively.

Graphical Symbols for Drawings

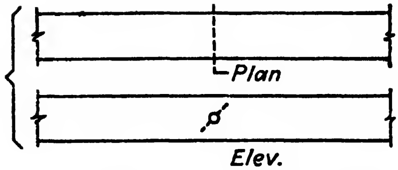
Ductwork

45. Duct (1st Figure, Width; 2nd, Depth)	
46. Direction of Flow	
47. Inclined Drop in Respect to Air Flow	
48. Inclined Rise in Respect to Air Flow	
49. Supply Duct Section	
50. Exhaust Duct Section	
51. Recirculation Duct Section	
52. Fresh Air Duct Section	
53. Other Duct Sections	
54. Register	R
55. Grille	G
56. Supply Outlet	
57. Exhaust Inlet	
58. Top Register or Grille	
59. Center Register or Grille	
60. Bottom Register or Grille	
61. Top and Bottom Register or Grille	
62. Ceiling Register or Grille	
63. Louver Opening	
64. Adjustable Plaque	

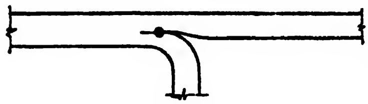
Graphical Symbols for Drawings

Ductwork

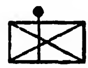
65. Volume Damper



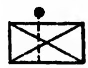
66. Deflecting Damper



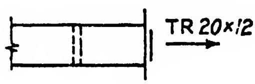
67. Deflecting Damper, Up



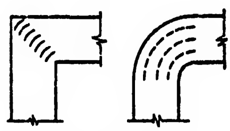
68. Deflecting Damper, Down



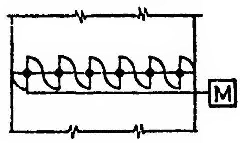
69. Adjustable Blank Off



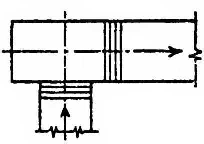
70. Turning Vanes



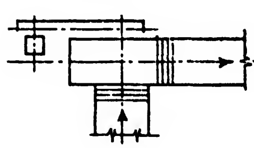
71. Automatic Dampers



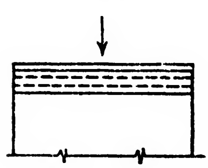
72. Canvas Connections



73. Fan and Motor With Guard



74. Intake Louvers and Screen

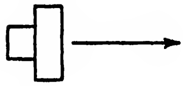


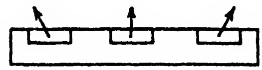
Graphical Symbols for Drawings — *Heating and Ventilating*

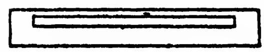
75. Heat Transfer Surface, Plan


76. Wall Radiator, Plan


77. Wall Radiator on Ceiling, Plan


78. Unit Heater (Propeller), Plan


79. Unit Heater (Centrifugal Fan), Plan


80. Unit Ventilator, Plan


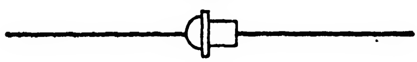
Traps

81. Thermostatic


82. Blast Thermostatic

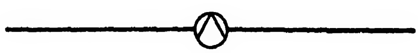

83. Float and Thermostatic


84. Float


85. Boiler Return


Valves

86. Reducing Pressure


87. Air Line


88. Lock and Shield


89. Diaphragm


90. Air Eliminator


91. Strainer


92. Thermometer

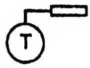

93. Thermostat


Graphical Symbols for Drawings

Refrigerating

94. Thermostat (Self Contained)

95. Thermostat (Remote Bulb)



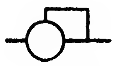
96. Pressurestat

97. Hand Expansion Valve

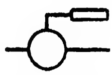
98. Automatic Expansion Valve

99. Thermostatic Expansion Valve

100. Evaporator Press. Regulating Valve, Throttling Type

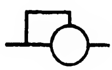


101. Evaporator Press. Regulating Valve, Thermostatic Throttling Type



102. Evaporator Press. Regulating Valve, Snap-Action Valve

103. Compressor Suction Pressure Limiting Valve, Throttling Type



104. Hand Shut Off Valve

105. Thermal Bulb

106. Scale Trap

107. Dryer

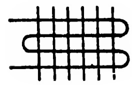
108. Strainer

109. High Side Float

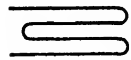
110. Low Side Float

111. Gage

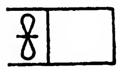
112. Finned Type Cooling Unit, Natural Convection



113. Pipe Coil



114. Forced Convection Cooling Unit



115. Immersion Cooling Unit

116. Ice Making Unit

117. Heat Interchanger

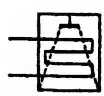
118. Condensing Unit, Air Cooled

119. Condensing Unit, Water Cooled

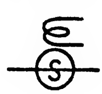
120. Compressor

121. Cooling Tower

122. Evaporative Condenser



123. Solenoid Valve



124. Pressurestat With High Pressure Cut-Out

IDENTIFICATION OF PIPING SYSTEMS BY COLOR

The color scheme for identification of piping systems listed in the following table and shown in Fig. 1 is reprinted from Part V, Fourth Edition, of the Engineering Standards of the *Heating, Piping and Air Conditioning Contractors National Association*[7].

All piping systems are classified according to the material carried in the pipes and colors are assigned as follows:

Class	Color
F—Fire-protection	Red
D—Dangerous materials	Yellow or Orange
S—Safe Materials	Green (or the achromatic colors, white, black, gray or aluminum)
and, when required	
P—Protective materials	Bright blue
V—Extra valuable materials	Deep purple

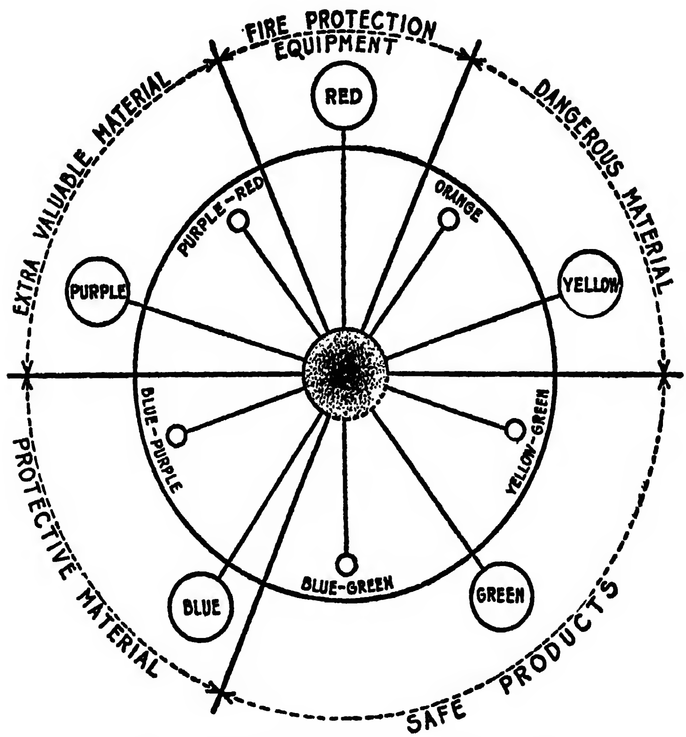


FIG. 1. MAIN CLASSIFICATION BY COLOR[a]

[a] From Scheme for Identification of Piping Systems, *Heating, Piping and Air Conditioning Contractors National Association*, Part V, Fourth Edition, p. 17. Used by permission.

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[7] See Scheme for Identification of Piping Systems, A13-1928, *American Standards Association*.

CHAPTER 3

THERMODYNAMICS

Thermodynamic Properties of Moist Air and of Water; Degree of Saturation; Goff Diagram for Moist Air; Derived Properties; Typical Air Conditioning Processes, Heating, Cooling, Adiabatic Mixing; Wet-Bulb Temperatures Below 32F; Dalton's Rule; Steady Flow Energy Equation, U. S. Standard Atmosphere

THE working substance of the air conditioning engineer is called *moist air*. In order to be able to apply the laws of conservation of energy and mass to the analysis of typical air conditioning processes, it is necessary to know the thermodynamic properties of moist air, particularly its enthalpy and volume. When the limitations imposed by the Second Law of Thermodynamics have to be considered, it is also necessary to know its entropy.

For the purpose of analysis, moist air may be regarded as a mixture of only two constituents, namely, *dry air* and *water vapor*. It has long been customary to predict the thermodynamic properties of the mixture from a knowledge of those of dry air and water vapor separately by means of Dalton's Rule. According to this rule: each constituent of a gas mixture occupies the whole volume of the mixture just as if no other constituent were present; it therefore exerts a *partial pressure* equal to the pressure it would exert if alone in the whole volume at the temperature of the mixture; the observed pressure of the mixture is the sum of these so-called partial pressures; the enthalpy of the mixture is the sum of separate contributions from the individual constituents as determined by their partial pressures, their weights, and the temperature of the mixture; and the entropy of the mixture is obtained in a similar manner.

Dalton's Rule has long been regarded erroneously as a fundamental law of nature. Actually it is not, and in many cases its predictions are quite unreliable. In the case of moist air at atmospheric pressure, it happens to give a close approximation to the truth; but as progress is made the need for greater accuracy than the rule can afford is felt even in this case. Fortunately, most of the complications involved in following a correct procedure, based on the predictions of statistical mechanics, are met in preparing suitable tables of thermodynamic properties; and, once these tables have been prepared, their use in the analysis of typical air conditioning processes is actually simplified by abandonment of the rule together with its fictitious concepts of partial pressure, relative humidity, etc.

Thermodynamic Properties of Moist Air

Table 1, Thermodynamic Properties of Moist Air (Standard Atmospheric Pressure, 29.921 In. Hg), contains results of a cooperative investigation between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Towne Scientific School, University of Pennsylvania. These results are being studied by an International Joint Committee on Psychrometric Data as a possible starting point from which to reach agreement on standard properties of moist air. A detailed explanation of the data and methods used in constructing Table 1 is given in a paper[1] presented before the ASHVE.

Table 1. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^6$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^5$	
−160	0.2120	7.520	0.000	7.520	−38.504	0.000	−38.504	−0.10300	0.00000	−0.10300	−222.00	−0.4907	0.1009	−160
−159	0.2394	7.545	0.000	7.545	−38.262	0.000	−38.262	−0.10219	0.00000	−0.10219	−221.68	−0.4896	0.1139	−159
−158	0.2703	7.571	0.000	7.571	−38.021	0.000	−38.021	−0.10139	0.00000	−0.10139	−221.36	−0.4885	0.1286	−158
−157	0.3049	7.596	0.000	7.596	−37.779	0.000	−37.779	−0.10059	0.00000	−0.10059	−221.04	−0.4874	0.1450	−157
−156	0.3435	7.622	0.000	7.622	−37.538	0.000	−37.538	−0.09980	0.00000	−0.09980	−220.72	−0.4864	0.1635	−156
−155	0.3869	7.647	0.000	7.647	−37.296	0.000	−37.296	−0.09901	0.00000	−0.09901	−220.40	−0.4853	0.1842	−155
−154	0.4354	7.673	0.000	7.673	−37.055	0.000	−37.055	−0.09822	0.00000	−0.09822	−220.07	−0.4843	0.2073	−154
−153	0.4897	7.699	0.000	7.699	−36.813	0.000	−36.813	−0.09743	0.00000	−0.09743	−219.75	−0.4832	0.2331	−153
−152	0.5502	7.724	0.000	7.724	−36.572	0.000	−36.572	−0.09664	0.00000	−0.09664	−219.42	−0.4822	0.2620	−152
−151	0.6178	7.750	0.000	7.750	−36.330	0.000	−36.330	−0.09586	0.00000	−0.09586	−219.10	−0.4811	0.2912	−151
−150	0.6932	7.775	0.000	7.775	−36.088	0.000	−36.088	−0.09508	0.00000	−0.09508	−218.77	−0.4800	0.3301	−150
−149	0.7772	7.801	0.000	7.801	−35.847	0.000	−35.847	−0.09430	0.00000	−0.09430	−218.44	−0.4789	0.3701	−149
−148	0.8709	7.826	0.000	7.826	−35.606	0.000	−35.606	−0.09352	0.00000	−0.09352	−218.11	−0.4779	0.4146	−148
−147	0.9750	7.851	0.000	7.851	−35.364	0.000	−35.364	−0.09274	0.00000	−0.09274	−217.78	−0.4768	0.4641	−147
−146	1.091	7.876	0.000	7.876	−35.123	0.000	−35.123	−0.09198	0.00000	−0.09198	−217.45	−0.4758	0.5194	−146
−145	1.219	7.902	0.000	7.902	−34.881	0.000	−34.881	−0.09121	0.00000	−0.09121	−217.12	−0.4747	0.5807	−145
−144	1.362	7.927	0.000	7.927	−34.640	0.000	−34.640	−0.09044	0.00000	−0.09044	−216.78	−0.4737	0.6488	−144
−143	1.521	7.953	0.000	7.953	−34.399	0.000	−34.399	−0.08967	0.00000	−0.08967	−216.45	−0.4726	0.7243	−143
−142	1.698	7.978	0.000	7.978	−34.157	0.000	−34.157	−0.08892	0.00000	−0.08892	−216.11	−0.4716	0.8082	−142
−141	1.893	8.004	0.000	8.004	−33.916	0.000	−33.916	−0.08816	0.00000	−0.08816	−215.78	−0.4705	0.9011	−141
−140	2.109	8.029	0.000	8.029	−33.674	0.000	−33.674	−0.08740	0.00000	−0.08740	−215.44	−0.4695	1.004	−140
−139	2.348	8.054	0.000	8.054	−33.433	0.000	−33.433	−0.08664	0.00000	−0.08664	−215.11	−0.4684	1.118	−139
−138	2.613	8.079	0.000	8.079	−33.192	0.000	−33.192	−0.08589	0.00000	−0.08589	−214.77	−0.4674	1.244	−138
−137	2.906	8.105	0.000	8.105	−32.951	0.000	−32.951	−0.08514	0.00000	−0.08514	−214.43	−0.4663	1.383	−137
−136	3.229	8.130	0.000	8.130	−32.709	0.000	−32.709	−0.08440	0.00000	−0.08440	−214.09	−0.4653	1.537	−136
−135	3.586	8.156	0.000	8.156	−32.468	0.000	−32.468	−0.08365	0.00000	−0.08365	−213.75	−0.4642	1.707	−135
−134	3.980	8.181	0.000	8.181	−32.226	0.000	−32.226	−0.08291	0.00000	−0.08291	−213.41	−0.4632	1.895	−134
−133	4.414	8.207	0.000	8.207	−31.985	0.000	−31.985	−0.08217	0.00000	−0.08217	−213.07	−0.4621	2.102	−133
−132	4.893	8.232	0.000	8.232	−31.744	0.000	−31.744	−0.08144	0.00000	−0.08144	−212.72	−0.4611	2.330	−132
−131	5.419	8.258	0.000	8.258	−31.503	0.000	−31.503	−0.08070	0.00000	−0.08070	−212.38	−0.4600	2.581	−131
−130	6.000	8.283	0.000	8.283	−31.262	0.000	−31.262	−0.07997	0.00000	−0.07997	−212.03	−0.4590	2.858	−130
−129	6.637	8.309	0.000	8.309	−31.021	0.000	−31.021	−0.07924	0.00000	−0.07924	−211.68	−0.4579	3.182	−129

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp t(F)	Humidity Ratio $W_s \times 10^7$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^6$	
−128	0.7339	8.334	0.000	8.334	−30.780	0.000	−30.780	−0.07851	0.00000	−0.07851	−211.33	−0.4569	0.3492	−128
−127	0.8111	8.360	0.000	8.360	−30.539	0.000	−30.539	−0.07778	0.00000	−0.07778	−210.98	−0.4559	0.3863	−127
−126	0.8958	8.385	0.000	8.385	−30.298	0.000	−30.298	−0.07707	0.00000	−0.07707	−210.63	−0.4548	0.4267	−126
−125	0.9887	8.411	0.000	8.411	−30.057	0.000	−30.057	−0.07634	0.00000	−0.07634	−210.28	−0.4538	0.4710	−125
−124	1.091	8.436	0.000	8.436	−29.816	0.000	−29.816	−0.07562	0.00000	−0.07562	−209.93	−0.4527	0.5197	−124
−123	1.202	8.461	0.000	8.461	−29.575	0.000	−29.575	−0.07490	0.00000	−0.07490	−209.58	−0.4517	0.5730	−123
−122	1.325	8.486	0.000	8.486	−29.334	0.000	−29.334	−0.07419	0.00000	−0.07419	−209.23	−0.4506	0.6314	−122
−121	1.459	8.512	0.000	8.512	−29.093	0.000	−29.093	−0.07348	0.00000	−0.07348	−208.88	−0.4496	0.6953	−121
−120	1.606	8.537	0.000	8.537	−28.852	0.000	−28.852	−0.07277	0.00000	−0.07277	−208.52	−0.4485	0.7653	−120
−119	1.767	8.563	0.000	8.563	−28.611	0.000	−28.611	−0.07206	0.00000	−0.07206	−208.17	−0.4475	0.8419	−119
−118	1.942	8.588	0.000	8.588	−28.370	0.000	−28.370	−0.07135	0.00000	−0.07135	−207.81	−0.4464	0.9256	−118
−117	2.134	8.613	0.000	8.613	−28.129	0.000	−28.129	−0.07064	0.00000	−0.07064	−207.45	−0.4454	1.017	−117
−116	2.343	8.639	0.000	8.639	−27.889	0.000	−27.889	−0.06994	0.00000	−0.06994	−207.09	−0.4444	1.117	−116
−115	2.571	8.664	0.000	8.664	−27.648	0.000	−27.648	−0.06924	0.00000	−0.06924	−206.73	−0.4433	1.226	−115
−114	2.820	8.690	0.000	8.690	−27.407	0.000	−27.407	−0.06854	0.00000	−0.06854	−206.37	−0.4423	1.345	−114
−113	3.092	8.715	0.000	8.715	−27.166	0.000	−27.166	−0.06784	0.00000	−0.06784	−206.01	−0.4412	1.475	−113
−112	3.388	8.741	0.000	8.741	−26.926	0.000	−26.926	−0.06715	0.00000	−0.06715	−205.65	−0.4402	1.617	−112
−111	3.711	8.766	0.000	8.766	−26.685	0.000	−26.685	−0.06646	0.00000	−0.06646	−205.29	−0.4392	1.771	−111
−110	4.063	8.792	0.000	8.792	−26.444	0.000	−26.444	−0.06577	0.00000	−0.06577	−204.92	−0.4381	1.939	−110
−109	4.445	8.817	0.000	8.817	−26.204	0.000	−26.204	−0.06508	0.00000	−0.06508	−204.46	−0.4371	2.121	−109
−108	4.861	8.842	0.000	8.842	−25.963	0.001	−25.962	−0.06439	0.00000	−0.06439	−204.19	−0.4360	2.320	−108
−107	5.314	8.868	0.000	8.868	−25.772	0.001	−25.721	−0.06370	0.00000	−0.06370	−203.83	−0.4350	2.536	−107
−106	5.806	8.893	0.000	8.893	−25.481	0.001	−25.480	−0.06302	0.00000	−0.06302	−203.46	−0.4339	2.771	−106
−105	6.340	8.919	0.000	8.919	−25.240	0.001	−25.239	−0.06234	0.00000	−0.06234	−203.09	−0.4329	3.026	−105
−104	6.920	8.944	0.000	8.944	−25.000	0.001	−24.999	−0.06167	0.00000	−0.06167	−202.72	−0.4318	3.303	−104
−103	7.549	8.970	0.000	8.970	−24.759	0.001	−24.758	−0.06099	0.00000	−0.06099	−202.35	−0.4308	3.603	−103
−102	8.232	8.995	0.000	8.995	−24.518	0.001	−24.517	−0.06032	0.00000	−0.06032	−201.98	−0.4298	3.929	−102
−101	8.972	9.020	0.000	9.020	−24.278	0.001	−24.277	−0.05964	0.00000	−0.05964	−201.61	−0.4287	4.283	−101
−100	9.772	9.046	0.000	9.046	−24.037	0.001	−24.036	−0.05897	0.00000	−0.05897	−201.23	−0.4277	4.666	−100
−99	10.63	9.071	0.000	9.071	−23.797	0.001	−23.796	−0.05830	0.00000	−0.05830	−200.86	−0.4266	5.081	−99
−98	11.57	9.097	0.000	9.097	−23.556	0.001	−23.555	−0.05764	0.00000	−0.05764	−200.48	−0.4256	5.530	−98
−97	12.59	9.122	0.000	9.122	−23.316	0.001	−23.315	−0.05697	0.00000	−0.05697	−200.11	−0.4245	6.016	−97

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^6$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^4$	
−96	1.369	9.147	0.000	9.147	−23.075	0.001	−23.074	−0.05631	0.00000	−0.05631	−199.73	−0.4235	0.6542	−96
−95	1.489	9.173	0.000	9.173	−22.835	0.002	−22.833	−0.05565	0.00000	−0.05565	−199.35	−0.4225	0.7111	−95
−94	1.617	9.198	0.000	9.198	−22.594	0.002	−22.592	−0.05500	0.00000	−0.05500	−198.97	−0.4214	0.7725	−94
−93	1.756	9.224	0.000	9.224	−22.353	0.002	−22.341	−0.05434	0.00000	−0.05434	−198.59	−0.4204	0.8388	−93
−92	1.906	9.249	0.000	9.249	−22.113	0.002	−22.111	−0.05369	0.00001	−0.05368	−198.21	−0.4193	0.9105	−92
−91	2.068	9.275	0.000	9.275	−21.872	0.002	−21.870	−0.05303	0.00001	−0.05302	−197.83	−0.4183	0.9879	−91
−90	2.242	9.300	0.000	9.300	−21.631	0.002	−21.629	−0.05237	0.00001	−0.05236	−197.44	−0.4173	1.071	−90
−89	2.430	9.325	0.000	9.325	−21.391	0.003	−21.388	−0.05172	0.00001	−0.05171	−197.06	−0.4162	1.161	−89
−88	2.634	9.351	0.000	9.351	−21.150	0.003	−21.147	−0.05107	0.00001	−0.05106	−196.67	−0.4152	1.259	−88
−87	2.852	9.376	0.000	9.376	−20.909	0.003	−20.906	−0.05042	0.00001	−0.05041	−196.29	−0.4142	1.363	−87
−86	3.089	9.401	0.000	9.401	−20.669	0.003	−20.666	−0.04978	0.00001	−0.04977	−195.90	−0.4131	1.476	−86
−85	3.342	9.426	0.000	9.426	−20.428	0.003	−20.425	−0.04913	0.00001	−0.04912	−195.51	−0.4121	1.597	−85
−84	3.615	9.451	0.000	9.451	−20.188	0.004	−20.184	−0.04849	0.00001	−0.04848	−195.12	−0.4110	1.728	−84
−83	3.909	9.477	0.000	9.477	−19.947	0.004	−19.943	−0.04785	0.00001	−0.04784	−194.73	−0.4100	1.868	−83
−82	4.225	9.502	0.000	9.502	−19.706	0.004	−19.702	−0.04721	0.00001	−0.04720	−194.34	−0.4090	2.019	−82
−81	4.564	9.527	0.000	9.527	−19.466	0.005	−19.461	−0.04658	0.00001	−0.04657	−193.95	−0.4079	2.181	−81
−80	4.930	9.553	0.000	9.553	−19.225	0.005	−19.220	−0.04595	0.00001	−0.04594	−193.55	−0.4069	2.356	−80
−79	5.322	9.578	0.000	9.578	−18.984	0.005	−18.979	−0.04532	0.00002	−0.04530	−193.16	−0.4059	2.543	−79
−78	5.742	9.604	0.000	9.604	−18.744	0.006	−18.738	−0.04469	0.00002	−0.04467	−192.76	−0.4048	2.744	−78
−77	6.193	9.629	0.000	9.629	−18.503	0.006	−18.497	−0.04406	0.00002	−0.04404	−192.37	−0.4038	2.960	−77
−76	6.677	9.654	0.000	9.654	−18.263	0.007	−18.256	−0.04343	0.00002	−0.04341	−191.97	−0.4027	3.192	−76
−75	7.196	9.680	0.000	9.680	−18.022	0.007	−18.015	−0.04280	0.00002	−0.04278	−191.57	−0.4017	3.441	−75
−74	7.753	9.705	0.000	9.705	−17.782	0.008	−17.774	−0.04218	0.00003	−0.04215	−191.17	−0.4007	3.707	−74
−73	8.349	9.730	0.000	9.730	−17.541	0.008	−17.533	−0.04155	0.00003	−0.04152	−190.77	−0.3996	3.992	−73
−72	8.990	9.756	0.000	9.756	−17.301	0.009	−17.292	−0.04093	0.00003	−0.04090	−190.37	−0.3986	4.298	−72
−71	9.675	9.781	0.000	9.781	−17.060	0.010	−17.050	−0.04031	0.00003	−0.04028	−189.97	−0.3975	4.625	−71
−70	10.40	9.806	0.000	9.806	−16.820	0.011	−16.809	−0.03969	0.00003	−0.03966	−189.56	−0.3965	4.976	−70
−69	11.19	9.831	0.000	9.831	−16.579	0.011	−16.568	−0.03908	0.00004	−0.03904	−189.16	−0.3954	5.351	−69
−68	12.03	9.856	0.000	9.856	−16.339	0.012	−16.327	−0.03846	0.00004	−0.03842	−188.75	−0.3944	5.752	−68
−67	12.92	9.882	0.000	9.882	−16.098	0.013	−16.085	−0.03785	0.00004	−0.03781	−188.35	−0.3934	6.181	−67
−66	13.88	9.907	0.000	9.907	−15.858	0.014	−15.844	−0.03724	0.00004	−0.03720	−187.94	−0.3924	6.640	−66
−65	14.91	9.932	0.000	9.932	−15.617	0.015	−15.602	−0.03663	0.00005	−0.03658	−187.53	−0.3913	7.130	−65

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^6$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
−64	1.601	9.958	0.000	9.958	−15.377	0.016	−15.361	−0.03602	0.00005	−0.03597	−187.12	−0.3903	0.7654	−64
−63	1.718	9.983	0.000	9.983	−15.137	0.018	−15.119	−0.03541	0.00005	−0.03536	−186.71	−0.3893	0.8213	−63
−62	1.843	10.009	0.000	10.009	−14.896	0.019	−14.877	−0.03481	0.00006	−0.03475	−186.30	−0.3882	0.8810	−62
−61	1.976	10.034	0.000	10.034	−14.656	0.020	−14.636	−0.03420	0.00006	−0.03414	−185.89	−0.3872	0.9447	−61
−60	2.118	10.059	0.000	10.059	−14.416	0.022	−14.394	−0.03360	0.00006	−0.03354	−185.47	−0.3861	1.0127	−60
−59	2.269	10.085	0.000	10.085	−14.175	0.023	−14.152	−0.03300	0.00007	−0.03293	−185.06	−0.3851	1.0852	−59
−58	2.431	10.110	0.000	10.110	−13.935	0.025	−13.910	−0.03240	0.00007	−0.03233	−184.64	−0.3841	1.1624	−58
−57	2.603	10.135	0.000	10.135	−13.695	0.027	−13.668	−0.03180	0.00008	−0.03172	−184.23	−0.3830	1.2447	−57
−56	2.786	10.161	0.000	10.161	−13.454	0.029	−13.425	−0.03120	0.00008	−0.03112	−183.81	−0.3820	1.3324	−56
−55	2.982	10.186	0.000	10.186	−13.214	0.031	−13.183	−0.03061	0.00009	−0.03052	−183.39	−0.3810	1.4258	−55
−54	3.190	10.211	0.000	10.211	−12.974	0.033	−12.941	−0.03002	0.00009	−0.02993	−182.97	−0.3799	1.5253	−54
−53	3.411	10.237	0.000	10.237	−12.733	0.035	−12.698	−0.02943	0.00010	−0.02933	−182.55	−0.3789	1.6312	−53
−52	3.646	10.262	0.001	10.263	−12.493	0.038	−12.455	−0.02884	0.00011	−0.02873	−182.13	−0.3779	1.7438	−52
−51	3.897	10.288	0.001	10.289	−12.253	0.041	−12.212	−0.02825	0.00011	−0.02814	−181.71	−0.3769	1.8635	−51
−50	4.163	10.313	0.001	10.314	−12.012	0.043	−11.969	−0.02766	0.00012	−0.02754	−181.29	−0.3758	1.9910	−50
−49	4.446	10.338	0.001	10.339	−11.772	0.046	−11.726	−0.02707	0.00012	−0.02695	−180.87	−0.3748	2.1264	−49
−48	4.747	10.364	0.001	10.365	−11.532	0.049	−11.483	−0.02649	0.00013	−0.02636	−180.44	−0.3738	2.2702	−48
−47	5.066	10.389	0.001	10.390	−11.292	0.053	−11.239	−0.02590	0.00013	−0.02577	−180.02	−0.3728	2.4230	−47
−46	5.406	10.414	0.001	10.415	−11.051	0.056	−10.995	−0.02532	0.00014	−0.02518	−179.59	−0.3717	2.5854	−46
−45	5.766	10.440	0.001	10.441	−10.811	0.060	−10.751	−0.02474	0.00015	−0.02459	−179.16	−0.3707	2.7578	−45
−44	6.149	10.465	0.001	10.466	−10.571	0.064	−10.507	−0.02416	0.00016	−0.02400	−178.73	−0.3696	2.9408	−44
−43	6.555	10.490	0.001	10.491	−10.330	0.068	−10.262	−0.02358	0.00017	−0.02341	−178.30	−0.3686	3.1349	−43
−42	6.985	10.516	0.001	10.517	−10.090	0.073	−10.017	−0.02301	0.00019	−0.02282	−177.87	−0.3676	3.3408	−42
−41	7.441	10.541	0.001	10.542	−9.850	0.078	−9.772	−0.02243	0.00020	−0.02223	−177.44	−0.3665	3.5591	−41
−40	7.925	10.566	0.001	10.567	−9.609	0.083	−9.526	−0.02186	0.00021	−0.02165	−177.01	−0.3655	3.7906	−40
−39	8.437	10.592	0.001	10.593	−9.369	0.089	−9.280	−0.02129	0.00022	−0.02107	−176.58	−0.3645	4.0359	−39
−38	8.980	10.617	0.002	10.619	−9.129	0.094	−9.035	−0.02072	0.00024	−0.02048	−176.14	−0.3634	4.2958	−38
−37	9.556	10.642	0.002	10.644	−8.889	0.100	−8.789	−0.02015	0.00025	−0.01990	−175.71	−0.3624	4.5711	−37
−36	10.16	10.668	0.002	10.670	−8.648	0.106	−8.542	−0.01958	0.00026	−0.01932	−175.27	−0.3614	4.8626	−36
−35	10.81	10.693	0.002	10.695	−8.408	0.113	−8.295	−0.01902	0.00028	−0.01874	−174.84	−0.3604	5.1713	−35
−34	11.49	10.718	0.002	10.720	−8.168	0.121	−8.047	−0.01845	0.00030	−0.01815	−174.40	−0.3593	5.4980	−34
−33	12.21	10.744	0.002	10.746	−7.927	0.128	−7.799	−0.01789	0.00032	−0.01757	−173.96	−0.3583	5.8437	−33

[a]Compiled by John A. Goff and S. Gratch.

Table 1. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^4$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
−32	1.298	10.769	0.002	10.771	−7.687	0.136	−7.551	−0.01733	0.00034	−0.01699	−173.52	−0.3573	0.62093	−32
−31	1.378	10.794	0.002	10.796	−7.447	0.145	−7.302	−0.01677	0.00036	−0.01641	−173.08	−0.3563	0.65979	−31
−30	1.464	10.820	0.002	10.822	−7.207	0.154	−7.053	−0.01621	0.00038	−0.01583	−172.64	−0.3552	0.70046	−30
−29	1.554	10.845	0.003	10.848	−6.966	0.163	−6.803	−0.01565	0.00040	−0.01525	−172.20	−0.3542	0.74365	−29
−28	1.649	10.870	0.003	10.873	−6.726	0.173	−6.553	−0.01509	0.00043	−0.01466	−171.75	−0.3531	0.78928	−28
−27	1.750	10.896	0.003	10.899	−6.486	0.184	−6.302	−0.01453	0.00045	−0.01408	−171.31	−0.3521	0.83748	−27
−26	1.856	10.921	0.003	10.924	−6.246	0.196	−6.050	−0.01398	0.00048	−0.01350	−170.86	−0.3511	0.88838	−26
−25	1.969	10.946	0.004	10.950	−6.005	0.207	−5.798	−0.01342	0.00051	−0.01291	−170.42	−0.3500	0.94212	−25
−24	2.087	10.972	0.004	10.976	−5.765	0.219	−5.546	−0.01287	0.00054	−0.01233	−169.97	−0.3490	0.99885	−24
−23	2.212	10.997	0.004	11.001	−5.525	0.232	−5.293	−0.01232	0.00057	−0.01175	−169.52	−0.3480	1.0587	−23
−22	2.344	11 022	0.004	11.026	−5.285	0.246	−5.039	−0.01177	0.00061	−0.01116	−169.07	−0.3469	1.1219	−22
−21	2.483	11.048	0.004	11.052	−5.044	0.261	−4.783	−0.01122	0.00064	−0.01058	−168.62	−0.3459	1.1885	−21
−20	2.630	11.073	0.005	11.078	−4.804	0.277	−4.527	−0.01067	0.00068	−0.00999	−168.17	−0.3449	1.2587	−20
−19	2.785	11.098	0.005	11.103	−4.564	0.293	−4.271	−0.01012	0.00072	−0.00940	−167.72	−0.3439	1.3327	−19
−18	2.948	11.124	0.005	11.129	−4.324	0.310	−4.014	−0.00958	0.00076	−0.00882	−167.26	−0.3428	1.4107	−18
−17	3.120	11.149	0.006	11.155	−4.083	0.328	−3.755	−0.00904	0.00080	−0.00824	−166.81	−0.3418	1.4929	−17
−16	3.301	11.174	0.006	11.180	−3.843	0.348	−3.495	−0.00850	0.00084	−0.00766	−166.35	−0.3408	1.5795	−16
−15	3.491	11.200	0.006	11.206	−3.603	0.368	−3.235	−0.00796	0.00089	−0.00707	−165.90	−0.3398	1.6706	−15
−14	3.692	11.225	0.007	11.232	−3.363	0.389	−2.974	−0.00743	0.00094	−0.00649	−165.44	−0.3387	1.7666	−14
−13	3.903	11.250	0.007	11.257	−3.123	0.412	−2.711	−0.00689	0.00099	−0.00590	−164.98	−0.3377	1.8677	−13
−12	4.125	11.275	0.008	11.283	−2.882	0.436	−2.446	−0.00636	0.00104	−0.00532	−164.52	−0.3367	1.9740	−12
−11	4.359	11.301	0.008	11.309	−2.642	0.461	−2.181	−0.00582	0.00109	−0.00473	−164.06	−0.3357	2.0859	−11
−10	4.606	11.326	0.008	11.334	−2.402	0.487	−1.915	−0.00529	0.00115	−0.00414	−163.60	−0.3346	2.2035	−10
−9	4.865	11.351	0.008	11.359	−2.162	0.514	−1.648	−0.00475	0.00121	−0.00354	−163.14	−0.3336	2.3272	−9
−8	5.137	11.376	0.009	11.385	−1.922	0.543	−1.379	−0.00422	0.00128	−0.00294	−162.67	−0.3326	2.4573	−8
−7	5.423	11.401	0.010	11.411	−1.681	0.574	−1.107	−0.00369	0.00135	−0.00234	−162.21	−0.3316	2.5940	−7
−6	5.724	11.427	0.010	11.437	−1.441	0.606	−0.835	−0.00316	0.00142	−0.00174	−161.74	−0.3305	2.7377	−6
−5	6.040	11.452	0.011	11.463	−1.201	0.639	−0.562	−0.00263	0.00149	−0.00114	−161.28	−0.3295	2.8886	−5
−4	6.371	11.477	0.012	11.489	−0.961	0.675	−0.286	−0.00210	0.00157	−0.00053	−160.81	−0.3285	3.0472	−4
−3	6.720	11.502	0.013	11.515	−0.721	0.712	−0.009	−0.00157	0.00165	0.00008	−160.34	−0.3275	3.2137	−3
−2	7.085	11.528	0.013	11.541	−0.480	0.751	0.271	−0.00105	0.00174	0.00069	−159.87	−0.3264	3.3885	−2
−1	7.469	11.553	0.014	11.567	−0.240	0.792	0.552	−0.00052	0.00183	0.00131	−159.40	−0.3254	3.5720	−1

[a]Compiled by John A. Goff and S. Gratch.

Table 1. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^2$	
0	0.7872	11.578	0.015	11.593	0.000	0.835	0.835	0.00000	0.00192	0.00192	−158.93	−0.3244	3.7645	0
1	0.8295	11.604	0.015	11.619	0.240	0.880	1.120	0.00052	0.00202	0.00254	−158.46	−0.3234	3.9666	1
2	0.8739	11.629	0.016	11.645	0.480	0.928	1.408	0.00104	0.00212	0.00316	−157.99	−0.3223	4.1785	2
3	0.9204	11.654	0.017	11.671	0.721	0.977	1.698	0.00156	0.00223	0.00379	−157.52	−0.3213	4.4007	3
4	0.9692	11.679	0.018	11.697	0.961	1.030	1.991	0.00208	0.00234	0.00442	−157.04	−0.3203	4.6337	4
5	1.020	11.705	0.019	11.724	1.201	1.085	2.286	0.00260	0.00246	0.00506	−156.57	−0.3193	4.8779	5
6	1.074	11.730	0.020	11.750	1.441	1.142	2.583	0.00312	0.00258	0.00570	−156.09	−0.3182	5.1339	6
7	1.130	11.756	0.021	11.777	1.681	1.202	2.883	0.00364	0.00271	0.00635	−155.61	−0.3172	5.4022	7
8	1.199	11.781	0.022	11.803	1.922	1.266	3.188	0.00415	0.00285	0.00700	−155.13	−0.3162	5.6832	8
9	1.251	11.806	0.024	11.830	2.162	1.332	3.494	0.00467	0.00299	0.00766	−154.65	−0.3152	5.9776	9
10	1.315	11.831	0.025	11.856	2.402	1.401	3.803	0.00518	0.00314	0.00832	−154.17	−0.3141	6.2858	10
11	1.383	11.857	0.026	11.883	2.642	1.474	4.116	0.00569	0.00330	0.00899	−153.69	−0.3131	6.6085	11
12	1.454	11.882	0.028	11.910	2.882	1.550	4.432	0.00620	0.00346	0.00966	−153.21	−0.3121	6.9462	12
13	1.528	11.907	0.029	11.936	3.123	1.630	4.753	0.00671	0.00363	0.01034	−152.73	−0.3111	7.2997	13
14	1.606	11.933	0.030	11.963	3.363	1.713	5.076	0.00721	0.00380	0.01101	−152.24	−0.3100	7.6696	14
15	1.687	11.958	0.032	11.990	3.603	1.800	5.403	0.00772	0.00399	0.01171	−151.76	−0.3090	8.0565	15
16	1.772	11.983	0.034	12.017	3.843	1.892	5.735	0.00822	0.00418	0.01240	−151.27	−0.3080	8.4612	16
17	1.861	12.009	0.035	12.044	4.083	1.988	6.071	0.00873	0.00438	0.01311	−150.78	−0.3070	8.8843	17
18	1.953	12.034	0.038	12.072	4.324	2.088	6.412	0.00923	0.00459	0.01382	−150.29	−0.3059	9.3267	18
19	2.051	12.059	0.040	12.099	4.564	2.192	6.756	0.00973	0.00481	0.01454	−149.80	−0.3049	9.7889	19
20	2.152	12.084	0.042	12.126	4.804	2.302	7.106	0.01023	0.00504	0.01527	−149.31	−0.3039	10.272	20
21	2.258	12.110	0.044	12.154	5.044	2.416	7.460	0.01073	0.00528	0.01601	−148.82	−0.3029	10.777	21
22	2.369	12.135	0.046	12.181	5.284	2.536	7.820	0.01123	0.00553	0.01676	−148.33	−0.3018	11.305	22
23	2.485	12.160	0.049	12.209	5.525	2.661	8.186	0.01173	0.00579	0.01752	−147.84	−0.3008	11.856	23
24	2.606	12.186	0.051	12.237	5.765	2.792	8.557	0.01223	0.00607	0.01830	−147.34	−0.2998	12.431	24
25	2.733	12.211	0.054	12.265	6.005	2.929	8.934	0.01273	0.00635	0.01908	−146.85	−0.2988	13.032	25
26	2.865	12.236	0.057	12.293	6.245	3.072	9.317	0.01322	0.00665	0.01987	−146.35	−0.2977	13.659	26
27	3.003	12.262	0.059	12.321	6.485	3.221	9.706	0.01372	0.00696	0.02068	−145.85	−0.2967	14.313	27
28	3.147	12.287	0.062	12.349	6.726	3.377	10.103	0.01421	0.00728	0.02149	−145.36	−0.2957	14.996	28
29	3.297	12.312	0.065	12.377	6.966	3.540	10.506	0.01470	0.00761	0.02231	−144.86	−0.2947	15.709	29
30	3.454	12.338	0.068	12.406	7.206	3.709	10.915	0.01519	0.00796	0.02315	−144.36	−0.2936	16.452	30
31	3.617	12.363	0.071	12.434	7.446	3.887	11.333	0.01568	0.00832	0.02400	−143.86	−0.2926	17.227	31
32	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	−143.36	−0.2916	18.035	32
*32	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	0.04	0.0000	18.037	32*
33	3.944	12.413	0.079	12.492	7.927	4.242	12.169	0.01666	0.00904	0.02570	1.05	0.0020	18.778	33
34	4.107	12.438	0.082	12.520	8.167	4.418	12.585	0.01715	0.00940	0.02655	2.06	0.0041	19.546	34

[a]Compiled by John A. Goff and S. Gratch.

*Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP t(F)	HUMIDITY RATIO $W_s \times 10^3$	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
35	4.275	12.464	0.085	12.549	8.407	4.601	13.008	0.01764	0.00977	0.02741	3.06	0.00061	0 20342	35
36	4.450	12.489	0.089	12.578	8.647	4.791	13.438	0.01812	0.01016	0.02828	4.07	0.0081	0.21166	36
37	4.631	12.514	0.093	12.607	8.887	4.987	13.874	0.01861	0.01056	0.02917	5.07	0.0102	0.22020	37
38	4.818	12.540	0.097	12.637	9.128	5.101	14.319	0.01909	0 01097	0.03006	6.08	0 0122	0.22004	38
39	5.012	12.565	0.101	12.666	9.368	5.403	14.771	0.01957	0.01139	0.03096	7.08	0.0142	0.23819	39
40	5.213	12.590	0.105	12.695	9.608	5.622	15.230	0.02005	0.01183	0.03188	8.09	0.0162	0.24767	40
41	5.421	12.616	0.109	12.725	9.848	5.849	15.697	0.02053	0.01228	0.03281	9.09	0.0182	0.25748	41
42	5.638	12.641	0.114	12.755	10.088	6.084	16.172	0.02101	0.01275	0.03376	10.09	0.0202	0.26763	42
43	5.860	12.666	0.119	12.785	10.329	6.328	16.657	0.02149	0.01323	0.03472	11.10	0.0222	0.27813	43
44	6.091	12.691	0.124	12.815	10.569	6.580	17.149	0.02197	0.01373	0.03570	12.10	0.0242	0.28899	44
45	6.331	12.717	0.129	12.846	10.809	6.841	17.650	0.02245	0.01425	0.03670	13.10	0.0262	0.30023	45
46	6.578	12.742	0.134	12.876	11.049	7.112	18.161	0.02293	0.01478	0.03771	14.10	0.0282	0.31185	46
47	6.835	12.767	0.140	12.907	11.289	7.391	18.680	0.02340	0.01534	0.03874	15.11	0.0302	0.32386	47
48	7.100	12.792	0.146	12.938	11.530	7.681	19.211	0.02387	0.01591	0.03978	16.11	0.0321	0.33629	48
49	7.374	12.818	0.151	12.969	11.770	7.981	19.751	0.02434	0.01650	0.04084	17.11	0.0341	0.34913	49
50	7.658	12.843	0.158	13.001	12.010	8.291	20.301	0.02481	0.01711	0.04192	18.11	0.0361	0.36240	50
51	7.952	12.868	0.164	13.032	12.250	8.612	20.862	0.02528	0.01774	0.04302	19.11	0.0381	0.37611	51
52	8.256	12 894	0.170	13.064	12.491	8.945	21.436	0.02575	0.01839	0.04414	20.11	0.0400	0.39028	52
53	8.569	12.919	0.178	13.097	12.731	9.289	22.020	0.02622	0.01906	0.04528	21.12	0.0420	0.40492	53
54	8.894	12.944	0.185	13.129	12.971	9.644	22.615	0.02669	0.01976	0.04645	22.12	0.0439	0.42004	54
55	9 229	12.970	0.192	13.162	13.211	10.01	23.22	0.02716	0.02047	0.04763	23.12	0.0459	0.43565	55
56	9.575	12.995	0.200	13.195	13.452	10.39	23.84	0.02762	0.02121	0.04883	24.12	0.0478	0.45176	56
57	9.934	13.020	0.208	13.228	13.692	10.79	24.48	0.02809	0.02197	0.05006	25.12	0.0497	0.46840	57
58	10.30	13.045	0.216	13.261	13.932	11.19	25.12	0.02855	0.02276	0.05131	26.12	0.0517	0.48558	58
59	10.69	13.071	0.224	13.295	14.172	11.61	25.78	0.02902	0.02357	0.05259	27.12	0.0536	0.50330	59
60	11.08	13.096	0.233	13.329	14.413	12.05	26.46	0.02948	0.02441	0.05389	28.12	0.0555	0.52159	60
61	11.49	13.121	0.242	13.363	14.653	12.50	27.15	0.02994	0.02527	0.05521	29.12	0.0574	0.54047	61
62	11.91	13.147	0.251	13.398	14.893	12.96	27.85	0.03040	0.02616	0.05656	30.12	0.0594	0.55994	62
63	12.35	13.172	0.261	13.433	15.134	13.44	28.57	0.03086	0.02708	0.05794	31.12	0.0613	0.58002	63
64	12.80	13.197	0.271	13.468	15.374	13.94	29.31	0.03132	0.02803	0.05935	32.12	0.0632	0.60073	64
65	13.26	13.222	0.282	13.504	15.614	14.45	30.06	0.03177	0.02901	0.06078	33.11	0.0651	0.62209	65
66	13.74	13.247	0.292	13.539	15.855	14.98	30.83	0.03223	0.03002	0.06225	34.11	0.0670	0.64411	66
67	14.24	13.273	0.303	13.576	16.095	15.53	31.62	0.03269	0.03106	0.06375	35.11	0.0689	0.66681	67
68	14.75	13.298	0.315	13.613	16.335	16.09	32.42	0.03314	0.03213	0.06527	36.11	0.0708	0.69019	68
69	15.28	13.323	0.327	13.650	16.576	16.67	33.25	0.03360	0.03323	0.06683	37.11	0.0727	0.71430	69

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^2$	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
70	1.582	13.348	0.339	13.687	16.816	17.27	34.09	0.03405	0.03437	0.06842	38.11	0.0746	0.73915	70
71	1.639	13.373	0.351	13.724	17.056	17.89	34.95	0.03450	0.03554	0.07004	39.11	0.0765	0.76475	71
72	1.697	13.398	0.364	13.762	17.297	18.53	35.83	0.03495	0.03675	0.07170	40.11	0.0784	0.79112	72
73	1.757	13.424	0.377	13.801	17.537	19.20	36.74	0.03540	0.03800	0.07340	41.11	0.0803	0.81828	73
74	1.819	13.449	0.392	13.841	17.778	19.88	37.66	0.03585	0.03928	0.07513	42.10	0.0821	0.84624	74
75	1.882	13.474	0.407	13.881	18.018	20.59	38.61	0.03630	0.04060	0.07690	43.10	0.0840	0.87504	75
76	1.948	13.499	0.422	13.921	18.259	21.31	39.57	0.03675	0.04197	0.07872	44.10	0.0859	0.90470	76
77	2.016	13.525	0.437	13.962	18.499	22.07	40.57	0.03720	0.04337	0.08057	45.10	0.0877	0.93523	77
78	2.086	13.550	0.453	14.003	18.740	22.84	41.58	0.03765	0.04482	0.08247	46.10	0.0896	0.96665	78
79	2.158	13.575	0.470	14.045	18.980	23.64	42.62	0.03810	0.04631	0.08441	47.10	0.0914	0.99899	79
80	2.233	13.601	0.486	14.087	19.221	24.47	43.69	0.03854	0.04784	0.08638	48.10	0.0933	1.0323	80
81	2.310	13.626	0.504	14.130	19.461	25.32	44.78	0.03899	0.04942	0.08841	49.09	0.0952	1.0665	81
82	2.389	13.651	0.523	14.174	19.702	26.20	45.90	0.03943	0.05105	0.09048	50.09	0.0970	1.1017	82
83	2.471	13.676	0.542	14.218	19.942	27.10	47.04	0.03987	0.05273	0.09260	51.09	0.0989	1.1379	83
84	2.555	13.702	0.560	14.262	20.183	28.04	48.22	0.04031	0.05446	0.09477	52.09	0.1007	1.1752	84
85	2.642	13.727	0.581	14.308	20.423	29.01	49.43	0.04075	0.05624	0.09699	53.09	0.1025	1.2135	85
86	2.731	13.752	0.602	14.354	20.663	30.00	50.66	0.04119	0.05807	0.09926	54.08	0.1043	1.2529	86
87	2.824	13.777	0.624	14.401	20.904	31.03	51.93	0.04163	0.05995	0.10158	55.08	0.1062	1.2934	87
88	2.919	13.803	0.645	14.448	21.144	32.09	53.23	0.04207	0.06189	0.10396	56.08	0.1080	1.3351	88
89	3.017	13.828	0.668	14.496	21.385	33.18	54.56	0.04251	0.06389	0.10640	57.08	0.1098	1.3779	89
90	3.118	13.853	0.692	14.545	21.625	34.31	55.93	0.04295	0.06596	0.10890	58.08	0.1116	1.4219	90
91	3.223	13.879	0.716	14.595	21.865	35.47	57.33	0.04339	0.06807	0.11146	59.07	0.1135	1.4671	91
92	3.330	13.904	0.741	14.645	22.106	36.67	58.78	0.04382	0.07025	0.11407	60.07	0.1153	1.5135	92
93	3.441	13.929	0.768	14.697	22.346	37.90	60.25	0.04426	0.07249	0.11675	61.07	0.1171	1.5612	93
94	3.556	13.954	0.795	14.749	22.587	39.18	61.77	0.04469	0.07480	0.11949	62.07	0.1188	1.6102	94
95	3.673	13.980	0.822	14.802	22.827	40.49	63.32	0.04513	0.07718	0.12231	63.07	0.1206	1.6606	95
96	3.795	14.005	0.851	14.856	23.068	41.85	64.92	0.04556	0.07963	0.12519	64.06	0.1224	1.7123	96
97	3.920	14.030	0.881	14.911	23.308	43.24	66.55	0.04600	0.08215	0.12815	65.06	0.1242	1.7654	97
98	4.049	14.056	0.911	14.967	23.548	44.68	68.23	0.04643	0.08474	0.13117	66.06	0.1260	1.8199	98
99	4.182	14.081	0.942	15.023	23.789	46.17	69.96	0.04686	0.08741	0.13427	67.06	0.1278	1.8759	99
100	4.319	14.106	0.975	15.081	24.029	47.70	71.73	0.04729	0.09016	0.13745	68.06	0.1296	1.9333	100
101	4.460	14.131	1.009	15.140	24.270	49.28	73.55	0.04772	0.09299	0.14071	69.05	0.1314	1.9923	101
102	4.606	14.157	1.043	15.200	24.510	50.91	75.42	0.04815	0.09591	0.14406	70.05	0.1332	2.0528	102
103	4.756	14.182	1.079	15.261	24.751	52.59	77.34	0.04858	0.09891	0.14749	71.05	0.1350	2.1149	103
104	4.911	14.207	1.117	15.324	24.991	54.32	79.31	0.04900	0.1020	0.1510	72.05	0.1367	2.1786	104

[a]Compiled by John A. Goff and S. Gratch

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO W_s x 10	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap Press In. Hg p_s	
105	0.5070	14.232	1.155	15.387	25.232	56.11	81.34	0.04943	0.1052	0.1546	73.04	0.1385	2.2439	105
106	0.5234	14.258	1.194	15.452	25.472	57.95	83.42	0.04985	0.1085	0.1584	74.04	0.1403	2.3109	106
107	0.5404	14.283	1.235	15.518	25.713	59.85	85.56	0.05028	0.1118	0.1621	75.04	0.1421	2.3797	107
108	0.5578	14.308	1.278	15.586	25.953	61.80	87.76	0.05070	0.1153	0.1660	76.04	0.1438	2.4502	108
109	0.5758	14.333	1.321	15.654	26.194	63.82	90.03	0.05113	0.1189	0.1700	77.04	0.1456	2.5225	109
110	0.5944	14.359	1.365	15.724	26.434	65.91	92.34	0.05155	0.1226	0.1742	78.03	0.1472	2.5966	110
111	0.6135	14.384	1.412	15.796	26.675	68.05	94.72	0.05197	0.1264	0.1784	79.03	0.1491	2.6726	111
112	0.6333	14.409	1.460	15.869	26.915	70.27	97.18	0.05239	0.1302	0.1826	80.03	0.1508	2.7505	112
113	0.6536	14.435	1.509	15.944	27.156	72.55	99.71	0.05281	0.1342	0.1870	81.03	0.1525	2.8304	113
114	0.6746	14.460	1.560	16.020	27.397	74.91	102.31	0.05323	0.1384	0.1916	82.03	0.1543	2.9123	114
115	0.6962	14.485	1.613	16.098	27.637	77.34	104.98	0.05365	0.1426	0.1963	83.02	0.1560	2.9962	115
116	0.7185	14.510	1.668	16.178	27.878	79.85	107.73	0.05407	0.1470	0.2011	84.02	0.1577	3.0821	116
117	0.7415	14.536	1.723	16.259	28.119	82.43	110.55	0.05449	0.1515	0.2060	85.02	0.1595	3.1701	117
118	0.7652	14.561	1.782	16.343	28.359	85.10	113.46	0.05490	0.1562	0.2111	86.02	0.1612	3.2603	118
119	0.7897	14.586	1.842	16.428	28.600	87.86	116.46	0.05532	0.1610	0.2163	87.02	0.1629	3.3527	119
120	0.8149	14.611	1.905	16.516	28.841	90.70	119.54	0.05573	0.1659	0.2216	88.01	0.1646	3.4474	120
121	0.8410	14.637	1.968	16.605	29.082	93.64	122.72	0.05615	0.1710	0.2272	89.01	0.1664	3.5443	121
122	0.8678	14.662	2.034	16.696	29.322	96.66	125.98	0.05656	0.1763	0.2329	90.01	0.1681	3.6436	122
123	0.8955	14.687	2.103	16.790	29.563	99.79	129.35	0.05698	0.1817	0.2387	91.01	0.1698	3.7452	123
124	0.9242	14.712	2.174	16.886	29.804	103.0	132.8	0.05739	0.1872	0.2446	92.01	0.1715	3.8493	124
125	0.9537	14.738	2.247	16.985	30.044	106.4	136.4	0.05780	0.1930	0.2508	93.01	0.1732	3.9558	125
126	0.9841	14.763	2.323	17.086	30.285	109.8	140.1	0.05821	0.1989	0.2571	94.01	0.1749	4.0649	126
127	1.016	14.788	2.401	17.189	30.526	113.4	143.9	0.05862	0.2050	0.2636	95.00	0.1766	4.1765	127
128	1.048	14.813	2.482	17.295	30.766	117.0	147.8	0.05903	0.2113	0.2703	96.00	0.1783	4.2907	128
129	1.082	14.839	2.565	17.404	31.007	120.8	151.8	0.05944	0.2178	0.2772	97.00	0.1800	4.4076	129
130	1.116	14.864	2.652	17.516	31.248	124.7	155.9	0.05985	0.2245	0.2844	98.00	0.1817	4.5272	130
131	1.152	14.889	2.742	17.631	31.489	128.8	160.3	0.06026	0.2314	0.2917	99.00	0.1834	4.6495	131
132	1.189	14.915	2.834	17.749	31.729	133.0	164.7	0.06067	0.2386	0.2993	100.00	0.1851	4.7747	132
133	1.227	14.940	2.930	17.870	31.970	137.3	169.3	0.06108	0.2459	0.3070	101.00	0.1868	4.9028	133
134	1.267	14.965	3.029	17.994	32.211	141.8	174.0	0.06148	0.2536	0.3151	102.00	0.1885	5.0337	134
135	1.308	14.990	3.132	18.122	32.452	146.4	178.9	0.06189	0.2614	0.3233	103.00	0.1902	5.1676	135
136	1.350	15.016	3.237	18.253	32.692	151.2	183.9	0.06229	0.2695	0.3318	104.00	0.1918	5.3046	136
137	1.393	15.041	3.348	18.389	32.933	156.1	189.0	0.06270	0.2778	0.3405	105.00	0.1935	5.4446	137
138	1.439	15.066	3.462	18.528	33.174	161.2	194.4	0.06310	0.2865	0.3496	106.00	0.1952	5.5878	138
139	1.485	15.091	3.580	18.671	33.414	166.5	199.9	0.06350	0.2954	0.3589	107.00	0.1969	5.7342	139

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	Fahr. Temp. t(F)
140	0.1534	15.117	3.702	18.819	33.655	172.0	205.7	0.06390	0.3047	0.3686	107.99	0.1985	5.8838	140
141	0.1584	15.142	3.829	18.971	33.896	177.7	211.6	0.06430	0.3142	0.3785	108.99	0.2002	6.0367	141
142	0.1636	15.167	3.961	19.128	34.136	183.6	217.7	0.06470	0.3241	0.3888	109.99	0.2018	6.1930	142
143	0.1689	15.192	4.098	19.290	34.377	189.7	224.1	0.06510	0.3343	0.3994	110.99	0.2035	6.3527	143
144	0.1745	15.218	4.239	19.457	34.618	196.0	230.6	0.06549	0.3449	0.4104	111.99	0.2051	6.5160	144
145	0.1803	15.243	4.386	19.629	34.859	202.5	237.4	0.06589	0.3559	0.4218	112.99	0.2068	6.6828	145
146	0.1862	15.268	4.539	19.807	35.099	209.3	244.4	0.06629	0.3672	0.4335	113.99	0.2084	6.8532	146
147	0.1924	15.293	4.698	19.991	35.340	216.4	251.7	0.06669	0.3790	0.4457	114.99	0.2101	7.0273	147
148	0.1989	15.319	4.862	20.181	35.581	223.7	259.3	0.06708	0.3912	0.4583	115.99	0.2117	7.2051	148
149	0.2055	15.344	5.033	20.377	35.822	231.3	267.1	0.06748	0.4038	0.4713	116.99	0.2134	7.3867	149
150	0.2125	15.369	5.211	20.580	36.063	239.2	275.3	0.06787	0.4169	0.4848	117.99	0.2150	7.5722	150
151	0.2197	15.394	5.396	20.790	36.304	247.3	283.6	0.06827	0.4304	0.1987	118.99	0.2167	7.7616	151
152	0.2271	15.420	5.587	21.007	36.545	255.9	292.4	0.06866	0.4445	0.5132	119.99	0.2183	7.9550	152
153	0.2349	15.445	5.788	21.233	36.785	264.7	301.5	0.06908	0.4591	0.5282	120.99	0.2200	8.1525	153
154	0.2430	15.470	5.996	21.466	37.026	273.9	310.9	0.06945	0.4743	0.5438	121.99	0.2216	8.3541	154
155	0.2514	15.496	6.213	21.709	37.267	283.5	320.8	0.06984	0.4901	0.5599	122.99	0.2232	8.5599	155
156	0.2602	15.521	6.439	21.960	37.508	293.5	331.0	0.07023	0.5066	0.5768	123.99	0.2248	8.7701	156
157	0.2693	15.546	6.675	22.221	37.749	303.9	341.7	0.07062	0.5237	0.5943	124.99	0.2265	8.9846	157
158	0.2788	15.571	6.922	22.493	37.990	314.7	352.7	0.07101	0.5415	0.6125	125.99	0.2281	9.2036	158
159	0.2887	15.597	7.178	22.775	38.231	326.0	364.2	0.07140	0.5600	0.6314	127.00	0.2297	9.4271	159
160	0.2990	15.622	7.446	23.068	38.472	337.8	376.3	0.07179	0.5793	0.6511	128.00	0.2313	9.6556	160
161	0.3098	15.647	7.727	23.374	38.713	350.1	388.8	0.07218	0.5994	0.6716	129.00	0.2329	9.8876	161
162	0.3211	15.672	8.020	23.692	38.954	363.0	402.0	0.07257	0.6204	0.6930	130.00	0.2345	10.125	162
163	0.3329	15.698	8.326	24.024	39.195	376.5	415.7	0.07296	0.6423	0.7153	131.00	0.2361	10.367	163
164	0.3452	15.723	8.648	24.371	39.436	390.5	429.9	0.07334	0.6652	0.7385	132.00	0.2377	10.614	164
165	0.3581	15.748	8.985	24.733	39.677	405.3	445.0	0.07373	0.6892	0.7629	133.00	0.2393	10.866	165
166	0.3716	15.773	9.339	25.112	39.918	420.8	460.7	0.07411	0.7142	0.7883	134.00	0.2409	11.123	166
167	0.3858	15.709	9.708	25.507	40.159	437.0	447.2	0.07450	0.7405	0.8150	135.01	0.2426	11.385	167
168	0.4007	15.824	10.098	25.922	40.400	454.0	494.4	0.07488	0.7680	0.8429	136.01	0.2441	11.652	168
169	0.4163	15.849	10.508	26.357	40.641	471.8	512.4	0.07527	0.7969	0.8722	137.01	0.2457	11.925	169
170	0.4327	15.874	10.938	26.812	40.882	490.6	531.5	0.07565	0.8273	0.9030	138.01	0.2473	12.203	170
171	0.4500	15.900	11.391	27.291	41.123	510.4	551.5	0.07603	0.8592	0.9352	139.01	0.2489	12.486	171
172	0.4682	15.925	11.870	27.795	41.364	531.3	572.7	0.07641	0.8927	0.9691	140.01	0.2505	12.775	172
173	0.4875	15.950	12.376	28.326	41.605	553.3	594.9	0.07680	0.9281	1.0049	141.01	0.2521	13.069	173
174	0.5078	15.975	12.911	28.886	41.846	576.5	618.3	0.07718	0.9654	1.0426	142.02	0.2537	13.369	174

[a]Compiled by John A. Goff and S. Gratch.

Table 1. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (Concluded)

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb) (°F) s_w	Vap. Press In. Hg p_s	
175	0.5292	16.001	13.475	29.476	42.087	601.1	643.2	0.07756	1.005	1.083	143.02	0.2553	13.675	175
176	0.5519	16.026	14.074	30.100	42.328	627.1	669.4	0.07794	1.047	1.125	144.02	0.2568	13.987	176
177	0.5760	16.051	14.710	30.761	42.569	654.7	697.3	0.07832	1.091	1.169	145.02	0.2584	14.304	177
178	0.6016	16.076	15.386	31.462	42.810	684.1	726.9	0.07870	1.137	1.216	146.03	0.2600	14.628	178
179	0.6288	16.102	16.104	32.206	43.051	715.2	758.3	0.07908	1.187	1.266	147.03	0.2616	14.958	179
180	0.6578	16.127	16.870	32.997	43.292	748.5	791.8	0.07946	1.240	1.319	148.03	0.2631	15.294	180
181	0.6887	16.152	17.689	33.841	43.534	783.9	827.4	0.07984	1.296	1.376	149.03	0.2647	15.636	181
182	0.7218	16.177	18.565	34.742	43.775	821.9	865.7	0.08021	1.357	1.437	150.04	0.2662	15.985	182
183	0.7572	16.203	19.504	35.707	44.016	862.5	906.5	0.08059	1.421	1.502	151.04	0.2678	16.340	183
184	0.7953	16.228	20.513	36.741	44.257	906.2	950.5	0.08096	1.490	1.571	152.04	0.2693	16.702	184
185	0.8363	16.253	21.601	37.854	44.498	953.2	997.7	0.08134	1.565	1.646	153.05	0.2709	17.071	185
186	0.8805	16.278	22.775	39.053	44.740	1004	1049	0.08171	1.645	1.727	154.05	0.2724	17.446	186
187	0.9283	16.304	24.047	40.351	44.981	1059	1104	0.08208	1.731	1.813	155.05	0.2740	17.828	187
188	0.9802	16.329	25.427	41.756	45.222	1119	1164	0.08245	1.825	1.907	156.06	0.2755	18.217	188
189	1.037	16.354	26.934	43.288	45.463	1184	1229	0.08283	1.928	2.011	157.06	0.2771	18.614	189
190	1.099	16.379	28.580	44.959	45.704	1255	1301	0.08320	2.039	2.122	158.07	0.2786	19.017	190
191	1.166	16.405	30.385	46.790	45.946	1332	1378	0.08357	2.161	2.245	159.07	0.2802	19.427	191
192	1.241	16.430	32.375	48.805	46.187	1418	1464	0.08394	2.296	2.380	160.07	0.2817	19.845	192
193	1.324	16.455	34.581	51.036	46.428	1513	1559	0.08431	2.444	2.528	161.08	0.2833	20.271	193
194	1.416	16.480	37.036	53.516	46.670	1619	1666	0.08468	2.609	2.694	162.08	0.2848	20.704	194
195	1.519	16.506	39.785	56.291	46.911	1737	1784	0.08505	2.794	2.879	163.09	0.2864	21.145	195
196	1.635	16.531	42.885	59.416	47.153	1871	1918	0.08542	3.002	3.087	164.09	0.2879	21.594	196
197	1.767	16.556	46.402	62.958	47.394	2022	2069	0.08579	3.238	3.324	165.10	0.2895	22.050	197
198	1.917	16.581	50.426	67.007	47.636	2195	2243	0.08616	3.507	3.593	166.10	0.2910	22.514	198
199	2.091	16.607	55.074	71.681	47.877	2395	2443	0.08653	3.817	3.904	167.11	0.2925	22.987	199
200	2.295	16.632	60.510	77.142	48.119	2629	2677	0 08689	4.179	4.266	168 11	0.2940	23.468	200

[a]Compiled by John A. Goff and S. Gratch.

In Table 1 there are 15 columns of figures, each column being headed by a suitable symbol. In the following sub-paragraphs are given brief explanations of the data in Table 1 under the appropriate column headings.

t(F) = Fahrenheit temperature defined in terms of absolute temperature T by the relation,

$$T = t + 459.69 \tag{1}$$

This particular Fahrenheit scale differs slightly from that derived from the International Centigrade Scale t(C) by the definition,

$$t(\mathrm{F}) = 1.8t(\mathrm{C}) + 32 \tag{2}$$

However, the maximum difference between the two Fahrenheit scales appears not to exceed 0.01 Fahrenheit deg in the range 32 to 212 F.

W_s = humidity ratio *at saturation.* By humidity ratio is meant the ratio, by weight, of water vapor to dry air, pounds of water vapor per pound of dry air. By saturation is meant the point where coexistence of the vapor phase (moist air) with a condensed phase (liquid or solid) is possible at the given temperature and pressure (standard atmospheric pressure in the case of Table 1). At given values of temperature and pressure the humidity ratio W can have any value from zero to W_s.

v_a = specific volume of dry air, cubic feet per pound.

$v_{as} = v_s - v_a$, the difference between the volume of moist air *at saturation,* per pound of dry air, and the specific volume of the dry air itself, cubic feet per pound of dry air.

v_s = volume of moist air *at saturation* per pound of dry air, cubic feet per pound of dry air.

h_a = specific enthalpy of dry air, Btu per pound. It will be noticed that the specific enthalpy of dry air has been assigned the value zero at 0 F, standard atmospheric pressure. The energy unit Btu is related to the foot-pound, though not exactly, by definition, as follows: 1 Btu = 778.18 ft-lb.

$h_{as} = h_s - h_a$, the difference between the enthalpy of moist air *at saturation,* per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_s = enthalpy of moist air *at saturation* per pound of dry air, Btu per pound of dry air.

s_a = specific entropy of dry air, Btu per (pound) (F). It will be noticed that the specific entropy of dry air has been assigned the value zero at 0 F and standard atmospheric pressure.

$s_{as} = s_s - s_a$, the difference between the entropy of moist air *at saturation,* per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (F).

s_s = entropy of moist air *at saturation* per pound of dry air, Btu per (pound of dry air) (F).

h_w = specific enthalpy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per pound of water. The specific enthalpy of liquid water has been assigned the value zero at 32 F, saturation pressure. It will be noticed that, under this assignment, the specific enthalpy of liquid water at 32 F, standard atmospheric pressure, assumes the value 0.04 Btu/lb$_w$.

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (F). The specific entropy of liquid water has been assigned the value zero at 32 F, saturation pressure. It will be noticed that, under this assignment, the specific entropy of liquid water at 32 F, standard atmospheric pressure, is also zero, though not exactly.

Table 2. Thermodynamic Properties of Water at Saturation[a]

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^7$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap $v_{ig} \times 10^{-8}$	Sat. Vapor $v_g \times 10^{-8}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−160	0.4949	1.008	0.01722	36.07	36.07	−222.05	1212.43	990.38	−0.4907	4.0456	3.5549	−160
−159	0.5592	1.138	0 01722	32.03	32.03	−221.73	1212.55	990.82	−0.4896	4.0325	3.5429	−159
−158	0.6312	1.285	0.01723	28.47	28.47	−221.41	1212.67	991.26	−0.4886	4.0196	3.5310	−158
−157	0.7121	1.450	0 01723	25.32	25 32	−221.09	1212.79	991.70	−0.4875	4.0067	3.5192	−157
−156	0.8026	1.634	0.01723	22.54	22.54	−220.76	1212.90	992.14	−0.4864	3.9939	3.5075	−156
−155	0.9040	1.840	0 01723	20 08	20 08	−220.44	1213.02	992.58	−0.4854	3.9812	3.4958	−155
−154	1.017	2.072	0 01723	17 90	17.90	−220.12	1213.15	993 03	−0.4843	3.9685	3.4842	−154
−153	1.144	2.329	0.01723	15.97	15.97	−219.79	1213.26	993.47	−0.4832	3.9559	3.4727	−153
−152	1.286	2.618	0.01723	14.26	14.26	−219.47	1213.38	993.91	−0.4822	3.9435	3.4613	−152
−151	1.444	2.939	0.01723	12.74	12.74	−219.14	1213.49	994.35	−0.4811	3.9311	3.4500	−151
−150	1.620	3.298	0.01723	11.39	11.39	−218.82	1213.62	994.80	−0.4801	3.9188	3.4387	−150
−149	1.816	3.698	0.01723	10.19	10.19	−218.49	1213.73	995.24	−0.4790	3.9065	3.4275	−149
−148	2.035	4.143	0.01724	9.123	9.123	−218.16	1213.84	995 68	−0.4780	3.8944	3.4164	−148
−147	2.278	4.639	0.01724	8.174	8.174	−217.83	1213.95	996.12	−0.4769	3.8823	3.4054	−147
−146	2.549	5.190	0.01724	7.330	7.330	−217.50	1214 06	996.56	−0.4758	3.8702	3.3944	−146
−145	2.850	5.803	0 01724	6.577	6.577	−217.17	1214.17	997.00	−0.4748	3.8583	3.3835	−145
−144	3.184	6.483	0.01724	5.905	5.905	−216.83	1214.28	997.45	−0.4737	3.8464	3.3727	−144
−143	3.556	7.240	0 01724	5.305	5.305	−216.50	1214.39	997 89	−0.4727	3.8346	3.3619	−143
−142	3.967	8.076	0.01724	4.770	4.770	−216.16	1214.49	998 33	−0.4716	3.8229	3.3513	−142
−141	4.423	9.005	0.01724	4.292	4.292	−215.83	1214.60	998.77	−0.4706	3.8112	3.3406	−141
−140	4.928	10 03	0.01724	3.864	3.864	−215.49	1214 70	999.21	−0.4695	3.7996	3.3301	−140
−139	5.487	11.17	0.01724	3.481	3.481	−215.16	1214.82	999.66	−0.4685	3.7881	3.3196	−139
−138	6.106	12.43	0.01724	3 138	3.138	−214.82	1214.92	1000.10	−0.4674	3.7766	3.3092	−138
−137	6.790	13.82	0.01724	2.831	2.831	−214.48	1215.02	1000.54	−0.4664	3.7653	3.2989	−137
−136	7.546	15.36	0.01725	2.555	2.555	−214.14	1215.12	1000 98	−0.4653	3.7540	3.2887	−136
−135	8.380	17.06	0.01725	2.308	2.308	−213 80	1215.22	1001 42	−0.4643	3 7428	3.2785	−135
−134	9.301	18 94	0 01725	2 086	2.086	−213.46	1215.32	1001.86	−0.4632	3.7315	3.2683	−134
−133	10.32	21.01	0.01725	1.886	1.886	−213.11	1215.42	1002.31	−0.4622	3.7205	3.2583	−133
−132	11.44	23.28	0.01725	1.707	1.707	−212.77	1215.52	1002.75	−0.4611	3.7093	3.2482	−132
−131	12.67	25.79	0.01725	1 545	1.545	−212.43	1215.62	1003.19	−0.4601	3.6984	3.2383	−131
−130	14.03	28.56	0.01725	1.400	1.400	−212 08	1215.71	1003 63	−0.4590	3.6874	3.2284	−130
−129	15.52	31.60	0.01725	1.269	1.269	−211.73	1215 80	1004.07	−0.4580	3.6766	3.2186	−129

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^6$ Lb/Sq In.	Absolute Pressure $p_s \times 10^6$ In. Hg	Specific Volume, cu ft per lb Sat. Solid v_i	Specific Volume Evap. $v_{ig} \times 10^{-7}$	Specific Volume Sat. Vapor $v_g \times 10^{-7}$	Enthalpy, Btu per lb Sat. Solid h_i	Enthalpy Evap. h_{ig}	Enthalpy Sat. Vapor h_g	Entropy, Btu per (Lb) (°F) Sat. Solid s_i	Entropy Evap. s_{ig}	Entropy Sat. Vapor s_g	Fahr. Temp. t(F)
−128	1.716	3.494	0.01726	11.51	11.51	−211.39	1215.91	1004.52	−0.4569	3.6658	3.2089	−128
−127	1.897	3.862	0.01726	10.45	10.45	−211.04	1216.00	1001.96	−0.4559	3.6551	3.1992	−127
−126	2.095	4.265	0.01726	9.489	9.489	−210.69	1216.09	1005.40	−0.4548	3.6444	3.1896	−126
−125	2.312	4.708	0.01726	8.622	8.622	−210.34	1216.18	1005.84	−0.4538	3.6338	3.1800	−125
−124	2.551	5.194	0.01726	7.839	7.839	−209.99	1216.27	1006.28	−0.4527	3.6232	3.1705	−124
−123	2.812	5.726	0.01726	7.131	7.131	−209.64	1216.37	1006.73	−0.4517	3.6127	3.1610	−123
−122	3.099	6.310	0.01726	6.491	6.491	−209.28	1216.45	1007.17	−0.4506	3.6022	3.1516	−122
−121	3.413	6.949	0.01726	5.911	5.911	−208.93	1216.54	1007.61	−0.4496	3.5919	3.1423	−121
−120	3.757	7.649	0.01726	5.386	5.386	−208.58	1216.63	1008.05	−0.4485	3.5815	3.1330	−120
−119	4.133	8.414	0.01726	4.911	4.911	−208.22	1216.71	1008.49	−0.4475	3.5713	3.1238	−119
−118	4.544	9.251	0.01727	4.480	4.480	−207.86	1216.80	1008.94	−0.4464	3.5611	3.1147	−118
−117	4.993	10.17	0.01727	4.088	4.088	−207.51	1216.89	1009.38	−0.4454	3.5510	3.1056	−117
−116	5.484	11.16	0.01727	3.733	3.733	−207.15	1216.97	1009.82	−0.4444	3.5409	3.0965	−116
−115	6.019	12.26	0.01727	3.411	3.411	−206.79	1217.05	1010.26	−0.4433	3.5308	3.0875	−115
−114	6.604	13.44	0.01727	3.118	3.118	−206.43	1217.13	1010.70	−0.4423	3.5209	3.0786	−114
−113	7.241	14.74	0.01727	2.852	2.852	−206.07	1217.21	1011.14	−0.4412	3.5109	3.0697	−113
−112	7.936	16.16	0.01727	2.610	2.610	−205.70	1217.29	1011.59	−0.4402	3.5011	3.0609	−112
−111	8.693	17.70	0.01727	2.389	2.389	−205.34	1217.37	1012.03	−0.4391	3.4912	3.0521	−111
−110	9.517	19.38	0.01728	2.189	2.189	−204.98	1217.45	1012.47	−0.4381	3.4815	3.0434	−110
−109	10.41	21.20	0.01728	2.006	2.006	−204.61	1217.52	1012.91	−0.4370	3.4718	3.0348	−109
−108	11.39	23.19	0.01728	1.839	1.839	−204.24	1217.60	1013.36	−0.4360	3.4621	3.0261	−108
−107	12.45	25.35	0.01728	1.687	1.687	−203.88	1217.68	1013.80	−0.4350	3.4526	3.0176	−107
−106	13.60	27.70	0.01728	1.549	1.549	−203.51	1217.75	1014.24	−0.4339	3.4430	3.0091	−106
−105	14.86	30.25	0.01728	1.422	1.422	−203.14	1217.82	1014.68	−0.4329	3.4335	3.0006	−105
−104	16.22	33.01	0.01728	1.307	1.307	−202.77	1217.89	1015.12	−0.4318	3.4240	2.9922	−104
−103	17.69	36.02	0.01728	1.201	1.201	−202.40	1217.96	1015.56	−0.4308	3.4146	2.9838	−103
−102	19.29	39.28	0.01728	1.104	1.104	−202.03	1218.04	1016.01	−0.4298	3.4053	2.9755	−102
−101	21.03	42.81	0.01729	1.016	1.016	−201.65	1218.10	1016.45	−0.4287	3.3960	2.9673	−101
−100	22.91	46.64	0.01729	0.9352	0.9352	−201.28	1218.17	1016.89	−0.4277	3.3868	2.9591	−100
−99	24.94	50.79	0.01729	0.8613	0.8613	−200.90	1218.23	1017.33	−0.4266	3.3775	2.9509	−99
−98	27.15	55.28	0.01729	0.7936	0.7936	−200.53	1218.30	1017.77	−0.4256	3.3684	2.9428	−98
−97	29.54	60.14	0.01729	0.7314	0.7314	−200.15	1218.37	1018.22	−0.4246	3.3593	2.9347	−97

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s x 10^6 Lb/Sq In.	In. Hg	Specific Volume, cu ft per lb Sat. Solid v_i	Evap. v_{ig} x 10^{-6}	Sat. Vapor v_g x 10^{-6}	Enthalpy, Btu per lb Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	Fahr. Temp. t(F)
−96	3.212	6.539	0.01729	6.745	6.745	−199.78	1218.44	1018.66	−0.4235	3.3502	2.9267	−96
−95	3.491	7.108	0.01729	6.223	6.223	−199.40	1218.50	1019.10	−0.4225	3.3412	2.9187	−95
−94	3.793	7.722	0.01729	5.743	5.743	−199.02	1218.56	1019.54	−0.4214	3.3322	2.9108	−94
−93	4.118	8.385	0.01730	5.303	5.303	−198.64	1218.62	1019.98	−0.4204	3.3233	2.9029	−93
−92	4.470	9.102	0.01730	4.899	4.899	−198.25	1218.68	1020.43	−0.4194	3.3145	2.8951	−92
−91	4.850	9.876	0.01730	4.528	4.528	−197.87	1218.74	1020.87	−0.4183	3.3056	2.8873	−91
−90	5.260	10.71	0.01730	4.186	4.186	−197.49	1218.80	1021.31	−0.4173	3.2969	2.8796	−90
−89	5.702	11.61	0.01730	3.872	3.872	−197.10	1218.85	1021.75	−0.4162	3.2880	2.8718	−89
−88	6.179	12.58	0.01730	3.583	3.583	−196.72	1218.92	1022.20	−0.4152	3.2794	2.8642	−88
−87	6.692	13.63	0.01730	3.317	3.317	−196.33	1218.97	1022.64	−0.4142	3.2708	2.8566	−87
−86	7.245	14.75	0.01730	3.072	3.072	−195.94	1219.02	1023.08	−0.4131	3.2621	2.8490	−86
−85	7.841	15.96	0.01730	2.846	2.846	−195.56	1219.08	1023.52	−0.4121	3.2536	2.8415	−85
−84	8.482	17.27	0.01731	2.638	2.638	−195.16	1219.12	1023.96	−0.4110	3.2450	2.8340	−84
−83	9.171	18.67	0.01731	2.446	2.446	−194.78	1219.18	1024.40	−0.4100	3.2366	2.8266	−83
−82	9.913	20.18	0.01731	2.270	2.270	−194.38	1219.23	1024.85	−0.4090	3.2282	2.8192	−82
−81	10.71	21.81	0.01731	2.106	2.106	−193.99	1219.28	1025.29	−0.4079	3.2197	2.8118	−81
−80	11.57	23.55	0.01731	1.955	1.955	−193.60	1219.33	1025.73	−0.4069	3.2114	2.8045	−80
−79	12.49	25.42	0.01731	1.816	1.816	−193.20	1219.37	1026.17	−0.4058	3.2030	2.7972	−79
−78	13.47	27.43	0.01731	1.687	1.687	−192.81	1219.43	1026.62	−0.4048	3.1948	2.7900	−78
−77	14.53	29.59	0.01732	1.568	1.568	−192.41	1219.47	1027.06	−0.4038	3.1866	2.7828	−77
−76	15.67	31.91	0.01732	1.458	1.458	−192.01	1219.51	1027.50	−0.4028	3.1784	2.7756	−76
−75	16.89	34.39	0.01732	1.356	1.356	−191.62	1219.56	1027.94	−0.4017	3.1702	2.7685	−75
−74	18.20	37.05	0.01732	1.262	1.262	−191.22	1219.60	1028.38	−0.4007	3.1622	2.7615	−74
−73	19.60	39.91	0.01732	1.175	1.175	−190.82	1219.65	1028.83	−0.3996	3.1540	2.7544	−73
−72	21.10	42.96	0.01732	1.094	1.094	−190.41	1219.68	1029.27	−0.3986	3.1460	2.7474	−72
−71	22.71	46.24	0.01732	1.020	1.020	−190.01	1219.72	1029.71	−0.3976	3.1381	2.7405	−71
−70	24.43	49.74	0.01732	0.9501	0.9501	−189.61	1219.76	1030.15	−0.3965	3.1301	2.7336	−70
−69	26.27	53.49	0.01732	0.8858	0.8858	−189.20	1219.80	1030.60	−0.3955	3.1222	2.7267	−69
−68	28.24	57.51	0.01733	0.8261	0.8261	−188.80	1219.84	1031.04	−0.3945	3.1143	2.7198	−68
−67	30.35	61.80	0.01733	0.7707	0.7707	−188.39	1219.87	1031.48	−0.3934	3.1064	2.7130	−67
−66	32.60	66.38	0.01733	0.7193	0.7193	−187.98	1219.90	1031.92	−0.3924	3.0987	2.7063	−66
−65	35.01	71.23	0.01733	0.6715	0.6715	−187.58	1219.94	1032.36	−0.3914	3.0910	2.6996	−65

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^3$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-5}$	Sat. Vapor $v_g \times 10^{-5}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−64	0.3758	0.7652	0.01733	6.272	6.272	−187.17	1219.98	1032.81	−0.3903	3.0832	2.6929	−64
−63	0.4033	0.8211	0.01733	5.859	5.859	−186.76	1220.01	1033.25	−0.3893	3.0755	2.6862	−63
−62	0.4326	0.8808	0.01734	5.476	5.476	−186.35	1220.04	1033.69	−0.3882	3.0678	2.6796	−62
−61	0.4639	0.9144	0.01734	5.120	5.120	−185.93	1220.06	1034.13	−0.3872	3.0602	2.6730	−61
−60	0.4972	1.012	0.01734	4.788	4.788	−185.52	1220.10	1034.58	−0.3862	3.0526	2.6664	−60
−59	0.5328	1.085	0.01734	4.479	4.479	−185.11	1220.13	1035.02	−0.3851	3.0450	2.6599	−59
−58	0.5708	1.162	0.01734	4.192	4.192	−184.69	1220.15	1035.46	−0.3841	3.0376	2.6535	−58
−57	0.6112	1.244	0.01734	3.925	3.925	−184.28	1220.18	1035.90	−0.3831	3.0301	2.6470	−57
−56	0.6543	1.332	0.01734	3.675	3.675	−183.86	1220.20	1036.34	−0.3820	3.0226	2.6406	−56
−55	0.7001	1.426	0.01734	3.443	3.443	−183.44	1220.23	1036.79	−0.3810	3.0152	2.6342	−55
−54	0.7489	1.525	0.01735	3.226	3.226	−183.02	1220.25	1037.23	−0.3800	3.0079	2.6279	−54
−53	0.8009	1.631	0.01735	3.024	3.024	−182.60	1220.27	1037.67	−0.3789	3.0005	2.6216	−53
−52	0.8562	1.743	0.01735	2.836	2.836	−182.18	1220.29	1038.11	−0.3779	2.9932	2.6153	−52
−51	0.9151	1.863	0.01735	2.660	2.660	−181.76	1220.31	1038.55	−0.3769	2.9860	2.6091	−51
−50	0.9776	1.990	0.01735	2.496	2.496	−181.34	1220.34	1039.00	−0.3758	2.9786	2.6028	−50
−49	1.044	2.126	0.01735	2.343	2.343	−180.92	1220.36	1039.44	−0.3748	2.9715	2.5967	−49
−48	1.115	2.270	0.01736	2.200	2.200	−180.49	1220.37	1039.88	−0.3738	2.9643	2.5905	−48
−47	1.190	2.422	0.01736	2.066	2.066	−180.06	1220.38	1040.32	−0.3727	2.9571	2.5844	−47
−46	1.270	2.585	0.01736	1.941	1.941	−179.64	1220.40	1040.76	−0.3717	2.9501	2.5784	−46
−45	1.354	2.757	0.01736	1.824	1.824	−179.21	1220.42	1041.21	−0.3707	2.9430	2.5723	−45
−44	1.444	2.940	0.01736	1.715	1.715	−178.78	1220.43	1041.65	−0.3696	2.9359	2.5663	−44
−43	1.539	3.134	0.01736	1.612	1.612	−178.35	1220.44	1042.09	−0.3686	2.9289	2.5603	−43
−42	1.641	3.340	0.01736	1.516	1.516	−177.92	1220.45	1042.53	−0.3676	2.9219	2.5543	−42
−41	1.748	3.559	0.01736	1.427	1.427	−177.49	1220.47	1042.98	−0.3665	2.9149	2.5484	−41
−40	1.861	3.790	0.01737	1.343	1.343	−177.06	1220.48	1043.42	−0.3655	2.9080	2.5425	−40
−39	1.982	4.035	0.01737	1.264	1.264	−176.63	1220.49	1043.86	−0.3645	2.9012	2.5367	−39
−38	2.110	4.295	0.01737	1.191	1.191	−176.19	1220.49	1044.30	−0.3634	2.8942	2.5308	−38
−37	2.245	4.570	0.01737	1.122	1.122	−175.76	1220.50	1044.74	−0.3624	2.8874	2.5250	−37
−36	2.388	4.862	0.01737	1.057	1.057	−175.32	1220.51	1045.19	−0.3614	2.8807	2.5193	−36
−35	2.540	5.170	0.01737	0.9961	0.9961	−174.88	1220.51	1045.63	−0.3604	2.8739	2.5135	−35
−34	2.700	5.497	0.01737	0.9391	0.9391	−174.45	1220.52	1046.07	−0.3593	2.8671	2.5078	−34
−33	2.870	5.843	0.01738	0.8857	0.8857	−174.01	1220.52	1046.51	−0.3583	2.8604	2.5021	−33

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^3$ Lb/Sq In.	Absolute Pressure $p_s \times 10^3$ In. Hg	Specific Volume, cu ft per lb Sat. Solid v_i	Specific Volume Evap $v_{ig} \times 10^{-4}$	Specific Volume Sat. Vapor $v_g \times 10^{-4}$	Enthalpy, Btu per lb Sat. Solid h_i	Enthalpy Evap. h_{ig}	Enthalpy Sat. Vapor h_g	Entropy, Btu per (Lb) (°F) Sat. Solid s_i	Entropy Evap. s_{ig}	Entropy Sat. Vapor s_g	Fahr. Temp. t(F)
−32	0.3049	0.6208	0.01738	8.355	8.355	−173.57	1220.52	1046.95	−0.3573	2.8538	2.4965	−32
−31	0.3239	0.6595	0.01738	7.883	7.883	−173.13	1220.53	1047.40	−0.3562	2.8470	2.4908	−31
−30	0.3440	0.7003	0.01738	7.441	7.441	−172.68	1220.52	1047.84	−0.3552	2.8405	2.4853	−30
−29	0.3652	0.7435	0.01738	7.025	7.025	−172.24	1220.52	1048.28	−0.3542	2.8339	2.4797	−29
−28	0.3876	0.7891	0.01738	6.634	6.634	−171.80	1220.52	1048.72	−0.3532	2.8274	2.4742	−28
−27	0.4113	0.8373	0.01738	6.267	6.267	−171.35	1220.51	1049.16	−0.3521	2.8207	2.4686	−27
−26	0.4363	0.8882	0.01738	5.921	5.921	−170.91	1220.51	1049.60	−0.3511	2.8143	2.4632	−26
−25	0.4627	0.9420	0.01739	5.596	5.596	−170.46	1220.51	1050.05	−0.3501	2.8078	2.4577	−25
−24	0.4905	0.9987	0.01739	5.290	5.290	−170.01	1220.50	1050.49	−0.3490	2.8013	2.4523	−24
−23	0.5199	1.059	0.01739	5.003	5.003	−169.56	1220.49	1050.93	−0.3480	2.7949	2.4469	−23
−22	0.5509	1.122	0.01739	4.732	4.732	−169.12	1220.49	1051.37	−0.3470	2.7885	2.4415	−22
−21	0.5836	1.188	0.01739	4.477	4.477	−168.66	1220.48	1051.82	−0.3460	2.7822	2.4362	−21
−20	0.6181	1.259	0.01739	4.237	4.237	−168.21	1220.47	1052.26	−0.3449	2.7757	2.4308	−20
−19	0.6545	1.333	0.01739	4.011	4.011	−167.76	1220.46	1052.70	−0.3439	2.7695	2.4256	−19
−18	0.6928	1.410	0.01740	3.797	3.797	−167.31	1220.45	1053.14	−0.3429	2.7632	2.4203	−18
−17	0.7332	1.493	0.01740	3.596	3.596	−166.85	1220.43	1053.58	−0.3418	2.7568	2.4150	−17
−16	0.7757	1.579	0.01740	3.407	3.407	−166.40	1220.42	1054.02	−0.3408	2.7506	2.4098	−16
−15	0.8204	1.670	0.01740	3.228	3.228	−165.94	1220.41	1054.47	−0.3398	2.7444	2.4046	−15
−14	0.8676	1.766	0.01740	3.060	3.060	−165.48	1220.39	1054.91	−0.3388	2.7383	2.3995	−14
−13	0.9172	1.867	0.01740	2.901	2.901	−165.03	1220.38	1055.35	−0.3377	2.7320	2.3943	−13
−12	0.9694	1.974	0.01740	2.750	2.750	−164.57	1220.36	1055.79	−0.3367	2.7259	2.3892	−12
−11	1.024	2.086	0.01740	2.609	2.609	−164.11	1220.34	1056.23	−0.3357	2.7198	2.3841	−11
−10	1.082	2.203	0.01741	2.475	2.475	−163.65	1220.32	1056.67	−0.3347	2.7138	2.3791	−10
−9	1.143	2.327	0.01741	2.349	2.349	−163.18	1220.30	1057.12	−0.3336	2.7076	2.3740	−9
−8	1.207	2.457	0.01741	2.229	2.229	−162.72	1220.28	1057.56	−0.3326	2.7016	2.3690	−8
−7	1.274	2.594	0.01741	2.116	2.116	−162.26	1220.26	1058.00	−0.3316	2.6956	2.3640	−7
−6	1.344	2.737	0.01741	2.010	2.010	−161.79	1220.23	1058.44	−0.3306	2.6896	2.3590	−6
−5	1.419	2.888	0.01741	1.909	1.909	−161.33	1220.21	1058.88	−0.3295	2.6836	2.3541	−5
−4	1.496	3.047	0.01742	1.814	1.814	−160.86	1220.18	1059.32	−0.3285	2.6777	2.3492	−4
−3	1.578	3.213	0.01742	1.723	1.723	−160.39	1220.15	1059.76	−0.3275	2.6718	2.3443	−3
−2	1.664	3.388	0.01742	1.638	1.638	−159.92	1220.13	1060.21	−0.3264	2.6658	2.3394	−2
−1	1.754	3.572	0.01742	1.557	1.557	−159.45	1220.10	1060.65	−0.3254	2.6600	2.3346	−1

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	Absolute Pressure p_s In. Hg	Specific Volume, cu ft per lb: Sat. Solid v_i	Specific Volume: Evap. $v_{ig} \times 10^{-3}$	Specific Volume: Sat. Vapor $v_g \times 10^{-3}$	Enthalpy, Btu per lb: Sat. Solid h_i	Enthalpy: Evap. h_{ig}	Enthalpy: Sat. Vapor h_g	Entropy, Btu per (Lb) (°F): Sat. Solid s_i	Entropy: Evap. s_{ig}	Entropy: Sat. Vapor s_g	Fahr. Temp. t(F)
0	0.01849	0.03764	0.01742	14.81	14.81	−158.98	1220.07	1061.09	−0.3244	2.6541	2.3297	0
1	0.01948	0.03966	0.01742	14.08	14.08	−158.51	1220.04	1061.53	−0.3234	2.6483	2.3249	1
2	0.02052	0.04178	0.01742	13.40	13.40	−158.04	1220.01	1061.97	−0.3224	2.6425	2.3201	2
3	0.02616	0.04400	0.01743	12.75	12.75	−157.56	1219.97	1062.41	−0.3213	2.6367	2.3154	3
4	0.02276	0.04633	0.01743	12.14	12.14	−157.09	1219.94	1062.85	−0.3203	2.6309	2.3106	4
5	0.02396	0.04878	0.01743	11.55	11.55	−156.61	1219.90	1063.29	−0.3193	2.6252	2.3059	5
6	0.02521	0.05134	0.01743	11.00	11.00	−156.14	1219.88	1063.74	−0.3182	2.6194	2.3012	6
7	0.02653	0.05402	0.01743	10.48	10.48	−155.66	1219.84	1064.18	−0.3172	2.6138	2.2966	7
8	0.02791	0.05683	0.01743	9.979	9.979	−155.18	1219.80	1064.62	−0.3162	2.6081	2.2919	8
9	0.02936	0.05977	0.01744	9.507	9.507	−154.70	1219.76	1065.06	−0.3152	2.6025	2.2873	9
10	0.03087	0.06286	0.01744	9.060	9.060	−154.22	1219.72	1065.50	−0.3112	2.5969	2.2827	10
11	0.03246	0.06608	0.01744	8.636	8.636	−153.74	1219.68	1065.94	−0.3131	2.5912	2.2781	11
12	0.03412	0.06946	0.01744	8.234	8.234	−153.26	1219.64	1066.38	−0.3121	2.5857	2.2736	12
13	0.03585	0.07300	0.01744	7.851	7.851	−152.77	1219.59	1066.82	−0.3111	2.5801	2.2690	13
14	0.03767	0.07669	0.01744	7.489	7.489	−152.29	1219.55	1067.26	−0.3101	2.5746	2.2645	14
15	0.03957	0.08056	0.01744	7.144	7.144	−151.80	1219.50	1067.70	−0.3090	2.5690	2.2600	15
16	0.04156	0.08461	0.01745	6.817	6.817	−151.32	1219.46	1068.14	−0.3080	2.5635	2.2555	16
17	0.04363	0.08884	0.01745	6.505	6.505	−150.83	1219.41	1068.58	−0.3070	2.5581	2.2511	17
18	0.04581	0.09326	0.01745	6.210	6.210	−150.34	1219.36	1069.02	−0.3060	2.5526	2.2466	18
19	0.04808	0.09789	0.01745	5.929	5.929	−149.85	1219.31	1069.46	−0.3049	2.5471	2.2422	19
20	0.05045	0.1027	0.01745	5.662	5.662	−149.36	1219.26	1069.90	−0.3039	2.5417	2.2378	20
21	0.05293	0.1078	0.01745	5.408	5.408	−148.87	1219.21	1070.34	−0.3029	2.5364	2.2335	21
22	0.05552	0.1130	0.01746	5.166	5.166	−148.38	1219.16	1070.78	−0.3019	2.5310	2.2291	22
23	0.05823	0.1186	0.01746	4.936	4.936	−147.88	1219.10	1071.22	−0.3008	2.5256	2.2248	23
24	0.06105	0.1243	0.01746	4.717	4.717	−147.39	1219.05	1071.66	−0.2998	2.5203	2.2205	24
25	0.06400	0.1303	0.01746	4.509	4.509	−146.89	1218.98	1072.09	−0.2988	2.5150	2.2162	25
26	0.06708	0.1366	0.01746	4.311	4.311	−146.40	1218.93	1072.53	−0.2978	2.5097	2.2119	26
27	0.07030	0.1431	0.01746	4.122	4.122	−145.90	1218.87	1072.97	−0.2968	2.5045	2.2077	27
28	0.07365	0.1500	0.01746	3.943	3.943	−145.40	1218.81	1073.41	−0.2957	2.4991	2.2034	28
29	0.07715	0.1571	0.01747	3.771	3.771	−144.90	1218.75	1073.85	−0.2947	2.4939	2.1992	29
30	0.08080	0.1645	0.01747	3.608	3.608	−144.40	1218.69	1074.29	−0.2937	2.4887	2.1950	30
31	0.08461	0.1723	0.01747	3.453	3.453	−143.90	1218.63	1074.73	−0.2927	2.4835	2.1908	31
32	0.08858	0.1803	0.01747	3.305	3.305	−143.40	1218.56	1075.16	−0.2916	2.4783	2.1867	32

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	Absolute Pressure p_s In. Hg	Specific Volume, cu ft per lb: Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Enthalpy, Btu per lb: Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy, Btu per (lb) (°F): Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	Fahr. Temp. t(F)
*32	0.088586	0.18036	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32*
33	0.092227	0.18778	0.01602	3180.5	3180.5	1.01	1074.59	1075.60	0.00205	2.1811	2.1831	33
34	0.095999	0.19546	0.01602	3061.7	3061.7	2.01	1074.03	1076.04	0.00409	2.1755	2.1796	34
35	0.099908	0.20342	0.01602	2947.8	2947.8	3.02	1073.46	1076.48	0.00612	2.1700	2.1761	35
36	0.10396	0.21166	0.01602	2838.7	2838.7	4.02	1072.90	1076.92	0.00815	2.1644	2.1726	36
37	0.10815	0.22020	0.01602	2734.1	2734.1	5.03	1072.33	1077.36	0.01018	2.1589	2.1691	37
38	0.11249	0.22904	0.01602	2633.8	2633.8	6.03	1071.77	1077.80	0.01220	2.1535	2.1657	38
39	0.11699	0.23819	0.01602	2537.6	2537.6	7.04	1071.20	1078.24	0.01422	2.1480	2.1622	39
40	0.12164	0.24767	0.01602	2445.4	2445.4	8.04	1070.64	1078.68	0.01623	2.1426	2.1588	40
41	0.12646	0.25748	0.01602	2356.9	2356.9	9.05	1070.06	1079.11	0.01824	2.1372	2.1554	41
42	0.13145	0.26763	0.01602	2272.0	2272.0	10.05	1069.50	1079.55	0.02024	2.1318	2.1520	42
43	0.13660	0.27813	0.01602	2190.5	2190.5	11.05	1068.94	1079.99	0.02224	2.1265	2.1487	43
44	0.14194	0.28899	0.01602	2112.3	2112.3	12.06	1068.37	1080.43	0.02423	2.1211	2.1453	44
45	0.14746	0.30023	0.01602	2037.3	2037.3	13.06	1067.81	1080.87	0.02622	2.1158	2.1420	45
46	0.15317	0.31185	0.01602	1965.2	1965.2	14.06	1067.24	1081.30	0.02820	2.1105	2.1387	46
47	0.15907	0.32387	0.01602	1896.0	1896.0	15.06	1066.68	1081.74	0.03018	2.1052	2.1354	47
48	0.16517	0.33629	0.01602	1829.5	1829.5	16.07	1066.11	1082.18	0.03216	2.0999	2.1321	48
49	0.17148	0.34913	0.01602	1765.7	1765.7	17.07	1065.55	1082.62	0.03413	2.0947	2.1288	49
50	0.17799	0.36240	0.01602	1704.3	1704.3	18.07	1064.99	1083.06	0.03610	2.0895	2.1256	50
51	0.18473	0.37611	0.01602	1645.4	1645.4	19.07	1064.42	1083.49	0.03806	2.0842	2.1223	51
52	0.19169	0.39028	0.01602	1588.7	1588.7	20.07	1063.86	1083.93	0.04002	2.0791	2.1191	52
53	0.19888	0.40492	0.01603	1534.3	1534.3	21.07	1063.30	1084.37	0.04197	2.0739	2.1159	53
54	0.20630	0.42003	0.01603	1481.9	1481.9	22.08	1062.72	1084.80	0.04392	2.0688	2.1127	54
55	0.21397	0.43564	0.01603	1431.5	1431.5	23.08	1062.16	1085.24	0.04587	2.0637	2.1096	55
56	0.22188	0.45176	0.01603	1383.1	1383.1	24.08	1061.60	1085.68	0.04781	2.0586	2.1064	56
57	0.23006	0.46840	0.01603	1336.5	1336.5	25.08	1061.04	1086.12	0.04975	2.0535	2.1033	57
58	0.23849	0.48558	0.01603	1291.7	1291.7	26.08	1060.47	1086.55	0.05168	2.0485	2.1002	58
59	0.24720	0.50330	0.01603	1248.6	1248.6	27.08	1059.91	1086.99	0.05361	2.0434	2.0970	59
60	0.25618	0.52160	0.01603	1207.1	1207.1	28.08	1059.34	1087.42	0.05553	2.0385	2.0940	60
61	0.26545	0.54047	0.01604	1167.2	1167.2	29.08	1058.78	1087.86	0.05746	2.0334	2.0909	61

[a]Compiled by John A. Goff and S. Gratch.
*Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
62	0.27502	0.55994	0.01604	1128.7	1128.7	30.08	1058.22	1088.30	0.05937	2.0284	2.0878	62
63	0.28488	0.58002	0.01604	1091.7	1091.7	31.08	1057.65	1088.73	0.06129	2.0235	2.0848	63
64	0.29505	0.60073	0.01604	1056.1	1056.1	32.08	1057.09	1089.17	0.06320	2.0186	2.0818	64
65	0.30554	0.62209	0.01604	1021.7	1021.7	33.08	1056.52	1089.60	0.06510	2.0136	2.0787	65
66	0.31636	0.64411	0.01604	988.63	988.65	34.07	1055.97	1090.04	0.06700	2.0087	2.0757	66
67	0.32750	0.66681	0.01605	956.76	956.78	35.07	1055.40	1090.47	0.06890	2.0039	2.0728	67
68	0.33900	0.69021	0.01605	926.06	926.08	36.07	1054.84	1090.91	0.07080	1.9990	2.0698	68
69	0.35084	0.71432	0.01605	896.47	896.49	37.07	1054.27	1091.34	0.07269	1.9941	2.0668	69
70	0.36304	0.73916	0.01605	867.95	867.97	38.07	1053.71	1091.78	0.07458	1.9893	2.0639	70
71	0.37561	0.76476	0.01605	840.45	840.47	39.07	1053.14	1092.21	0.07646	1.9845	2.0610	71
72	0.38856	0.79113	0.01606	813.95	813.97	40.07	1052.58	1092.65	0.07834	1.9797	2.0580	72
73	0.40190	0.81829	0.01606	788.38	788.40	41.07	1052.01	1093.08	0.08022	1.9749	2.0551	73
74	0.41564	0.84626	0.01606	763.73	763.75	42.06	1051.46	1093.52	0.08209	1.9701	2.0522	74
75	0.42979	0.87506	0.01606	739.95	739.97	43.06	1050.89	1093.95	0.08396	1.9654	2.0494	75
76	0.44435	0.90472	0.01606	717.01	717.03	44.06	1050.32	1094.38	0.08582	1.9607	2.0465	76
77	0.45935	0.93524	0.01607	694.88	694.90	45.06	1049.76	1094.82	0.08769	1.9560	2.0437	77
78	0.47478	0.96666	0.01607	673.52	673.54	46.06	1049.19	1095.25	0.08954	1.9513	2.0408	78
79	0.49066	0.99900	0.01607	652.91	652.93	47.06	1048.62	1095.68	0.09140	1.9466	2.0380	79
80	0.50701	1.0323	0.01607	633.01	633.03	48.05	1048.07	1096.12	0.09325	1.9419	2.0352	80
81	0.52382	1.0665	0.01608	613.80	613.82	49.05	1047.50	1096.55	0.09510	1.9373	2.0324	81
82	0.54112	1.1017	0.01608	595.25	595.27	50.05	1046.93	1096.98	0.09694	1.9328	2.0297	82
83	0.55892	1.1380	0.01608	577.34	577.36	51.05	1046.37	1097.42	0.09878	1.9281	2.0269	83
84	0.57722	1.1752	0.01608	560.04	560.06	52.05	1045.80	1097.85	0.10062	1.9236	2.0242	84
85	0.59604	1.2136	0.01609	543.33	543.35	53.05	1045.23	1098.28	0.10246	1.9189	2.0214	85
86	0.61540	1.2530	0.01609	527.19	527.21	54.04	1044.67	1098.71	0.10429	1.9144	2.0187	86
87	0.63530	1.2935	0.01609	511.60	511.62	55.04	1044.10	1099.14	0.10611	1.9099	2.0160	87
88	0.65575	1.3351	0.01610	496.52	496.54	56.04	1043.54	1099.58	0.10794	1.9054	2.0133	88
89	0.67678	1.3779	0.01610	481.96	481.98	57.04	1042.97	1100.01	0.10976	1.9008	2.0106	89
90	0.69838	1.4219	0.01610	467.88	467.90	58.04	1042.40	1100.44	0.11158	1.8963	2.0079	90
91	0.72059	1.4671	0.01610	454.26	454.28	59.03	1041.84	1100.87	0.11339	1.8919	2.0053	91

[a]Compiled by John A. Goff and S. Gratch.

Table 2. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
92	0.74340	1.5136	0.01611	441.10	441.12	60.03	1041.27	1101.30	0.11520	1.8874	2.0026	92
93	0.76684	1.5613	0.01611	428.38	428.40	61.03	1040.70	1101.73	0.11701	1.8830	2.0000	93
94	0.79091	1.6103	0.01611	416.07	416.09	62.03	1040.13	1102.16	0.11881	1.8786	1.9974	94
95	0.81584	1.6607	0.01612	404.17	404.19	63.03	1039.56	1102.59	0.12061	1.8741	1.9947	95
96	0.84103	1.7124	0.01612	392.65	392.67	64.02	1039.00	1103.02	0.12241	1.8698	1.9922	96
97	0.86711	1.7655	0.01612	381.51	381.53	65.02	1038.43	1103.45	0.12420	1.8654	1.9896	97
98	0.89388	1.8200	0.01612	370.73	370.75	66.02	1037.86	1103.88	0.12600	1.8610	1.9870	98
99	0.92137	1.8759	0.01613	360.30	360.32	67.02	1037.29	1104.31	0.12778	1.8566	1.9844	99
100	0.94959	1.9334	0.01613	350.20	350.22	68.02	1036.72	1104.74	0.12957	1.8523	1.9819	100
101	0.97854	1.9923	0.01614	340.42	340.44	69.01	1036.16	1105.17	0.13135	1.8480	1.9793	101
102	1.0083	2.0529	0.01614	330.96	330.98	70.01	1035.58	1105.59	0.13313	1.8437	1.9768	102
103	1.0388	2.1149	0.01614	321.80	321.82	71.01	1035.01	1106.02	0.13490	1.8394	1.9743	103
104	1.0700	2.1786	0.01614	312.93	312.95	72.01	1034.44	1106.45	0.13667	1.8351	1.9718	104
105	1.1021	2.2440	0.01615	304.34	304.36	73.01	1033.87	1106.88	0.13844	1.8309	1.9693	105
106	1.1351	2.3110	0.01615	296.02	296.04	74.01	1033.29	1107.30	0.14021	1.8266	1.9668	106
107	1.1688	2.3798	0.01616	287.96	287.98	75.00	1032.73	1107.73	0.14197	1.8224	1.9644	107
108	1.2035	2.4503	0.01616	280.14	280.16	76.00	1032.16	1108.16	0.14373	1.8182	1.9619	108
109	1.2390	2.5226	0.01616	272.58	272.60	77.00	1031.58	1108.58	0.14549	1.8140	1.9595	109
110	1.2754	2.5968	0.01617	265.24	265.26	78.00	1031.01	1109.01	0.14724	1.8098	1.9570	110
111	1.3128	2.6728	0.01617	258.14	258.16	79.00	1030.44	1109.44	0.14899	1.8056	1.9546	111
112	1.3510	2.7507	0.01617	251.25	251.27	80.00	1029.86	1109.86	0.15074	1.8015	1.9522	112
113	1.3902	2.8306	0.01618	244.57	244.59	80.99	1029.30	1110.29	0.15248	1.7973	1.9498	113
114	1.4305	2.9125	0.01618	238.10	238.12	81.99	1028.72	1110.71	0.15423	1.7932	1.9474	114
115	1.4717	2.9963	0.01618	231.82	231.84	82.99	1028.15	1111.14	0.15596	1.7890	1.9450	115
116	1.5139	3.0823	0.01619	225.73	225.75	83.99	1027.57	1111.56	0.15770	1.7849	1.9426	116
117	1.5571	3.1703	0.01619	219.83	219.85	84.99	1026.99	1111.98	0.15943	1.7809	1.9403	117
118	1.6014	3.2606	0.01620	214.10	214.12	85.99	1026.42	1112.41	0.16116	1.7767	1.9379	118
119	1.6468	3.3530	0.01620	208.54	208.56	86.98	1025.85	1112.83	0.16289	1.7727	1.9356	119
120	1.6933	3.4477	0.01620	203.16	203.18	87.98	1025.28	1113.26	0.16461	1.7687	1.9333	120
121	1.7409	3.5446	0.01621	197.93	197.95	88.98	1024.70	1113.68	0.16634	1.7647	1.9310	121

[a]Compiled by John A. Goff and S Gratch

TABLE 2 THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	Absolute Pressure p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Specific Volume Evap. v_{fg}	Specific Volume Sat Vapor v_g	Enthalpy Btu per lb Sat. Liquid h_f	Enthalpy Evap. h_{fg}	Enthalpy Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Liquid s_f	Entropy Evap. s_{fg}	Entropy Sat. Vapor s_g	Fahr. Temp. t(F)
122	1.7897	3.6439	0.01621	192.85	192.87	89.98	1024.12	1114.10	0.16805	1.7606	1.9286	122
123	1.8396	3.7455	0.01622	187.93	187.95	90.98	1023.54	1114.52	0.16977	1.7566	1.9264	123
124	1.8907	3.8496	0.01622	183.15	183.17	91.98	1022.96	1114.94	0.17148	1.7526	1.9241	124
125	1.9430	3.9561	0 01622	178.51	178.53	92.98	1022.39	1115.37	0.17319	1.7486	1.9218	125
126	1.9066	4.0651	0.01623	174.00	174 02	93.98	1021.81	1115.79	0.17490	1.7446	1.9195	126
127	2.0514	4.1768	0.01623	169.63	169.65	94.97	1021.24	1116.21	0.17660	1.7407	1.9173	127
128	2.1075	4.2910	0 01624	165.38	165.40	95.97	1020.66	1116.63	0.17830	1.7367	1.9150	128
129	2.1649	4.4078	0.01624	161.26	161.28	96.97	1020.08	1117.05	0.18000	1.7328	1.9128	129
130	2.2237	4.5274	0.01625	157.25	157.27	97.97	1019.50	1117.47	0.18170	1.7289	1.9106	130
131	2.2838	4.6498	0.01625	153.36	153.38	98.97	1018.92	1117.89	0.18339	1.7250	1.9084	131
132	2.3452	4.7750	0.01626	149.58	149.60	99.97	1018.34	1118.31	0.18508	1.7211	1.9062	132
133	2.4081	4.9030	0.01626	145.91	145.93	100.97	1017.76	1118.73	0.18676	1.7172	1.9040	133
134	2.4725	5.0340	0.01626	142.34	142.36	101.97	1017.18	1119.15	0.18845	1.7134	1.9018	134
135	2.5382	5.1679	0.01627	138.87	138.89	102.97	1016.59	1119.56	0.19013	1.7095	1.8996	135
136	2.6055	5.3049	0.01627	135.50	135.52	103.97	1016.01	1119.98	0.19181	1.7056	1.8974	136
137	2.6743	5.4450	0.01628	132.22	132.24	104.97	1015.43	1120.40	0.19348	1.7018	1.8953	137
138	2.7446	5.5881	0.01628	129.04	129.06	105.97	1014.85	1120.82	0.19516	1.6979	1.8931	138
139	2.8165	5.7345	0 01629	125.94	125.96	106.97	1014.26	1121.23	0.19683	1.6942	1.8910	139
140	2.8900	5.8842	0.01629	122.94	122.96	107.96	1013.69	1121.65	0.19850	1.6903	1.8888	140
141	2.9651	6.0371	0.01630	120.01	120.03	108.96	1013.11	1122.07	0.20016	1.6865	1.8867	141
142	3.0419	6.1934	0.01630	117.16	117.18	109.96	1012.52	1122.48	0.20182	1.6828	1.8846	142
143	3.1204	6.3532	0.01631	114.40	114.42	110.96	1011.94	1122.90	0.20348	1.6790	1.8825	143
144	3.2006	6.5164	0.01631	111.70	111.72	111.96	1011.35	1123.31	0.20514	1.6753	1.8804	144
145	3.2825	6.6832	0.01632	109.09	109.11	112.96	1010.77	1123.73	0.20679	1.6715	1.8783	145
146	3.3662	6.8536	0.01632	106.54	106.56	113.96	1010.18	1124.14	0.20845	1.6678	1.8763	146
147	3.4517	7.0277	0.01633	104.06	104.08	114.96	1009.59	1124.55	0.21010	1.6641	1.8742	147
148	3.5390	7.2056	0.01633	101 65	101.67	115.96	1009.01	1124.97	0.21174	1.6604	1.8721	148
149	3.6282	7.3872	0.01634	99.306	99.322	116.96	1008.42	1125.38	0.21339	1.6567	1.8701	149
150	3.7194	7.5727	0 01634	97.022	97.038	117.96	1007.83	1125.79	0.21503	1.6530	1.8680	150
151	3.8124	7.7622	0.01635	94.799	94 815	118 96	1007.24	1126.20	0.21667	1.6493	1.8660	151

[a] Compiled by John A. Goff and S. Gratch.

Table 2. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap v_{fg}	Sat Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap s_{fg}	Sat. Vapor s_g	
152	3.9074	7.9556	0.01635	92.635	92.651	119.96	1006.66	1126.62	0.21830	1.6457	1.8640	152
153	4.0044	8.1532	0.01636	90.528	90.544	120.97	1006.06	1127.03	0.21994	1.6421	1.8620	153
154	4.1035	8.3548	0.01636	88.477	88.493	121.97	1005.47	1127.44	0.22157	1.6384	1.8600	154
155	4.2046	8.5607	0.01637	86.480	86.496	122.97	1004.88	1127.85	0.22320	1.6348	1.8580	155
156	4.3078	8.7708	0.01637	84.536	84.552	123.97	1004.29	1128.26	0.22482	1.6312	1.8560	156
157	4.4132	8.9853	0.01638	82.642	82.658	124.97	1003.70	1128.67	0.22645	1.6276	1.8540	157
158	4.5207	9.2042	0.01638	80.798	80.814	125.97	1003.11	1129.08	0.22807	1.6239	1.8520	158
159	4.6304	9.4276	0.01639	79.001	79.017	126 97	1002.51	1129.48	0.22969	1.6204	1.8501	159
160	4.7424	9.6556	0.01639	77.251	77.267	127.97	1001.92	1129.89	0.23130	1.6168	1.8481	160
161	4.8566	9.8882	0.01640	75.546	75.562	128.97	1001.33	1130.30	0.23292	1.6133	1.8462	161
162	4.9732	10.126	0.01640	73.885	73.901	129.97	1000.74	1130.71	0.23453	1.6097	1.8442	162
163	5.0921	10.368	0.01641	72.267	72.283	130.98	1000.13	1131.11	0.23614	1.6062	1.8423	163
164	5.2134	10.615	0.01642	70.690	70.706	131.98	999.54	1131.52	0.23774	1.6027	1.8404	164
165	5.3372	10.867	0.01642	69.153	69.169	132.98	998.94	1131.92	0.23935	1.5990	1.8384	165
166	5.4634	11.124	0.01643	67.654	67.670	133.98	998.35	1132.33	0.24095	1.5956	1.8365	166
167	5.5921	11.386	0.01643	66.194	66.210	134.98	997.75	1132.73	0.24255	1.5920	1.8346	167
168	5.7233	11.653	0.01644	64.770	64.786	135.98	997.16	1133.14	0.24414	1.5887	1.8328	168
169	5.8572	11.925	0.01644	63.382	63.398	136.90	996.55	1133.54	0.24574	1.5852	1.8309	169
170	5.9936	12.203	0.01645	62.029	62.045	137.99	995.95	1133.94	0.24733	1.5817	1.8290	170
171	6.1328	12.487	0.01645	60.710	60.726	138.99	995.36	1134.35	0.24892	1.5782	1.8271	171
172	6.2746	12.775	0.01646	59.423	59.439	139.99	994.76	1134.75	0.25051	1.5748	1.8253	172
173	6.4192	13.070	0.01647	58.168	58.184	141.00	994.15	1135.15	0.25209	1.5713	1.8234	173
174	6.5666	13.370	0.01647	56.944	56.960	142.00	993.55	1135.55	0.25367	1.5679	1.8216	174
175	6.7168	13.676	0.01648	55.750	55.766	143.00	992.95	1135.95	0.25525	1.5644	1.8197	175
176	6.8699	13.987	0.01648	54.586	54.602	144.00	992.35	1136.35	0.25683	1.5611	1.8179	176
177	7.0259	14.305	0.01649	53.450	53.466	145.00	991.75	1136.75	0.25841	1.5577	1.8161	177
178	7.1849	14.629	0.01650	52.341	52.357	146.01	991.14	1137.15	0.25998	1.5543	1.8143	178
179	7.3469	14.959	0.01650	51.260	51.276	147.01	990.54	1137.55	0.26155	1.5508	1.8124	179
180	7.5119	15.295	0.01651	50.203	50.220	148.01	989.93	1137.94	0.26312	1.5475	1.8106	180
181	7.6801	15.637	0.01651	49.173	49.190	149.02	989.32	1138.34	0.26468	1.5442	1.8089	181

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Concluded)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
182	7.8514	15.986	0.01652	48.168	48.185	150.02	988.72	1138.74	0.26625	1.5408	1.8071	182
183	8.0258	16.341	0.01652	47.187	47.204	151.02	988.12	1139.14	0.26781	1.5375	1.8053	183
184	8.2035	16.703	0.01653	46.229	46.246	152.03	987.50	1139.53	0.26937	1.5341	1.8035	184
185	8.3845	17.071	0.01654	45.294	45.311	153.03	986.89	1139.92	0.27093	1.5308	1.8017	185
186	8.5688	17.446	0.01654	44.381	44.398	154.04	986.28	1140.32	0.27248	1.5275	1.8000	186
187	8.7565	17.829	0.01655	43.489	43.506	155.04	985.67	1140.71	0.27404	1.5242	1.7982	187
188	8.9476	18.218	0.01656	42.619	42.636	156.04	985.07	1141.11	0.27559	1.5209	1.7965	188
189	9 1422	18.614	0.01656	41.769	41.786	157.05	984.45	1141.50	0.27713	1.5176	1.7947	189
190	9.3403	19.017	0.01657	40.939	40.956	158.05	983.84	1141.89	0.27868	1.5143	1.7930	190
191	9.5420	19.428	0.01658	40.128	40.145	159.06	983.22	1142.28	0.28022	1.5111	1.7913	191
192	9.7473	19.846	0.01658	39.337	39.354	160.06	982.61	1142.67	0.28176	1.5078	1.7896	192
193	9.9563	20.271	0.01659	38.563	38.580	161.06	982.00	1143.06	0.28330	1.5045	1.7878	193
194	10.169	20.704	0.01659	37.807	37.824	162.07	981.38	1143.45	0.28484	1.5013	1.7861	194
195	10.386	21.145	0.01660	37.069	37.086	163.08	980.76	1143.84	0.28638	1.4980	1.7844	195
196	10.606	21.594	0.01661	36.348	36.365	164.08	980.15	1144.23	0.28791	1.4949	1.7828	196
197	10.830	22.050	0.01661	35.643	35.660	165.08	979.54	1144.62	0.28944	1.4917	1.7811	197
198	11.058	22.515	0.01662	34.954	34.971	166.09	978.91	1145.00	0.29097	1.4884	1.7794	198
199	11.290	22.987	0.01663	34.281	34.298	167.10	978.29	1145.39	0.29250	1.4852	1.7777	199
200	11.526	23.468	0.01663	33.623	33.640	168.10	977.68	1145.78	0.29402	1.4820	1.7760	200
201	11.767	23.957	0.01664	32.980	32.997	169.11	977.05	1146.16	0.29554	1.4789	1.7744	201
202	12.011	24.455	0.01665	32.351	32.368	170.11	976.43	1146.54	0.29706	1.4756	1.7727	202
203	12.260	24.961	0.01665	31.737	31.754	171.12	975.81	1146.93	0.29858	1.4725	1.7711	203
204	12.513	25.476	0.01666	31.136	31.153	172.12	975.19	1147.31	0.30010	1.4693	1.7694	204
205	12.770	26.000	0.01667	30.549	30.566	173.13	974.56	1147.69	0.30161	1.4662	1.7678	205
206	13.031	26.532	0.01667	29.974	29.991	174.14	973.94	1148.08	0.30312	1.4631	1.7662	206
207	13.297	27.074	0.01668	29.413	29.430	175.14	973.32	1148.46	0.30463	1.4600	1.7646	207
208	13.568	27.625	0.01669	28.863	28.880	176.15	972.69	1148.84	0.30614	1.4568	1.7629	208
209	13.843	28.185	0.01669	28.326	28.343	177.16	972.06	1149.22	0.30765	1.4536	1.7613	209
210	14.123	28.754	0.01670	27.801	27.818	178.17	971.43	1149.60	0.30915	1.4506	1.7597	210
211	14.407	29.333	0.01671	27.287	27.304	179.17	970.81	1149.98	0.31065	1.4474	1.7581	211
212	14.696	29.921	0.01671	26.784	26.801	180.18	970.17	1150.35	0.31215	1.4444	1.7565	212

[a]Compiled by John A. Goff and S. Gratch.

TABLE 3. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

Abs. Press. In. Hg p	Temp F t	Specific Volume		Enthalpy			Entropy			Abs. Press. In. Hg p
		Sat. Liquid v_f	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
0.25	40.23	0.01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589	0.25
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	0.50
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635	0.75
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2.0387	1.00
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041	1.5
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797	2
4	125.43	0.01622	176.7	93.34	1022.7	1116.0	0.1738	1.7476	1.9214	4
6	140.78	0.01630	120.72	108.67	1013.6	1122.3	0.1996	1.6881	1.8877	6
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0.2186	1.6454	1.8640	8
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456	10
12	169.28	0.01644	63.03	137.18	996.7	1133.9	0.2460	1.5847	1.8307	12
14	176.05	0.01648	54.55	143.96	992.6	1136.6	0.2568	1.5613	1.8181	14
16	182.05	0.01652	48.14	149.98	988.9	1138.9	0.2662	1.5410	1.8072	16
18	187.45	0.01655	43.11	155.39	985.7	1141.1	0.2746	1.5231	1.7977	18
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	20
22	196.90	0.01661	35.73	164.87	979.8	1144.7	0.2891	1.4923	1.7814	22
24	201.09	0.01664	32.94	169.09	977.2	1146.3	0.2955	1.4789	1.7744	24
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0.3014	1.4665	1.7679	26
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619	28
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564	30
Lb/Sq In.										Lb/Sq In.
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	14.696
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	16
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	18
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	20
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242	22
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172	24
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	26
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	28
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	30
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941	32
34	257.58	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893	34
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848	36
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805	38
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	40
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724	42
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687	44
46	275.80	0.01722	9.209	244.75	927.7	1172.4	0.4038	1.2613	1.6652	46
48	278.45	0.01725	8.848	247.47	925.8	1173.3	0.4075	1.2542	1.6617	48
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	50
52	283.49	0.01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	52
54	285.90	0.01731	7.922	255.09	920.5	1175.6	0.4177	1.2346	1.6523	54
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1.6494	56
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466	58
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	60
62	294.85	0.01740	6.957	264.30	913.9	1178.2	0.4300	1.2112	1.6412	62
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	64
66	298.99	0.01744	6.560	268.55	910.8	1179.4	0.4356	1.2006	1.6362	66
68	300.98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338	68
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	70
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292	72
74	306.68	0.01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270	74
76	308.50	0.01754	5.743	278.37	903.7	1182.1	0.4484	1.1764	1.6248	76
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	78
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	80
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187	82
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168	84
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149	86
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130	88
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	90
92	321.83	0.01768	4.796	292.18	893.5	1185.7	0.4661	1.1433	1.6094	92
94	323.36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076	94
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060	96
98	326.35	0.01772	4.517	296.89	889.9	1186.8	0.4721	1.1322	1.6043	98
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	100
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	150
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	200
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	300
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	400
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	500

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes published by John Wiley and Sons, Inc., 1936 edition.

p_s = saturation pressure of pure water vapor, pounds per square inch or inches of Hg. Moist air can be saturated at any given values of temperature and pressure, though this requires that it have a definite humidity ratio W_s and that the coexisting condensed phase contain a definite, but very small quantity of dissolved air. On the other hand, pure water vapor (steam) cannot be saturated at any given values of temperature and pressure because its composition is invariable. It can, however, be saturated at any given temperature (below the critical temperature), though this requires that it have a definite pressure p_s and that the coexisting condensed phase have the same temperature and pressure. The values of saturation pressure listed in Table 1 have been computed from the formulas of Goff and Gratch[2].

Thermodynamic Properties of Water at Saturation

Table 2 offers revisions to existing steam table data[3] with extension downward to −160 F. These revisions and extension were a necessary preliminary to the construction of Table 1. A detailed explanation of the methods employed in constructing Table 2 is given in a paper[2] by John A. Goff and S. Gratch. As in Table 1 the temperature scale used as argument in Table 2 is the Fahrenheit scale defined in terms of absolute temperature T by Equation 1, whereas the Fahrenheit scale used as argument in existing steam tables is that derived from the International Centigrade scale by means of Equation 2. The symbols used as column headings in Table 2 are the same as those used in the steam tables and have the same meanings; therefore, a detailed explanation seems unnecessary.

Properties of water above 212 F from Keenan and Keyes[3] are given in Table 3.

DEGREE OF SATURATION

At given values of temperature and pressure the humidity ratio W of moist air can have any value between zero (dry air) and W_s (moist air at saturation). For convenience a parameter μ called alternatively *degree of saturation* is introduced through the definition,

$$W = \mu W_s \tag{3}$$

Obviously the degree of saturation μ can have any value from zero (dry air) to unity (moist air at saturation).

To a degree of approximation within the estimated uncertainty of the

TABLE 4. COEFFICIENTS A, B, C APPEARING IN EQUATIONS 4a, 5a, 6a, MAXIMUM VALUES OF CORRECTIONS DEFINED BY EQUATIONS 4a, 5a, 6a. DEGREE OF SATURATION AT WHICH THESE MAXIMA OCCUR, $\bar{\mu}_m$. MAXIMUM VALUE OF CORRECTION DEFINED BY EQUATION 6b. DEGREE OF SATURATION AT WHICH THIS MAXIMUM OCCURS, $\bar{\bar{\mu}}_m$.
(Standard Atmospheric Pressure)

t (F)	A (ft^3/lb$_a$)	B (Btu/lb$_a$)	C (Btu/F/lb$_a$)	$\bar{v}_{max}$ (ft^3/lb$_a$)	$\bar{h}_{max}$ (Btu/lb$_a$)	$\bar{s}_{max}$ (Btu/F/lb$_a$)	$\bar{\mu}_m$	$\bar{\bar{s}}_{max}$ (Btu/F/lb$_a$)	$\bar{\bar{\mu}}_m$
96	0.0018	0.0286	0.00004	0.0004	0.0069	0.00001	0.4925	0.0015	0.3650
112	0.0042	0.0650	0.00009	0.0010	0.0155	0.00002	0.4878	0.0025	0.3632
128	0.0096	0.1439	0.00020	0.0022	0.0332	0.00005	0.4805	0.0040	0.3602
144	0.0215	0.3149	0.00042	0.0047	0.0693	0.00009	0.4691	0.0065	0.3557
160	0.0487	0.6969	0.00091	0.0099	0.1418	0.00019	0.4511	0.0106	0.3485
176	0.1169	1.636	0.00207	0.0207	0.2903	0.00037	0.4213	0.0179	0.3363
192	0.3363	4.608	0.00567	0.0451	0.6180	0.00076	0.3662	0.0333	0.3129

data in Table 1 at temperatures below about 150 F, the volume v of moist air per pound of dry air at any degree of saturation μ may be computed from the simple relation,

$$v = v_a + \mu v_{as} \tag{4}$$

To obtain comparable accuracy at temperatures above about 150 F it is necessary to add a correction term $\bar{v}$ as follows,

$$\bar{v} = \frac{\mu(1 - \mu)A}{1 + aW_s\mu} \tag{4a}$$

where a denotes the ratio of the apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016), namely, 1.6078. In Table 4 are given, for each of several higher temperatures, the corresponding value of the coefficient A, the value of μ at which the correction term $\bar{v}$ attains its maximum value, and the maximum value of the correction term there attained.

At temperatures below about 150 F the enthalpy h of moist air per pound of dry air at any degree of saturation μ may be computed from the simple relation,

$$h = h_a + \mu h_{as} \tag{5}$$

To obtain comparable accuracy at temperatures above about 150 F it is necessary to add a correction term $\bar{h}$ as follows,

$$\bar{h} = \frac{\mu(1 - \mu)B}{1 + aW_s\mu} \tag{5a}$$

In Table 4 are listed values of the coefficient B and maximum values of the correction term $\bar{h}$, the latter occurring at the same values of μ as do those of the correction term $\bar{v}$.

Unfortunately, values of the entropy s of moist air per pound of dry air computed from the simple relation,

$$s = s_a + \mu s_{as} \tag{6}$$

do not approximate the true values as closely as might be desired except at temperatures considerably lower than 150 F. Only a relatively small portion of the error is contributed by the correction term

$$\bar{s} = \frac{\mu(1 - \mu)C}{1 + aW_s\mu} \tag{6a}$$

Table 4 lists values of the coefficient C and maximum values of the correction terms $\bar{s}$, the latter occurring at the same values of μ as do those of the correction terms $\bar{v}$ and $\bar{h}$. The larger portion of the error is contributed by the so-called *mixing entropy*. It can be expressed as an additional correction term $\bar{\bar{s}}$ as follows,

$$\bar{\bar{s}} = 0.1579\,[(1 + \mu aW_s)\log_{10}(1 + \mu aW_s - \mu aW_s \log_{10}(\mu) \\ - \mu(1 + aW_s)\log_{10}(1 + aW_s] \tag{6b}$$

Maximum values of $\bar{\bar{s}}$ and the values of μ at which they occur are given in Table 4. In Equation 6b $\log_{10}$ denotes logarithm to the base 10.

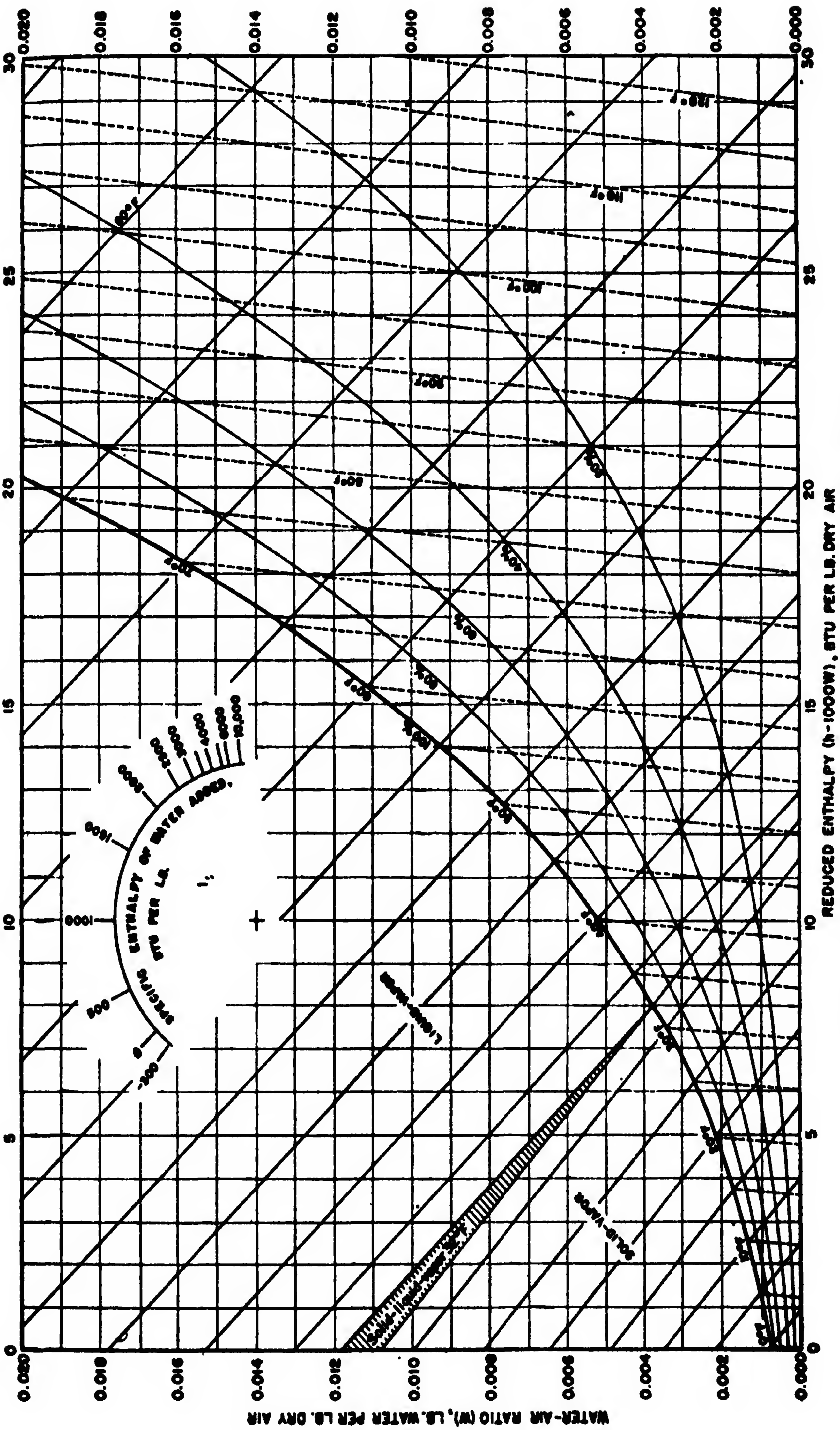

FIG. 1. ABRIDGMENT OF GOFF DIAGRAM FOR MOIST AIR

By John A. Goff, University of Pennsylvania, Thermodynamics Research Laboratory. Standard atmospheric pressure (29.921 in. Hg).

GOFF DIAGRAM FOR MOIST AIR

It is a fundamental proposition of thermodynamics that *when a fluid flows across a section fixed in space it convects with it an amount of energy equal to its enthalpy as determined by the pressure, temperature, and composition of the fluid at that section.* This fundamental proposition provides the correct procedure for applying the law of conservation of energy to the processes occurring most frequently in air conditioning practice. Thus if moist air is flowing through a duct it carries across any section of the duct energy of amount mh Btu per minute and water of amount mW pounds per minute, if m denotes the weight of dry air crossing the section per minute.

The foregoing considerations suggest the importance of having accurate knowledge regarding the enthalpy of the fluid in question and the desirability of using enthalpy as one of the coordinates in graphical representation. The use of enthalpy h and humidity ratio W as coordinates in the case of moist air is due to Mollier[4]. A convenient modification of the Mollier diagram introduced by Goff and designated Goff Diagram for Moist Air is obtained by taking humidity ratio W as ordinate and reduced enthalpy (h-1000W) as abscissa. A Goff Diagram modified in this way is enclosed in the envelope attached to the inside back cover and an abridgement of the Diagram is shown in Fig. 1.

The Goff Diagram is a constant-pressure chart, the one provided with this book being drawn for standard atmospheric pressure from the data in Table 1. Along the axis of abscissae ($W = 0$, $\mu = 0$) are plotted values of the specific enthalpy of dry air h_a at one-degree intervals of temperature. Values of humidity ratio at saturation W_s plotted against values of reduced enthalpy at saturation (h_s-1000W_s) determine the saturation curve (μ = 100 per cent). Lines of constant temperature connect points on the saturation curve with corresponding points on the dry-air axis and are inclined upward to the right. They are drawn straight in accordance with Equations 3 and 5 because the curvature contributed by the correction term 5a is inappreciable at all temperatures within the range of the chart. The portion of each isotherm lying between the dry-air axis and the saturation curve is divided into 10 equal parts by curves of constant per cent saturation. The per cent saturation of any point below the saturation curve is readily determined by linear interpolation along the isotherm through that point.

Each isotherm breaks at the saturation curve to incline upward to the left into the two-phase region above the saturation curve. The ordinate of a point in this region is the total weight of water in both the vapor phase (moist air) and the condensed phase (liquid or solid) per pound of dry air in both phases. Neglecting the very small amount of dissolved air in the condensed phase, it is the weight of water in both phases per pound of dry air in the vapor phase. The ordinate at the break in the isotherm through the point in question is the weight of water per pound of dry air in the vapor phase. Consequently, the difference between the two ordinates is the weight of condensed phase per pound of dry air in the vapor phase.

It has been stated that the region above the saturation curve is the two-phase region. This is so except in the wedge with apex on the saturation curve at 32 F where three distinct phases, namely, solid, liquid, and vapor coexist. In fact, this wedge separates the liquid-vapor region above the wedge from the solid-vapor region below it. A point inside the wedge divides the horizontal line extending through it from one boundary

to the other into two segments which are in the same ratio as are the weights of solid and liquid. The temperature is 32 F throughout the wedge.

The isotherms in the two-phase regions above the saturation curve have been extended downward to the right into the vapor-phase region below the saturation curve as lines of constant *thermodynamic wet-bulb temperature*. The definition of thermodynamic wet-bulb temperature will be given later.

On the Goff Diagram provided with THE 1949 GUIDE there has been drawn a *protractor* from which can be determined the direction in which the state point of a mixture of water and dry air will be moved *by simultaneous addition of energy and water without addition of dry air*. A particular direction is specified by the numerical value of the *ratio of energy to water added* which ratio is designated as q and called the *specific enthalpy of water added, Btu per pound*. The protractor is useful in locating the *condition line* of a cooling load or heating load problem.

DERIVED PROPERTIES

Thermodynamic Wet-bulb Temperature. For any state of moist air there exists a temperature t^* at which liquid (or solid) water may be evaporated into the air to bring it to saturation at exactly this same temperature. The humidity ratio of the air is increased from a given initial value W to the value W_s^* corresponding to saturation at the temperature t^*; the enthalpy of the air is increased from a given initial value h to the value h_s^* corresponding to saturation at the temperature t^*; the weight of water added per pound of dry air is $W_s^* - W$ and this adds energy of amount $(W_s^* - W)\,h_w^*$, where h_w^* denotes the specific enthalpy of the water as added at the temperature t^*; therefore, if the process is strictly adiabatic,

$$h + (W_s^* - W)h_w^* = h_s^* \tag{7}$$

The solution of Equation 7 for given values of h and W is called *thermodynamic wet-bulb temperature*.

Example 1. Find the thermodynamic wet-bulb temperature of moist air at 80 F, 50 per cent saturation, atmospheric pressure.

Solution. From the data of Table 1, the enthalpy of the air is $h = 19.221 + 0.50 \times 24.47 = 31.46$ Btu/lb (Equation 5). To a first approximation this is the enthalpy at saturation at the thermodynamic wet-bulb temperature which is therefore approximately 67 F.

At 67 F the humidity ratio at saturation is 0.01424 lb_w/lb_a and the specific enthalpy of liquid water is 35.11 Btu/lb_w. The humidity ratio of the air is $W = 0.50 \times 0.02233 = 0.01117$ lb_w/lb_a (Equation 3). Therefore, to a second approximation, the enthalpy at saturation at the thermodynamic wet-bulb temperature is $h_s^* = 31.46 + (0.01424 - 0.01117) \times 35.11 = 31.57$ Btu/lb_a, Equation 7. Interpolation in Table 1 gives as final answer,

$$t^* = 66.94 \text{ F}$$

The answer can also be read directly on the Goff Diagram at the intersection of the 80 F dry-bulb and 50 per cent saturation lines.

The psychrometer is an instrument consisting of two thermometers one of which has the bulb covered with a suitable wick that has been dipped in liquid water and thoroughly wetted by it. On placing the wet-bulb of the instrument in an air stream, the liquid begins to evaporate from the wick and it is usually assumed that such evaporation brings the air immediately adjacent to the wick to saturation. At first this air may reach saturation at a higher or lower temperature than that of the liquid on the wick; but in a relatively short time the temperature of the liquid will have

changed to approach equality with that of the air touching the wick, even if this requires the liquid to freeze on the wick. Then the liquid (or solid) will continue for a time to evaporate into the air stream at such temperature as will bring a portion of the air stream to saturation at this same temperature. This equilibrium temperature is called *wet-bulb temperature.*

It is clear that the readings of an actual wet-bulb thermometer cannot be regarded as values of a thermodynamic property of moist air; for these readings are importantly affected by a number of non-thermodynamic factors including design, construction, installation, and technique of using the instrument. Thus, unless the wet-bulb is effectively shielded against radiation from relatively warm surfaces the process will not be strictly adiabatic as tacitly assumed in writing Equation 7. Also, partial drying of the wick will prevent the air immediately adjacent to it from reaching complete saturation as assumed in Equation 7. A working theory developed by Arnold[5] enables the calculation of corrections to be applied to the readings of the actual instrument in order to make them agree with the values of temperature calculated from Equation 7. Fortunately, and indeed fortuitously, these corrections can be made small, but to emphasize the necessity of making them in accurate experimentation, the temperature defined by Equation 7 is called *thermodynamic wet-bulb temperature.*

Example 2. Find the degree of saturation of moist air at 90 F dry-bulb, 63 F thermodynamic wet-bulb, atmospheric pressure.

Solution. Inserting numerical data from Table 1 into Equation 7 gives

$$(21.625 + 34.31\mu) + (0.01235 - 0.03118\mu) \times 31.12 = 28.57$$

The solution of this equation is direct and the final answer is

$$\mu = 19.67 \text{ per cent}$$

The per cent saturation may also be read directly at intersection of 90 F dry-bulb and 63 F thermodynamic wet-bulb lines on the Goff Diagram.

Example 3. Find the temperature to which moist air initially saturated at 40 F and at standard atmospheric pressure must be heated in order to have a thermodynamic wet-bulb temperature of 60 F.

Solution. On the Goff Diagram follow a horizontal line from the saturation curve at 40 F to its intersection with the 60 F thermodynamic wet-bulb line and read the corresponding temperature directly.

Inserting numerical data from Table 1 into Equation 7, this becomes

$$h_a + 0.005213h_{as}/W_s = 26.46 - (0.01108 - 0.005213) \times 28.12 = 26.295$$

At 85 F the lefthand member of this equation has the value 26.147; at 86 F its value is 26.389; by linear interpolation the answer is: $t = 85.61$ F.

Dew-Point Temperature. Corresponding to any given state of moist air there exists another state on the saturation curve having the same humidity ratio W and same pressure p as the given state. The temperature at this other state on the saturation curve is called the *dew-point temperature* of the given state. Obviously, if moist air is cooled at constant pressure and constant humidity ratio it will reach saturation when its temperature falls to a value equal to its dew-point temperature. This will usually be marked by the first appearance of a coexisting condensed phase. In one type of dew-point apparatus a continuous sample of air is passed over a mirror which can be cooled by external refrigeration and whose temperature can be accurately measured. The measured temperature at which the intensity of light reflected from the mirror is abruptly diminished by condensation is taken to be the dew-point temperature

of the air sample. Examples 4 and 5 illustrate the relation between the dew-point, degree of saturation and dry-bulb temperature.

Example 4. Find the dew-point temperature of moist air at 80 F, 50 per cent saturation, atmospheric pressure.

Solution. On the Goff Diagram follow a horizontal line from a given state point (80 F, 50 per cent) to the saturation curve and read the temperature at the intersection.

To solve from Table 1:

From the data in Table 1, the humidity ratio of the air is $W = 0.50 \times 0.02233 = 0.01117$ lb_w/lb_a. By interpolation this is found to be the humidity ratio at saturation at 60.22 F which is therefore the required answer.

Example 5. Find the degree of saturation of moist air at 90 F dry-bulb, 40 F (dew-point), atmospheric pressure.

Solution. On the Goff Diagram follow a horizontal line from 40 F on the saturation curve to the 90 F isotherm (dry-bulb) and read the degree of saturation directly.

To solve from Table 1:

From the data in Table 1, the humidity ratio of the air must be $W = 0.005213$. But the humidity ratio at saturation at 90 F is 0.03118; hence the degree of saturation is

$$\mu = 0.005213/0.03118 = 16.72 \text{ per cent}$$

TYPICAL AIR CONDITIONING PROCESSES

The use of Table 1 and the Goff Diagram in analyzing typical air conditioning processes is best explained by means of illustrative examples. In each of the following, it is to be understood that the process in question takes place at a constant pressure of 29.921 in. Hg, or standard atmospheric pressure.

Heating

The process of adding heat to moist air is represented by a horizontal line on the Goff Diagram. The length of the line between the initial and final state points is the increase of reduced enthalpy; but, since the humidity ratio is constant, it is also the increase of enthalpy itself and therefore the quantity of heat added per pound of dry air.

Example 6. Air initially at 20 F, 80 per cent saturation is heated to 120 F. Find the quantity of heat required to process 20,000 cfm of heated air.

Solution. From the data in Table 1: the initial humidity ratio is $0.80 \times 0.002152 = 0.001722$ lb_w/lb_a; the initial enthalpy is $4.804 + 0.80 \times 2.302 = 6.646$ Btu/lb_a; the final degree of saturation is $0.001722/0.08149 = 2.113$ per cent; the final enthalpy is $28.841 + 0.02113 \times 90.70 = 30.757$ Btu/lb_a.

It may be supposed that the air is heated between two sections of a duct. The quantities of energy convected across the two sections per pound of dry air crossing them are the two enthalpies calculated. Conservation of energy requires that the difference between these two enthalpies be the quantity of heat added; thus,

$${}_{A}q_{B} = 30.757 - 6.646 = 24.111 \text{ Btu/lb}_a.$$

The final volume is $14.611 + 0.02113 \times 1.905 = 14.651$ cu ft/lb_a. Since 20,000 cfm of heated air is to be processed, the total quantity of heat required is

$${}_{A}Q_{B} = 24.111 \times 20{,}000/14.651 = 32{,}914 \text{ Btu per minute.}$$

On the Goff Diagram the process is represented by the horizontal line AB, Fig. 2, whose length is the quantity of heat added per pound of dry air. The reduced enthalpy at A is 4.92 while that at B is 29.03, both being read directly from the chart. Since humidity ratio is constant the difference between these reduced enthalpies is also the difference between the enthalpies themselves, namely, 24.11 Btu/lb_a.

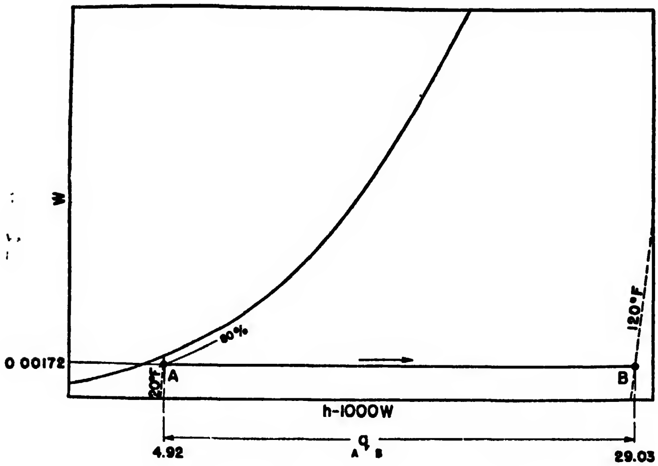

FIG. 2. ILLUSTRATION OF USE OF GOFF DIAGRAM IN SOLUTION OF EXAMPLE 6

Cooling

The process of cooling moist air is also represented by a horizontal line on the Goff Diagram. The line may extend across the saturation curve into the two-phase region, nevertheless, the length of the line between the initial and final states is the quantity of heat removed, or refrigeration supplied, per pound of dry air. By following the final isotherm downward to the right to the saturation curve and reading the ordinate there, the weight of water vapor per pound of dry air in the vapor phase is determined. The difference between the initial humidity ratio and this ordinate is the weight of condensed phase per pound of dry air in the final state.

Example 7. Air at 95 F and 50 per cent saturation is cooled to 70 F. Find the refrigeration required to process 20,000 cfm of uncooled air.

Solution. From the data in Table 1: the initial humidity ratio is $0.50 \times 0.03673 = 0.01837$ lb_w/lb_a; the initial enthalpy is $22.827 + 0.50 \times 40.49 = 43.072$ Btu/lb_a; the humidity ratio at saturation at the final temperature is 0.01582 lb_w/lb_a; the quantity of liquid formed is $0.01837 - 0.01582 = 0.00255$ lb_w/lb_a; the enthalpy of the final two-phase mixture is $34.09 + 0.00255 \times 38.11 = 34.187$ Btu/lb_a.

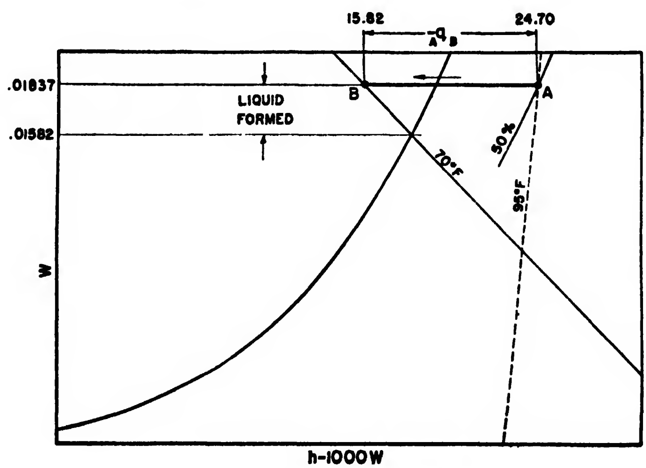

FIG. 3. ILLUSTRATION OF USE OF GOFF DIAGRAM IN SOLUTION OF EXAMPLE 7

It may be supposed that the air is cooled between two sections of a duct. The quantities of energy convected across the two sections per pound of dry air crossing them are the two enthalpies calculated. Conservation of energy requires that the difference between these two enthalpies be the quantity of heat removed, or refrigeration supplied, between the two sections. Therefore,

$$-{}_Aq_B = 43{,}072 - 34.187 = 8.885 \text{ Btu/lb}_a$$

The initial volume is $13.980 + 0.50 \times 0.822 = 14.391$ cu ft/lb$_a$. Since 20,000 cfm of air is to be processed, the total refrigeration required is

$$-{}_AQ_B = 8.885 \times 20{,}000 \div 14.391 = 12{,}348 \text{ Btu per minute}$$

On the Goff Diagram the process is represented by the horizontal line AB, Fig. 3, whose length is the quantity of refrigeration required per pound of dry air.

Adiabatic Mixing of Two Air Streams

A typical air conditioning process requiring special analysis is the adiabatic mixing of two air streams. Referring to Fig. 4, let m_1, m_2, m_3 denote the weights of dry air convected across sections F_1, F_2, F_3, respectively,

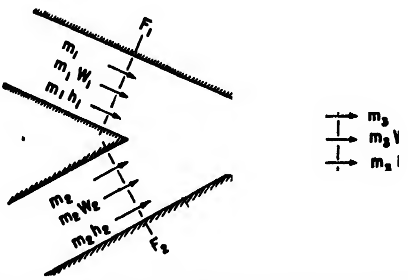

FIG. 4. ADIABATIC MIXING OF 2 AIR STREAMS

per minute. Then m_1W_1, m_2W_2, m_3W_3 and m_1h_1, m_2h_2, m_3h_3 will denote the weights of water and the quantities of energy similarly convected. If the mixing is adiabatic, it must be governed by the three equations,

$$\begin{aligned} m_1 + m_2 &= m_3 \\ m_1W_1 + m_2W_2 &= m_3W_3 \\ m_1h_1 + m_2h_2 &= m_3h_3 \end{aligned} \tag{8}$$

Elimination of m_3 gives,

according to which: *on the Goff Diagram the state point of the resulting mixture lies on the straight line connecting the state points of the two streams being mixed and divides the line into two segments which are in the same ratio as are the weights of dry air in the two streams.*

Example 8. Outside Air at 0 F and 80 per cent saturation is to be mixed adiabatically with recirculated Inside Air at 70 F and 20 per cent saturation in the ratio of one pound of dry air in the former to seven in the latter. Find the temperature and degree of saturation of the resulting mixture.

Solution. The humidity ratio W_3 and the enthalpy h_3 of the resulting mixture must satisfy Equations 9, namely,

$$\frac{0.003164 - W_3}{W_3 - 0.000630} = \frac{20.270 - h_3}{h_3 - 0.668} = \frac{1}{7}$$

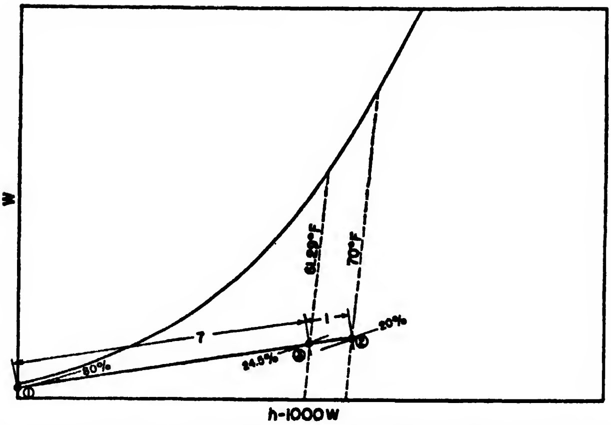

Fig. 5. Illustration of Use of Goff Diagram in Solution of Example 8

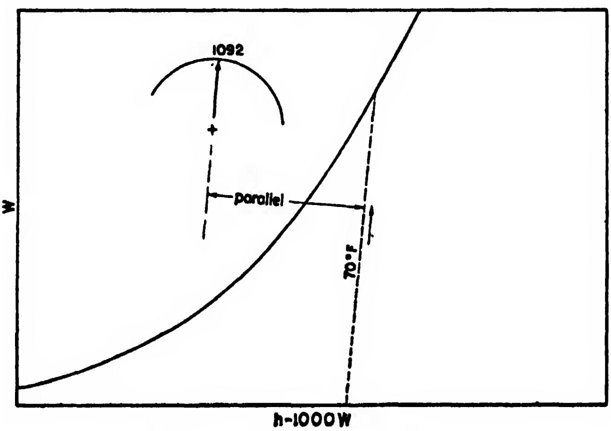

Fig. 6. Illustration of Use of Goff Diagram in Solution of Example 9

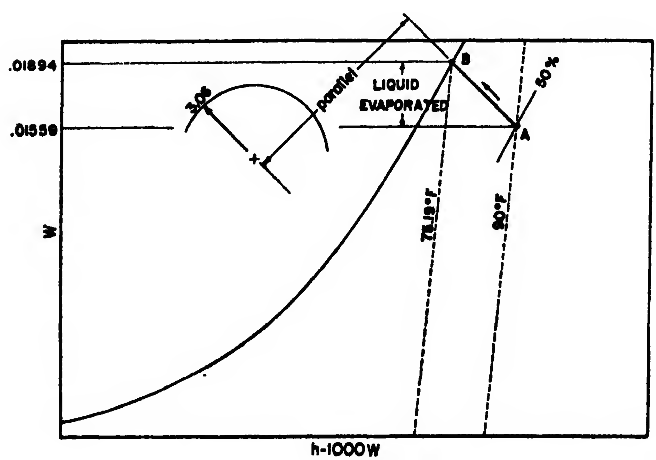

Fig. 7. Illustration of Use of Goff Diagram in Solution of Example 10

from which: $W_3 = 0.002847$, $h_3 = 17.820$. At the temperature of the resulting mixture, therefore,

$$h_a + 0.002847\ h_{as}/W_s = 17.820$$

At 61 F the lefthand member of this equation has the value 17.750; at 62 F its value is 17.991; by interpolation the temperature of the resulting mixture is 61.29 F where the humidity ratio at saturation, also by interpolation, is 0.01161; hence the degree of saturation of the resulting mixture is

$$\mu = 0.002847 \div 0.01161 = 24.52 \text{ per cent}$$

On the Goff Diagram, Fig. 5, a straight line is drawn between point 1 (0 F, 80 per cent) and point 2 (70 F, 20 per cent); then point 3 is located on the line one-eighth of the distance from point 2 to point 1. The temperature and degree of saturation at point 3 are read directly.

Adiabatic Mixing with Injected Water

Another typical air conditioning process is that of injecting water into an air stream to mix with it adiabatically. Let $W_2 - W_1$, denote the increase in humidity ratio of the air; this is obviously the quantity of water injected per pound of dry air; it follows that the quantity of energy injected per pound of dry air is $(W_2 - W_1)h_w$, where h_w denotes the specific enthalpy of the water as injected; if the process is adiabatic this produces an equal increase in the enthalpy of the air, namely, $h_2 - h_1$; therefore,

$$h_2 - h_1 = h_w(W_2 - W_1) \qquad (10)$$

according to which: *the process of injecting water into an air stream to mix adiabatically with it is represented by a straight line on the Goff Diagram whose direction is fixed by the specific enthalpy of the water as injected.* The protractor drawn on the Goff Diagram provided with this book provides a convenient means for determining this direction.

Example 9. It is desired to increase the humidity ratio of air at 70 F dry-bulb, without changing its temperature. Under what conditions may water be injected in order to accomplish the desired result?

Solution. At 70 F the increase of enthalpy per unit increase of humidity ratio is $h_{as}/W_s = 17.27 \div 0.01582 = 1092$ Btu per pound of water. This must be the specific enthalpy of the water added if the state point of the air is to be moved along the 70 F isotherm. Saturated steam at 668 F has this specific enthalpy[3].

On the Goff Diagram, Fig. 6, it is seen that the 70 F isotherm is parallel to the line on the protractor for a specific enthalpy of 1092 Btu per pound.

Adiabatic Saturation

Any process by which the state point of moist air is moved to the saturation curve adiabatically may properly be called adiabatic saturation.

Example 10. Liquid water chilled to 35 F is evaporated into an air stream initially at 90 F and 50 per cent saturation. How much water must be evaporated to bring the air to saturation at what temperature?

Solution. The initial enthalpy of the air is $21.625 + 0.50 \times 34.31 = 38.780$ Btu/lb$_a$; the initial humidity ratio is $0.50 \times 0.03118 = 0.01559$ lb$_w$/lb$_a$; the specific enthalpy of the chilled water is 3.06 Btu/lb$_w$; therefore, the temperature at which the air reaches the saturation curve must be such that the enthalpy h_s and humidity ratio W_s at saturation satisfy the equation,

$$h_s - (W_s - 0.01559) \times 3.06 = 38.780$$

The solution is 75.19 F where the humidity ratio at saturation is 0.01894; consequently, the weight of water evaporated is $0.01894 - 0.01559 = 0.00335$ lb per pound of dry air.

On the Goff Diagram, Fig. 7, a line is drawn through the initial state point (90 F,

50 per cent saturation) in the direction given by the protractor for a specific enthalpy of 3.06 Btu/lb_w.

If air is saturated adiabatically with spray water which is recirculated, the water will ultimately assume a temperature such that the air is brought to saturation at exactly the same temperature; that is, the water will assume the thermodynamic wet-bulb temperature of the air.

Example 11. Air at 75 F and 60 per cent saturation is saturated adiabatically with recirculated spray water. Find the resulting temperature and the weight of water added per pound of dry air.

Solution. In view of the foregoing remarks the solution of this example reduces to the determination of the thermodynamic wet-bulb temperature of the air. Its humidity ratio is $0.60 \times 0.01882 = 0.01129$; its enthalpy is $18.018 + 0.60 \times 20.59 = 30.372$; Equation 7 defining thermodynamic wet-bulb temperature becomes

$$h_s^* - (W_s^* - 0.01129)h_w^* = 30.372$$

At 65 F the value of the lefthand member is 29.995; at 66 F its value is 30.746; by interpolation the thermodynamic wet-bulb temperature is 65.51 F where the humidity

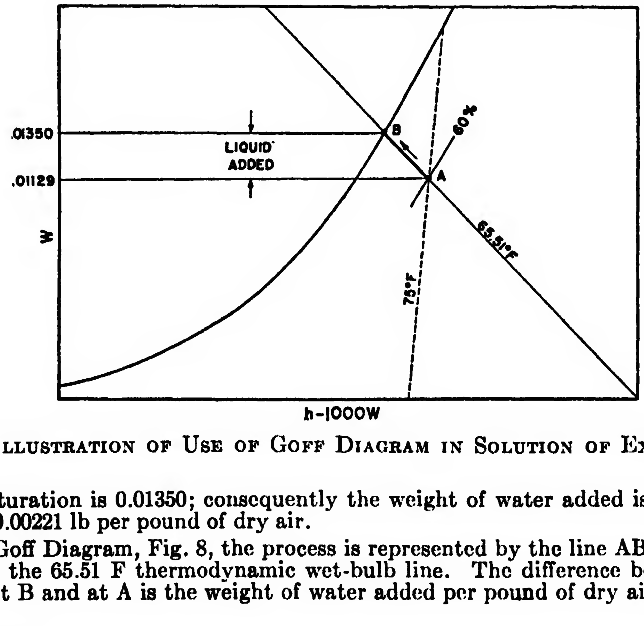

FIG. 8. ILLUSTRATION OF USE OF GOFF DIAGRAM IN SOLUTION OF EXAMPLE 11

ratio at saturation is 0.01350; consequently the weight of water added is $0.01350 - 0.01129 = 0.00221$ lb per pound of dry air.

On the Goff Diagram, Fig. 8, the process is represented by the line AB which is a segment of the 65.51 F thermodynamic wet-bulb line. The difference between the ordinates at B and at A is the weight of water added per pound of dry air.

Cooling Load

The problem of calculating the cooling load for an air conditioned space usually reduces to the determination of the quantity of *inside air* that must be withdrawn and the condition to which it must be brought by suitable processing so that its return to the conditioned space will have the net effect of removing given amounts of energy and water from the space.

Let M denote the weight of dry air withdrawn with *inside air* per hour. With it will be withdrawn energy of amount Mh_1 and water of amount MW_1 per hour, where h_1 and W_1 denote the enthalpy and humidity ratio of the *inside air*, respectively. The weight of dry air returned with the *conditioned air* will necessarily be the same as that withdrawn with the *inside air*, but with it must be returned a smaller quantity of energy Mh and a smaller quantity of water MW. Let ΔQ and ΔW denote the given

amounts of energy and water to be removed from the conditioned space per hour; then

$$Mh = Mh_1 - \Delta Q$$
$$MW = MW_1 - \Delta W$$

Eliminating M and letting q denote the ratio of energy removed to water removed, that is, $q = \Delta Q/\Delta W$,

$$\frac{h - h_1}{W - W_1} = q \tag{11}$$

according to which: *all possible states for the conditioned air lie on a straight line on the Goff Diagram passing through the state point of the inside air in the direction specified by the numerical value of the ratio q.* This line is called the *condition line* for the given problem. If the condition line crosses the saturation curve, the point of intersection is called the *apparatus dew-point* for the given problem.

The protractor of the Goff Diagram facilitates the drawing of the condition line and the locating of the apparatus dew-point. For this purpose the numerical value of the ratio q is to be regarded as a value of the *specific enthalpy of water added, Btu per pound.*

Example 12. A condition of 80 F dry-bulb, and 67 F thermodynamic wet-bulb, is to be maintained in a clothing store, outside conditions being 95 F dry-bulb, and 75 F thermodynamic wet-bulb. The energy gain from normal heat transmission is estimated at 16,000 Btu per hour, that from solar radiation at 48,000 Btu per hour. The energy generated by lights, fans, etc. is estimated at 13,900 Btu per hour. The ventilation requirement is 30,000 cu ft per hour. The number of occupants is 50. Find the apparatus dew-point.

Solution. The properties of *inside air* and *outside air* are readily calculated from the data in Table 1, see especially Example 2.

		Inside Air	Outside Air
μ	=	0.5024	0.3848
h	=	31.514	38.408
W	=	0.01122	0.01413
v	=	—	14.296

The weight of dry air entering with the *ventilating air* is 30,000/14.296 = 2098.5 lb per hour which brings with it energy of amount 2098.5 × 38.408 = 80.595 Btu per hour and water of amount 2098.5 × 0.01413 = 29.659 lb per hour.

The weight of dry air displaced from the store by the *ventilating air* is 2098.5 lb per hour which takes with it energy of amount 2098.5 × 31.514 = 66,132 Btu per hour and water of amount 2098.5 × 0.01122 = 23.541 lb per hour.

Each occupant may be regarded as a normal person standing at rest and evaporating (1386 grains) 0.198 lb of water per hour (value obtained by interpolation between Curves D and C Fig. 7, Chapter 12) at about 79 F. From this source there is water of amount 50 × 0.198 = 9.90 lb per hour and energy of amount 9.90 × 1095.7 = 10,847 Btu per hour added to the conditioned space. In addition each occupant loses 225 Btu per hour by conduction, convection, and radiation, making a total for 50 persons of 11,300 Btu per hour.

The net energy gain is 16,000 + 48,000 + 13,900 + 80,595 − 66,132 + 10,847 + 11,300 = 114,510 Btu per hour. The net water gain is 29.659 − 23.541 + 9.90 = 16.018 lb per hour. Accordingly the direction of the condition line is fixed by the ratio, q = 114,510 ÷ 16.018 = 7148.8 Btu per pound of water.

On the Goff Diagram, Fig. 9, the direction of the condition line is given by the protractor for a *specific enthalpy of water added* of 7148.8 Btu per pound. The line itself passes through the state point of the inside air and intersects the saturation curve at the apparatus dew-point.

According to Equation 11 the enthalpy h_s and humidity ratio W_s at the apparatus dew-point must satisfy the equation

$$7148.8W_s - h_s = 7148.8 \times 0.01122 - 31.514 = 48.681$$

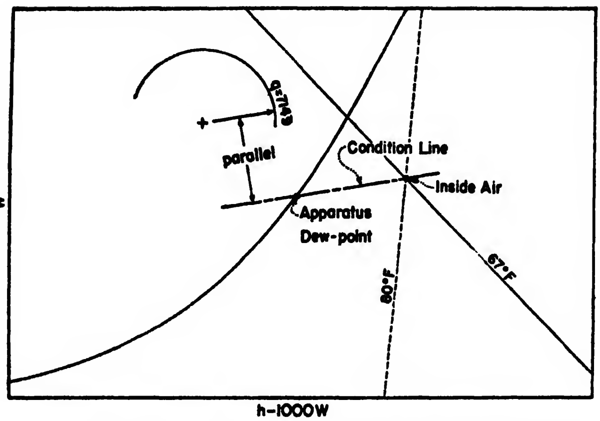

FIG. 9. ILLUSTRATION OF USE OF GOFF DIAGRAM IN SOLUTION OF EXAMPLE 12

At 58 F the left-hand member has the value 48.513; at 59 F its value is 50.641; by interpolation the apparatus dew-point is 58.08 F.

It would be a mistake to assume that the refrigeration to be supplied is equal to the net energy to be removed; for in general water is to be removed simultaneously and unless this is removed as liquid at 32 F it will automatically take some energy with it. Thus, unless the water is removed as solid (ice) the refrigeration to be supplied will be somewhat less than the net energy to be removed.

Exampl' 13. Referring to the cooling load problem of Example 12, suppose that the conditioning process consists of cooling a portion of the *inside air* to the apparatus dew-point temperature, separating out the liquid thus formed, and returning the resulting saturated mixture to the conditioned space. Find the quantity of *inside air* that must be processed n this manner and the corresponding quantity of refrigeration required.

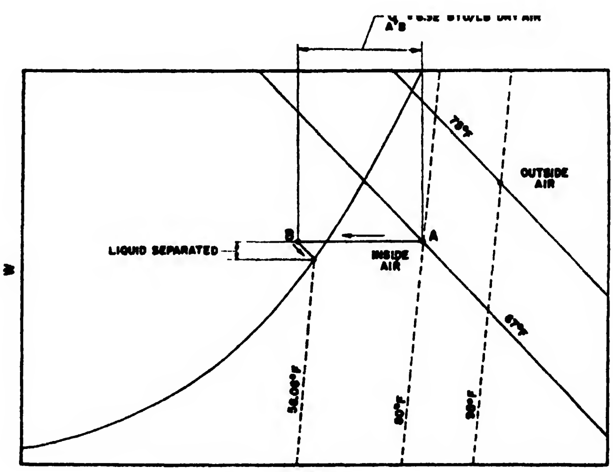

FIG. 10. ILLUSTRATION OF USE OF GOFF DIAGRAM IN SOLUTION OF EXAMPLE 13

Solution. During the cooling operation the enthalpy of the inside air is reduced to the value,

$$h = 25.17 + (0.01122 - 0.01033) \times 26.20 = 25.193$$

where 25.17 and 0.01033 are the values of enthalpy and humidity ratio at saturation at the apparatus dew-point temperature and 26.20 is the specific enthalpy of liquid water at that temperature. It follows that the quantity of refrigeration required is 31.514 − 25.193 = 6.321 Btu per pound of dry air.

The *inside air* being processed leaves the store with an enthalpy of 31.514 and is returned with an enthalpy of 25.17; it therefore removes energy of amount 6.344 Btu per pound of dry air. This means that the weight of dry air involved in the process is 114,510/6.344 = 18,050 lb per hour and that the total refrigeration to be supplied is 18,050 × 6.321 = 114,090 Btu per hour, or 9.508 tons.

The quantity of liquid separated out during the conditioning process is 18,050 × (0.01122 − 0.01033) = 16.018 lb per hour as required. In leaving the apparatus it takes with it energy of amount 16.018 × 26.20 = 420 Btu per hour. This plus the refrigeration accounts for the total energy removal of 114,510 Btu per hour as required.

On the Goff Diagram, Fig. 10, the cooling operation is represented by line AB whose length is the quantity of refrigeration per pound of dry air; the separation of the liquid formed in the cooling operation is represented by line BC whose projection on the ordinate axis is the quantity of liquid so separated per pound of dry air. Point C is the apparatus dew-point and lies on the condition line as required.

In practice it may not be feasible to choose the apparatus dew-point as the point on the condition line to which to condition the *inside air* because to do so would require an excessive number of air changes in the given space. Or it may be that the condition line does not cross the saturation curve at all so that the apparatus dew-point as defined does not exist. Finally, it is rarely possible to obtain complete saturation in conventional air conditioning apparatus. Nevertheless the requirements of the cooling load problem can be exactly met if the conditioned air is brought to *any* point on the condition line of the problem.

Heating Load

The condition line is also useful in the analysis of heating load problems as may best be illustrated by means of an illustrative example.

Example 14. A certain space is to be maintained at 70 F and 50 per cent saturation with outside conditions at 0 F and 80 per cent saturation. The normal heat transmission through walls, partitions, floor, roof, glass and doors is estimated at 75,000 Btu per hour. Energy gained from lights and appliances is estimated at 15,000 Btu per hour. Energy and water gains from occupants are to be disregarded in the calculations. Double doors and windows are used so that infiltration is negligible. The ventilation requirement is 30,000 cu ft per hour of *outside air*.

The requirements of the problem are to be met in the following manner; preheat the *ventilating air*; mix it adiabatically with recirculated *inside air*; saturate the mixture adiabatically with recirculated spray water; heat the resulting mixture to 105 F and return it to the conditioned space as *supply air*.

Analysis. Every pound of dry air admitted to the system (air conditioned space plus air conditioning apparatus) with the *ventilating air* displaces a pound of dry air from the system with *inside air*. Since the *ventilating air* is not admitted directly to the space, then for every pound of dry air withdrawn with *inside air* there is a pound of dry air returned with *supply air*. This has to have the net effect of adding energy of amount 60,000 Btu per hour and water of amount *zero* pounds per hour. Thus the ratio q determining the direction of the condition line is infinite, which means that the condition line is horizontal as indicated by the protractor on the Goff Diagram.

The properties of *inside air* are: $h = 25.451$, $W = 0.007910$. Since the state point of the *supply air* must be on the condition line at 105 F, its properties are: $h = 33.986$, $W = 0.007910$. Therefore the weight of dry air withdrawn with *inside air* and returned with *supply air* is 60,000 ÷ (33.986 − 25.451) = 7029.9 lb per hour.

The properties of *outside air* are: $h = 0.668$, $W = 0.0006298$, $v = 11.590$. Therefore the weight of dry air introduced into the system with the *ventilating air* is 30,000 ÷

11.590 = 2588.4 lb per hour. This *ventilating air* is to be mixed adiabatically with *inside air* containing 7029.9 − 2588.4 = 4441.5 lb of dry air per hour; therefore, the humidity ratio of the mixture must be (2588.4 × 0.0006298 + 4441.5 × 0.007910) ÷ 7029.9 = 0.005229.

The condition line crosses the saturation curve at 50.86 F where the enthalpy is 20.782 and the humidity ratio is 0.007910. This is the state point to be reached by adiabatic saturation of the mixture of *ventilating air* and *inside air* with recirculated spray water. Accordingly, the state point of the mixture must lie on the 50.86 F thermodynamic wet-bulb line so that its enthalpy must have the value,

$$h = 20.782 - (0.007910 - 0.005229) \times 18.97 = 20.731$$

This requires that the enthalpy of the preheated *ventilating air* have the value,

$$h = (7029.9 \times 20.731 - 4441.5 \times 25.451) \div 2588.4 = 12.632$$

Since the humidity ratio of the preheated *ventilating air* is known to be 0.0006298, its temperature is readily found to be 49.75 F.

The quantity of heat required for preheating the *ventilating air* is 2588.4 × (12.632 − 0.668) = 30,968 Btu per hour; that to be added to the *supply air* is 7029.9 × (33.986 − 20.782) = 92,823 Btu per hour; the energy added with the spray water is 7029.9 × 18.97 × (0.007910 − 0.005229) = 357 Btu per hour; that introduced into the

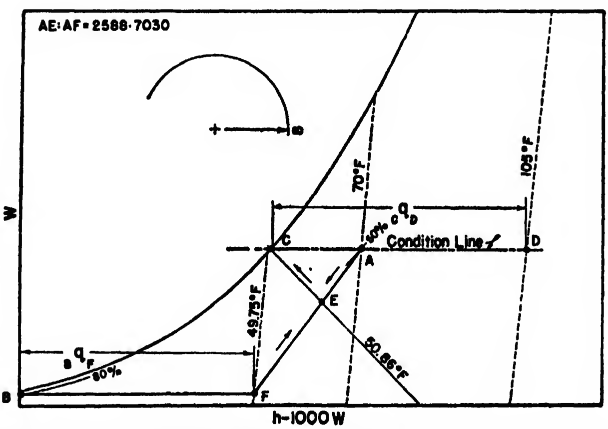

FIG. 11. ILLUSTRATION OF USE OF GOFF DIAGRAM IN SOLUTION OF EXAMPLE 14

system with the *ventilating air* is 2588.4 × 0.668 = 1729 Btu per hour; that carried out of the system with the *inside air* displaced by the *ventilating air* is 2588.4 × 25.451 = 65,877 Btu per hour; therefore, the net energy added to the system is 30,968 + 92,823 + 357 + 1729 − 65,877 = 60,000 Btu per hour as required.

On the Goff Diagram, Fig. 11, point A is the state point of the *inside air*. The condition line is horizontal so that point D is the state point of the *supply air*. The condition line crosses the saturation curve at point C so that the state point of the mixture of preheated *ventilating air* and *inside air* before adiabatic saturation with recirculated spray water must lie somewhere on the thermodynamic wet-bulb line through C. The state point of the *ventilating air* is point B, hence that of the preheated *ventilating air* must lie somewhere on the horizontal line through B. Its exact location is determined graphically by finding the straight line AF which is cut by the thermodynamic wet-bulb line through C into two segments such that AE:AF = 2588.4:7029.9. The length of the line BF is the quantity of heat required for preheating the *ventilating air* per pound of dry air; the length of the line CD is the quantity of heat to be added to the *supply air*, per pound of dry air.

WET-BULB TEMPERATURES BELOW 32F

A condition in which the water evaporating from the wick of a wet-bulb thermometer remains liquid at 32 F or lower is one of metastable equi-

librium and should therefore not be expected to occur in practice. The evidence that it does sometimes occur appears to be indirect and inconclusive. Stable equilibrium requires that the water freeze at 32 F or lower and is the condition to be expected in practice. On the Goff Diagram the lines of constant thermodynamic wet-bulb temperature have been drawn for stable equilibrium only. In other words it has been assumed that the water evaporating from the wick of the wet-bulb thermometer freezes when its temperature falls to 32 F or lower.

Example 15. Find the temperature at which dry air has a thermodynamic wet-bulb temperature of 32 F.

Solution. If it is assumed that the water evaporating from the wick of the wet-bulb thermometer remains liquid, the specific enthalpy of the dry air must have the value

$$h_a = 11.758 - 0.04 \times 0.003788 = 11.758$$

corresponding to which the temperature is 48.95 F. On the other hand if it is assumed that the water freezes, the specific enthalpy of the dry air must have the value,

$$h_a = 11.758 + 143.36 \times 0.003788 = 12.301$$

corresponding to which the temperature is 51.21 F. The second assumption is the assumption of stable equilibrium and should be expected to represent the actual situation.

The corresponding answer, namely 51.21 F, is the one given by the Goff Diagram at intersection of 32 F thermodynamic wet-bulb and 0 per cent saturation.

DALTON'S RULE

As stated in the introduction the thermodynamic properties of moist air have hitherto been obtained from those of dry air and water vapor separately by application of Dalton's Rule. Actual departures from the rule are due principally, but not entirely, to intermolecular forces; therefore, in order to apply the rule with any measure of consistency it is necessary to idealize the situation by assuming that the effects of such intermolecular forces are negligible and that both the dry air and the water vapor behave like perfect gases. Making this assumption, the volume v_T occupied by n_a mols of dry air at temperature T and pressure p_a is $v_T = n_aRT/p_a$ while that occupied by n_w mols of water vapor at the same temperature but at pressure p_w is $v_T = n_wRT/p_w$. According to Dalton's Rule, if the dry air and water vapor are mixed, each occupies the whole volume of the mixture at the temperature of the mixture and the pressure of the mixture is the sum of the individual pressures. Mathematically,

$$v_T = \frac{n_aRT}{p_a} = \frac{n_wRT}{p_w} = \frac{(n_a + n_w)RT}{p} \tag{12}$$

It follows from these equations that the so-called *partial* pressure of each constituent is its mol-fraction times the observed pressure of the mixture; thus, for water vapor,

$$p_w = \frac{n_w}{n_a + n_w}\,p \tag{13}$$

and similarly for dry air. Equation 13 may be regarded as the Dalton Rule, definition of partial pressure in terms of the observable terms n_a, n_w, p.

The humidity ratio W is the mol ratio n_w/n_a times the ratio of molecular

weights, namely, 18.016/28.966 = 0.6220; hence Equation 13 can be written

$$W = 0.6220 \frac{p_w}{p - p_w} \tag{14}$$

Now, even if it is assumed that both the dry air and the water vapor behave like perfect gases, it does not follow that at saturation the *partial* pressure of the water vapor can be put equal to the saturation pressure of pure water at the temperature of the mixture because: (1) the coexisting liquid (or solid) phase is not pure water but contains a small amount of dissolved air, and (2) the coexisting liquid (or solid) phase has to support the observed pressure p and not just the saturation pressure p_s of pure water. These effects are calculable but are in general smaller than the effects of intermolecular forces which have already been ignored. Besides, the only legitimate reason for retaining Dalton's Rule is to gain simplicity; hence these effects should be disregarded also, and the humidity ratio at saturation estimated as follows,

$$W_s = 0.6220 \frac{p_s}{p - p_s} \tag{15}$$

In this chapter the ratio W/W_s has been called degree of saturation and denoted by the greek letter μ. The ratio p_w/p_s has long been called *relative humidity* and will be denoted by the Greek letter φ. Equations 14 and 15 can be combined to give

$$\mu = \varphi \frac{1 - p_s/p}{1 - \varphi p_s/p} \tag{16}$$

which can be inverted to give

$$\varphi = \frac{\mu}{1 - (1 - \mu)p_s/p} \tag{17}$$

Example 16. Find the relative humidity of moist air at 180 F, 20 per cent saturation.

Solution. Inserting numerical data from Table 1 into Equation 17, the answer is

$$\varphi = \frac{0.20}{1 - 0.80 \times 15.294/29.921} = 0.3384$$

Example 17. Find the degree of saturation of moist air at 70 F, 50 per cent relative humidity.

Solution. Inserting numerical data from Table 1 into Equation 16, the answer is

$$\mu = 0.50 \frac{1 - 0.73915/29.921}{1 - 0.50 \times 0.73915/29.921} = 0.4937$$

The foregoing examples show that there is a substantial difference between degree of saturation and relative humidity, particularly at higher temperatures. Of course, they both have the value zero for dry air and the value unity for saturated moist air regardless of the temperature.

A Dalton Rule expression for the volume of moist air per pound of dry air obtainable directly from Equation 12 is

$$v = \left(\frac{R_a T}{p}\right) + \mu\left(\frac{W_s R_w T}{p}\right) \tag{18}$$

where

R_a = gas constant for dry air = 1545.31 ÷ 28.966 = 53.349 (ft/F).
R_w = gas constant for water vapor = 1545.31 ÷ 18.016 = 85.774 (ft/F).

This expression is of the form of Equation 4.

According to Dalton's Rule the enthalpy of moist air is the sum of separate contributions from the dry air and the water vapor; thus,

$$h = h_a + \mu(W_s h_w) \tag{19}$$

where, to be consistent, the specific enthalpies h_a and h_w should be allowed to vary with temperature only, not with pressure or composition. This expression is of the form of Equation 5.

Within the accuracy of Dalton's Rule the following empirical equations give suitable values of h_a and h_w:

$$\begin{aligned} h_a &= 0.240t \\ h_w &= 0.444t + 1061 \end{aligned} \tag{20}$$

Equation 7 defining thermodynamic wet-bulb temperature may be written in the form,

$$h - h' + (W_s^* - W)h_w^* = h_s^* - h'$$

If the quantity h' that has been subtracted from both sides is understood to be the enthalpy at the thermodynamic wet-bulb temperature t^* but at the humidity ratio W, then within the accuracy of Dalton's Rule

$$\begin{aligned} h - h' &= (0.240 + 0.444W)(t - t^*) \\ h_s^* - h' &= (1061 + 0.444t^*)(W_s^* - W) \\ h_w^* &= t^* - 32 \end{aligned}$$

With these approximations Equation 7 becomes

$$W_s^* - W = \frac{0.240 + 0.444W}{1093 - 0.556t^*}(t - t^*) \tag{21}$$

Carrier[6] has modified Equation 21 by introducing further approximations as follows,

$$\begin{aligned} W_s^* &= 0.6220p_s^*/(p - p_s^*) \\ W &= 0.6220p_w/(p - p_s^*) \\ 0.444\,W &= 0 \end{aligned}$$

the first of which is part of Dalton's Rule. The result is

$$p_w = p_s^* - \frac{p - p_s^*}{2830 - 1.44t^*}(t - t^*) \tag{22}$$

except that the numerical values of the constants in the denominator of the rightmost term are somewhat different than Carrier's.

Equation 22 permits direct calculation of the partial pressure p_w from observed values of pressure p, temperature t, and wet-bulb temperature t^*, assuming that information is available regarding the saturation pressure p_s. The ratio p_w/p_s is the so-called relative humidity.

Example 18. **Find the relative humidity of moist air at 90 F dry-bulb, and 63 F (thermodynamic wet-bulb).**

Solution. At 63 F the value of the saturation pressure is 0.58002 in. Hg. Therefore, at atmospheric pressure (29.921 in. Hg),

$$p_w = 0.58002 - 29.341 \times 27/2739 = 0.2908 \text{ in. Hg}$$

The relative humidity is

$$\varphi = 0.2908/1.4219 = 0.2045$$

the denominator being the value of saturation pressure at 90 F.

From Equation 16 may be computed the corresponding degree of saturation, the result being

$$\mu = 19.67 \text{ per cent}$$

in remarkably close agreement with the answer to Example 2.

STEADY FLOW ENERGY EQUATION

In steady flow, the energy convected by the fluid at any section is the sum of (*a*) kinetic energy due to velocity; (*b*) gravitational energy due to elevation; (*c*) enthalpy due to the condition of pressure, temperature and composition of the fluid.

Kinetic Energy

There are reasons to believe that the so-called *velocity pressure* h_v read by a Pitot tube is simply the kinetic energy per unit volume of the fluid immediately upstream from the tube, as application of Bernoulli's Equation suggests. Thus

$$V = 1097.3 \sqrt{\frac{h_v}{\rho}} \tag{23}$$

where

V = velocity, feet per minute.
h_v = velocity pressure, inches of water at 60 F.
ρ = density of fluid, pounds per cubic foot.

In the case of flow through a duct, the velocity pressure is found to vary considerably over the section and a traverse has to be made. The cross-sectional area of the duct is divided into a number of equal concentric areas, and measuring stations are located at centroidal points in each area along two perpendicular diameters. Usually the ultimate object is to determine an *average* velocity $\overline{V}$ from which the weight of fluid crossing the section per unit time can be obtained on multiplying by the cross-sectional area of the duct and by the density of the fluid. This is obtained by simply averaging the square roots of all measured velocity pressures as follows:

$$\overline{V} = \frac{1097.3}{\sqrt{\rho}} (h_v^{1/2})_{av} \tag{24}$$

where

$\overline{V}$ = average velocity, feet per minute.
$(h_v^{1/2})_{av}$ = arithmetic average of the square roots of all measured velocity pressures, inches of water at 60 F.

But the item of present importance is the *average kinetic energy* con-

vected with each pound of fluid. Consistently with the previous discussion, this can be shown to be

$$\overline{KE} = 0.006678\, v \frac{(h_v^{3/2})_{av}}{(h_v^{1/2})_{av}} \tag{25}$$

$\overline{KE}$ = average kinetic energy, Btu per pound.
v = specific volume, cubic feet per pound.
$(h_v^{3/2})_{av}$ = arithmetic average of the 3/2-powers of all measured velocity pressures, inches of water at 60 F.

If the velocity pressure were uniform over the section, Equations 24 and 25 could be combined to give

$$\overline{KE} = \left(\frac{\bar{V}}{13{,}430}\right)^2 \tag{26}$$

But, it is interesting to note that if the velocity varies *parabolically* from zero at the walls to maximum at the center as it does in the case of purely viscous flow in a circular duct, *then the average kinetic energy is twice that given by* Equation 26.

Example 19. If 2000 cfm of air flow through an 8 in. diameter circular duct, find the average kinetic energy per pound of air.

Solution. The cross-sectional area of the duct is 0.349 sq ft; hence the average flow velocity is 5730 fpm. If the velocity were uniform over the section, the average kinetic energy would be $(5730 \div 13{,}430)^2 = 0.182$ Btu per pound. But it is more likely that the actual distribution of velocity would approximate that characteristic of viscous flow; hence the average kinetic energy would be more nearly $2 \times 0.182 = 0.364$ Btu per pound.

Gravitational Energy

The potential energy due to elevation Z (feet) above any convenient datum is simply $Z \div 778.3$ Btu per pound of fluid. In the case of moist air,

$$\overline{PE} = \frac{\bar{Z}(1 + W)}{778.3} \tag{27}$$

where

$\overline{PE}$ = average potential energy, Btu per pound dry air.
Z = average elevation, feet.
W = humidity ratio, pound water per pound dry air.

Enthalpy

No further discussion of enthalpy is required. It may be well to emphasize, however, that enthalpies have been figured on the basis of one pound of dry air.

Heat and Shaft Work

Between any two sections 1 and 2 in an apparatus through which steady flow occurs, there may be heat *absorbed* from outside ${}_1q_2$, Btu per pound of dry air, and shaft work *removed* to outside, ${}_1l_2$, Btu per pound of dry air. If heat is actually rejected to outside, ${}_1q_2$ is intrinsically negative; and if shaft work is actually put in from outside, ${}_1l_2$ is intrinsically negative.

Steady-flow Energy Equation

A complete energy accounting takes the form of Equation 28 which is usually referred to as the steady-flow energy equation.

$$ {}_1q_2 = (h_2 + \overline{KE}_2 + \overline{PE}_2) - (h_1 + \overline{KE}_1 + \overline{PE}_1) + {}_1l_2 \tag{28}$$

where

${}_1q_2$ = heat *added* from outside *between* sections 1 and 2, Btu per pound dry air.
h_2 = enthalpy of the mixture *at* section 2, Btu per pound dry air.
$\overline{KE}_2$ = average kinetic energy *at* section 2, Btu per pound dry air.
$\overline{PE}_2$ = average potential energy *at* section 2, Btu per pound dry air.
h_1 = enthalpy *at* section 1, Btu per pound dry air.
$\overline{KE}_1$ = average kinetic energy *at* section 1, Btu per pound dry air.
$\overline{PE}_1$ = average potential energy *at* section 1, Btu per pound dry air.
${}_1l_2$ = shaft work withdrawn *between* sections 1 and 2, Btu per pound dry air.

In Equation 28 all quantities are *per pound of dry air*. If Equation 25 is used in computing average kinetic energy, the result will be in Btu per pound of dry air if v is taken as volume per pound of dry air. If Equation 26 is used, multiplication by $(1 + W)$ as in Equation 27 is required though this is a refinement seldom justified.

Thermodynamic properties of water at saturation are given in Table 2 for the range -160 to $+212$ F.

U. S. STANDARD ATMOSPHERE

The so-called U. S. Standard Atmosphere is an essential standard of reference in aeronautics and as such has become important to the air conditioning engineer who frequently has to simulate atmospheric conditions at high altitudes in connection with aeronautical research. In defining this standard it is first assumed that temperature T varies linearly with altitude Z above sea level, at any rate up to the lower limit of the isothermal layer at 35,332 ft. Thus,

$$T = T_o - 0.0019812\ Z \tag{29}$$

or

$$\frac{dT}{dZ} = -0.0019812 \text{ (degree Centigrade per foot)} \tag{30}$$

The second assumption is the validity of the perfect gas laws, namely,

$$Pv = RT \tag{31}$$

A horizontal disc of air having unit cross-sectional area (1 sq ft) and vertical thickness dZ (ft) weights dZ/v (lb). This accounts for the difference of pressure dP (lb per sq ft) between the upper and lower faces of the disc; hence, using Equation 31

$$dZ = \frac{RT\,dP}{P} \tag{32}$$

Equations 30 and 32 can be combined to eliminate Z and then integrated to obtain the relation between pressure and temperature, namely,

$$\frac{T}{T_o} = \left(\frac{P}{P_o}\right)^{0.1903} \tag{33}$$

TABLE 5. PRESSURE AND TEMPERATURE FOR ALTITUDES IN U. STANDARD ATMOSPHERE

ALTITUDE FEET Z	PRESSURE IN. OF HG P	TEMP F t
− 1,000	31.02	+62.6
− 500	30.47	+60.8
0	29.921	+59.0
+ 500	29.38	+57.2
+ 1,000	28.86	+55.4
+ 5,000	24.89	+41.2
10,000	20.58	+23.4
15,000	16.88	+ 5.5
20,000	13.75	−12.3
25,000	11.10	−30.1
30,000	8.88	−47.9
35,000	7.04	−65.8
40,000	5.54	−67.0
45,000	4.36	−67.0
50,000	3.436	−67.0

The values $T_o = 288$ K and $P_o = 29.921$ in. Hg are parts of the definition of the standard atmosphere.

Values of pressure and temperature are listed in Table 5 for altitudes in the standard atmosphere from −1,000 to 50,000 ft above sea level. Values for altitudes below the lower limit of the isothermal layer conform to Equations 29 and 33. For further explanation, reference (7) should be consulted.

LETTER SYMBOLS USED IN CHAPTER 3

μ = degree of saturation or per cent saturation.

ρ = density of fluid, pounds per cubic foot.

φ = relative humidity (decimal).

a = ratio of apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016) = 1.6078.

A = coefficient from Table 4 for use in Equation 4a (obtained from Table 4).

B = coefficient to be used in Equation 5a (obtained from Table 4).

C = coefficient for use in Equation 6a (obtained from Table 4).

h = enthalpy of moist air, Btu per pound of dry air.

$\bar{h}$ = enthalpy correction term to be added above 150 F, to enthalpy.

h_a = specific enthalpy of dry air, Btu per pound.

h_{as} = $h_s - h_a$ = the difference between the enthalpy of moist air *at saturation* per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_s^* = enthalpy of moist air at saturation at *thermodynamic wet-bulb temperature*, Btu per pound of dry air.

LETTER SYMBOLS (Continued)

h_s = enthalpy of moist air at saturation per pound of dry air, Btu per pound of dry air.

h_v = velocity pressure, inches of water at 60 F.

h_w = specific enthalpy of condensed water (liquid or solid) at standard pressure, Btu per pound water.

h_w^* = specific enthalpy of water as added at the *thermodynamic wet-bulb* temperature t^*, Btu per pound of dry air.

K = Kelvin degrees.

KE = kinetic energy, Btu per pound.

$\overline{KE}$ = average kinetic energy, Btu per pound.

l = shaft work withdrawn, Btu per pound of air.

${}_1l_2$ = shaft work withdrawn between sections 1 and 2, Btu per pound of air.

m = weight of dry air crossing any duct section, pounds per minute.

m_1, m_2, m_3 = weights of dry air convected across sections F_1, F_2, F_3 respectively, pounds per minute.

M = weight of dry air withdrawn with inside air, pounds per hour.

n_a = mols of dry air.

n_w = mols of water vapor.

p = total pressure of a mixture of air and water vapor, pounds per square inch or inches Hg.

p_a = partial pressure of air, pounds per square inch or inches Hg.

p_s = *saturation* pressure of pure water vapor, pounds per square inch or inches Hg.

p_w = partial pressure of water vapor in mixture of air and water vapor, pounds per square inch or inches Hg.

P = atmospheric pressure, inches Hg.

P_0 = standard atmospheric pressure by definition 29.921 in. Hg.

PE = potential energy, Btu per pound dry air.

$\overline{PE}$ = average potential energy, Btu per pound dry air.

q = ratio of energy added (or removed) to water added (or removed), Btu per pound. Also called specific enthalpy of *water added*.

${}_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.

${}_Aq_B$ = heat added between sections A and B per pound of dry air, Btu per pound.

${}_AQ_B$ = total heat added between sections A and B, Btu per minute.

ΔQ = energy to be removed from or added to conditioned space, Btu per hour.

R = universal gas constant.

R_a = gas constant for dry air.

R_w = gas constant for water vapor.

LETTER SYMBOLS (Continued)

s = entropy of moist air per pound of dry air, Btu per (pound) (Fahrenheit degree).

s = correction to be added to entropy of moist air obtained from Equation 6.

$\bar{s}$ = additional correction to be added to entropy because of "mixing entropy" (obtained from Table 4). Correction to be added to value of s obtained from Equation 6.

s_a = specific entropy of dry air, Btu per (pound) (Fahrenheit degree, absolute).

s_{as} = the difference between the entropy of moist air *at saturation* per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (Fahrenheit degree, absolute).

s_s = entropy of moist air at saturation per pound of dry air, Btu per (pound of dry air) (Fahrenheit degree, absolute).

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (Fahrenheit degree, absolute).

t^* = thermodynamic wet-bulb temperature, Fahrenheit degrees.

t(F) = temperature, Fahrenheit degrees.

T = absolute temperature, Fahrenheit degrees.

T_o = standard atmospheric temperature, by definition 288 Kelvin degrees.

v = volume of moist air per pound of dry air, cubic feet per pound.

$\bar{v}$ = correction to be added to volume of moist air per pound of dry air, above 150 F.

v_a = specific volume of dry air, cubic feet per pound.

v_{as} = $v_s - v_a$, the difference between volume of moist air *at saturation*, per pound of dry air, and the volume of the dry air itself, cubic feet per pound of dry air.

v_s = volume of moist air at saturation per pound of dry air, cubic feet per pound of dry air.

v_T = total volume, cubic feet.

V = velocity, feet per minute.

$\bar{V}$ = average velocity, feet per minute.

W = humidity ratio, of moist air, pounds of water per pound of dry air.

ΔW = water to be removed from (or added to) conditioned space, pounds per hour.

W_s = humidity ratio, *at saturation*, weight of water vapor per pound of dry air, pound per pound.

W_s^* = humidity ratio corresponding to thermodynamic wet-bulb temperature t^*, pounds of water per pound of dry air.

Z = elevation above any datum, feet.

$\bar{Z}$ = average elevation, feet.

REFERENCES

[1] Thermodynamic Properties of Moist Air, by J. A. Goff and S. Gratch (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 125).

[2] Low Pressure Properties of Water in the Range −160 to 212 F, by J. A. Goff and S. Gratch (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February 1946).

[3] Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes (John Wiley and Sons, Inc., New York, 1936).

[4] Das i, x-Diagramm für Dampfluftgemische, by R. Mollier (ZVDI, 1929).

[5] The Theory of the Psychrometer, by J. H. Arnold (*Physics*, Vol. 4, 1933).

[6] Rational Psychrometric Formulae, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 33, 1911, p. 1005).

[7] National Advisory Committee for Aeronautics, Technical Report No. 538, 1935.

CHAPTER 4

FLUID FLOW

Theory; Pressure Loss in Circular and Non-Circular Pipes; Compressible Fluids; Nozzles and Orifices; Steam Flow Measurement; Metering Liquids; Nozzle Coefficients and Expansion Factors; Pitot Tube; Installation of Nozzles and Orifices, Variable Area Flow Meters

THE flow of fluids is part of the branch of engineering science known as fluid mechanics, which will be discussed here insofar as it applies to the work of engineers in the fields of heating, ventilating, and air conditioning. Probably air is the most frequently handled fluid, but other gases and liquids are often involved. Compressible fluids (gases) and incompressible fluids (liquids) vary somewhat in behavior, though in cases where pressure and density changes are small, the gases may be treated as incompressible fluids.

THEORY OF FLUID FLOW

The following energy equation for one dimensional steady flow processes will serve as a basis for the theory of the flow of fluids. This equation is presented in several ways in various texts, but a suitable form is

$$\frac{V_1^2}{2g_c} + Ju_1 + p_1v_1 + Jq + \frac{g}{g_c} z_1 = \frac{V_2^2}{2g_c} + Ju_2 + p_2v_2 + W + \frac{g}{g_c} z_2 \qquad (1)$$

where

V = velocity in feet per second.
g = gravitational acceleration, in feet per (second) (second).
g_c = gravitational conversion factor = 32.174 (pounds mass per pound force) × ft per (second) (second).
J = mechanical equivalent of heat = 778 foot pounds per Btu.
u = internal energy, in Btu per pound of fluid.
p = pressure in pounds per square foot.
v = specific volume, in cubic feet per pound.
W = mechanical work done by the fluid in foot pounds per pound of fluid.
q = heat transferred to the fluid in Btu per pound of fluid flowing.
z = elevation above some arbitrary datum, in feet.
Subscript 1 refers to the entrance, subscript 2 to the exit.

Introducing the enthalpy h, which by definition is $u + \frac{pv}{J}$, expressed in Btu per pound of fluid, Equation 1 becomes

$$\frac{V_1^2}{2g_c} + Jh_1 + Jq + \frac{g}{g_c} z_1 = \frac{V_2^2}{2g} + Jh_2 + W + \frac{g}{g_c} z_2 \qquad (2)$$

The equivalent differential form for energy Equation 1 is

$$\frac{1}{2g_c} dV^2 + J\,du + d(pv) + \frac{g}{g_c} dz - J\,dq + dW = 0 \qquad (3)$$

Replacing v by its equal $g/g_o\rho$ (where ρ is density in pounds weight per cubic foot) and rearranging, Equation 3 becomes

$$\frac{1}{2g}dV^2 + \frac{1}{\rho}dp + dz + \frac{g_c}{g}[J\,du + p\,dv - J\,dq + dW] = 0 \tag{4}$$

In the case of flow through a pipe, no outside work is performed so that $dW = 0$. Furthermore,

$$J\,du + p\,dv = JT\,ds = J\,dq + JT\,ds' \tag{5}$$

where,

ds = total change in entropy.

ds' = change in entropy due to internal irreversibility from turbulence and friction.

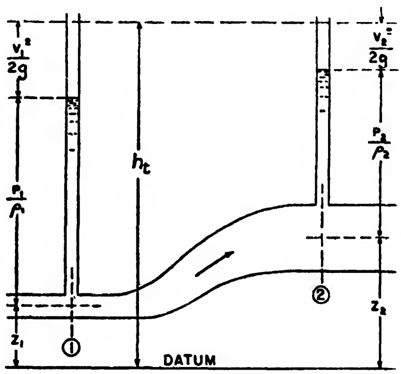

FIG. 1. RELATION OF VARIOUS FACTORS IN BERNOULLI EQUATION

Accordingly, Equation 4 may be written

$$\frac{1}{2g}dV^2 + \frac{dp}{\rho} + dz + \frac{g_c}{g}JT\,ds' = 0^* \tag{6}$$

In cases where there is no internal irreversibility, $ds' = 0$, and Equation 6 may be integrated to give

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_m} + \quad = \frac{V_2^2}{2g} + \frac{p_2}{\rho_m} + z_2 \tag{7}$$

where ρ_m is the proper mean density.

This is commonly called the Bernoulli equation, named after the Swiss mathematician and physician who first propounded the theory. $\frac{V^2}{2g}$ is known as the velocity head, $\frac{p}{\rho}$ is the pressure head, and z is the elevation head, all in feet of the fluid; the total head, h_t is the sum of the other three heads. Fig. 1 shows diagrammatically the relation of the various factors. The pressure at point 2 is lower than at point 1 because of the elevation of point 2 over point 1, and the velocity at point 2 is lower than at point 1 because of the larger pipe diameter at point 2. If the

* In the analysis of subsequent portions of this chapter the distinction between g and g_c will be omitted. Aside from dimensional consistency the factor, g/g_c, is not in general significant in fluid flow analysis.

pipe diameter were the same throughout, the velocity, and consequently the velocity head, would be the same at both points, but the higher elevation at point 2 would still be responsible for a loss in pressure. The utility of the equation is evident, though it should be remembered that in it the *effects of friction and turbulence are neglected*, and that Fig. 1 represents ideal conditions. It should also be noted that care must be

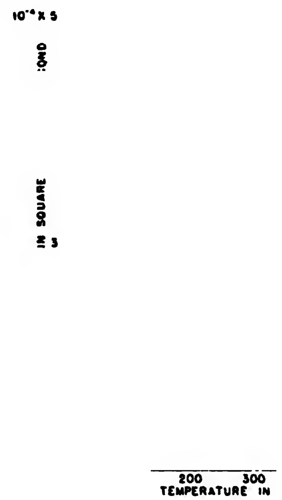

Fig. 2. Relation of Kinematic Viscosity to Temperature of Air

taken in determining the proper mean density. Accordingly, the Bernoulli equation is applied most conveniently to incompressible fluids for which density is constant.

Pressure Loss in Circular Pipes

The pressure loss in circular pipes is customarily expressed by the formula:

$$h_f = \frac{\quad}{2g\,d} \qquad (8)$$

where

h_f = the loss in head of the fluid under conditions of flow, in feet.
l = the length of the pipe, in feet.
V = the velocity, in feet per second.
g = the acceleration due to gravity = 32.174 ft per (second) (second).
d = the internal diameter of the pipe, in feet.
f = a dimensionless friction coefficient.

The formula is generally known by the name of Darcy or Fanning, though it seems to have been originated by d'Aubisson de Voisins in 1834.

The factor f is a function of the Reynolds number,

(9)

where

N_{Re} = Reynolds number.
ρ = the density in pounds per cubic foot.
μ = the absolute viscosity in pounds per foot-second.

Both f and the Reynolds number are dimensionless. To aid in computing the Reynolds number, values of $\frac{\mu}{\rho}$, the kinematic viscosity, are shown as a function of temperature for air in Fig. 2 and for water in Fig. 3.

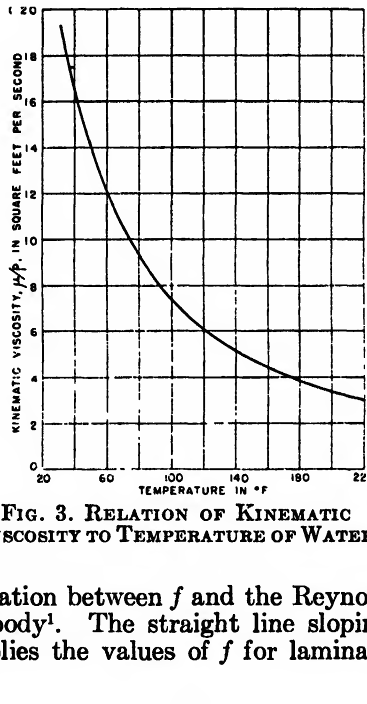

FIG. 3. RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF WATER

Fig. 4 shows the relation between f and the Reynolds number, adapted from a review by Moody[1]. The straight line sloping downward at the left of the chart supplies the values of f for laminar flow; it represents the formula:

$$64 \qquad (10)$$

With laminar flow, the velocity profile is a parabola, having the formula

$$V = \frac{\cdots}{\cdots} (r^2 - \qquad (11)$$

where

r = the radius of the pipe in feet.
L = distance perpendicularly from the axis of the pipe, in feet.

Accordingly, the maximum velocity occurs at the center of the pipe and is twice the average velocity; the average velocity is found when $L = 0.707\ r$. It is worth noting that roughness of the pipe wall has no effect on the loss in head for laminar flow.

Between values of the Reynolds number of 2000 and 4000, there is an

[1] Superior numbers refer to the references at the end of the chapter.

unstable region where the flow changes from laminar to turbulent, or vice versa. The actual value is impossible of prediction for any conditions of flow, though in general it may be said that the prevailing type of flow persists into the unstable region; however, once the change starts, it proceeds very rapidly.

When the flow is turbulent, the velocity profile is essentially parabolic over four fifths of the pipe diameter, but near the pipe walls, the effect of friction becomes evident, and in the boundary layer at the pipe wall the flow is laminar. Fig. 5 compares the velocity profiles for three different Reynolds numbers, but for the same average velocity.

The lower curve in the turbulent region in Fig. 4 represents the relation of f to the Reynolds number for smooth pipe, such as drawn brass tubing

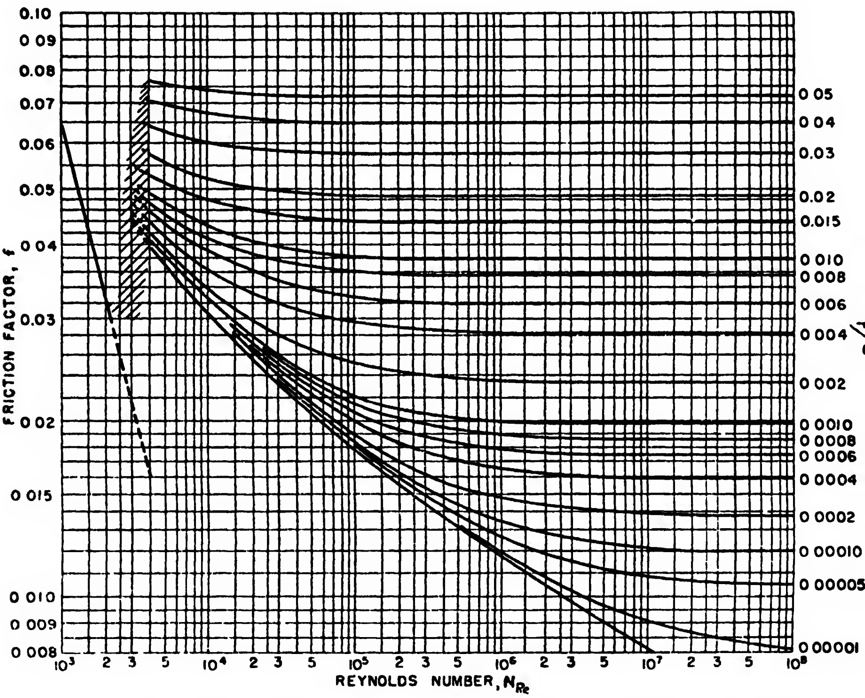

FIG. 4. RELATION BETWEEN FRICTION FACTOR AND REYNOLDS NUMBER
NOTE: The straight line at left shows values of Friction Factor for laminar flow.
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or glass tubing. The effect of roughness on f, which is a considerable factor in turbulent flow, is open to some conjecture; artificially roughened pipes, for instance, give results at variance with actual tests. The curves above the smooth pipe curve of Fig. 4 represent a summary of tests on rough pipe, each of them identified by a value of e/d with e signifying the absolute roughness in feet. Values of e/d for different pipes are given in Table 1.

To find the friction loss for any pipe, follow the curve with the proper value of e/d, to the pertinent value of N_{Re}; and from this point proceed horizontally to left margin to find the value of f to use in Equation 8.

TABLE 1. VALUES OF e/d FOR DIFFERENT KINDS OF PIPE

TYPE OF PIPE	e/d
Smooth drawn tubing	0.000005
Commercial steel or wrought iron	0.00015
Asphalted cast-iron	0.0004
Galvanized iron	0.0005
Cast-iron	0.00085
Wood stave	0.0006 to 0.003
Concrete	0.001 to 0.01
Riveted steel	0.003 to 0.03

The curves in Fig. 4 may be approximated very closely by the empirical formula[1]:

$$f = .0055\left[1 + \left(20{,}000\,\frac{e}{d} + \frac{10^6}{r}\right)^{1/3}\right] \tag{12}$$

Equation 8 is applicable to all liquids, and to gases when the pressure loss is less than 10 per cent of the initial pressure. When the loss in head is high, the formula to be used for gases is

$$\frac{p_1^2 - p_2^2}{p_1^2} = \frac{flV_1^2}{gd\, p_1 v_1} \tag{13}$$

which may be rearranged to give the loss in pressure,

$$p_1 - p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{gd\, p_1 v_1}}\right] \tag{14}$$

Pressure Loss in Non-Circular Pipes

The formulas for flow in pipes are based upon the use of pipes of circular cross-section. The formulas may be used with conduits of other shapes, and in conduits not flowing full, when the flow is turbulent, by using the hydraulic radius, R_H, which is really a ratio:

$$R_H = \frac{\text{area of cross-section}}{\text{wetted perimeter of cross-section}} \tag{15}$$

For instance, in a square duct, 1 ft on a side, handling air, the hydraulic radius is $\frac{1}{4}$ or 0.25. If the same duct is handling water, flowing 9 in. deep, the hydraulic radius is 0.75/2.5 or 0.30. Note in this latter case that the wetted perimeter does not include the distance across the free surface.

In the case of a round pipe

$$R_H = \frac{\pi d^2/4}{\pi d} = \frac{d}{4} \text{ or } d = 4R_H \tag{16}$$

Substituting Equation 16 in Equation 9,

$$N_{Re} = \frac{4R_H V\rho}{\mu} \tag{17}$$

and in the flow Equation 8,

$$flV^2$$

and finally in the compressible fluid flow Equation 14,

$$-p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{4gR_H p_1 v_1}}\right] \tag{19}$$

Equations 17, 18, and 19 may be used to compute the flow in pipes and ducts of non-circular section and in any type of conduit not flowing full. *They should not be used when the flow is laminar.*

FLOW OF COMPRESSIBLE FLUIDS

In the flow of compressible fluids, the large density variations make impracticable the use of the Bernoulli equation, (Equation 7). In certain special cases, however, the exact equations for compressible flow may be stated. If flow occurs with no friction or other internal irreversibility, Equation 6 becomes

$$\frac{1}{2g}\,dV^2 + \frac{dp}{\rho} = 0 \tag{20}$$

If in addition the flow is adiabatic,

$$p\rho^{-k} = p_1\rho_1^{-k} \tag{21}$$

so that Equation 20 becomes

$$\frac{1}{2g}\,dV^2 + \frac{p_1^{1/k}}{\rho_1}\,\frac{dp}{p^{1/k}} = 0 \tag{22}$$

or by integration,

$$\frac{1}{2g}\,(V_2^2 - V_1^2) + \frac{k}{k-1}\,\frac{p_1}{\rho_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \tag{23}$$

This extension to compressible flow of Bernoulli's equation reduces to the more familiar form if the pressure change is small.

The ratio of specific heats, k, is used extensively in fluid dynamics; values of k for various gases are given in Table 2.

TABLE 2. RATIO OF SPECIFIC HEAT AT CONSTANT PRESSURE TO SPECIFIC HEAT AT CONSTANT VOLUME FOR COMPRESSIBLE FLUIDS

COMPRESSIBLE FLUID	RATIO $k = c_p/c_v$
Helium and other monatomic gases	1.66
Air and other diatomic gases	1.40
Ammonia and hydrogen sulfide	1.34
Carbon dioxide, methane, natural gas, superheated steam, moist steam down to a quality of 97 per cent	1.28 to 1.32
Sulfur dioxide, ethylene, acetylene	1.24 to 1.26

It is convenient in the analysis of compressible flow to introduce the velocity of propagation of pressure impulses or, more familiarly, the sonic velocity, a. For perfect gases this is given by the equation:

$$a^2 = kgp/\rho = kgRT \tag{24}$$

Accordingly, Equation 23 may be written

$$\tfrac{1}{2}(V_2^2 - V_1^2) + \frac{a_1^2}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \tag{25}$$

or, by rearrangement,

$$\frac{p_2}{p_1} = \left[1 - \frac{k-1}{a_1}\left(\frac{V_2^2 - V_1^2}{2}\right)\right]^{\frac{k}{k-1}} \tag{26}$$

which permits the calculation of the ratio of pressures at entrance and exit of the steady flow device—pipe, orifice, or nozzle. From Equations 21 and 24 it follows that

$$\frac{a_2^2}{a_1^2} = \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \tag{27}$$

so that

$$\frac{k-1}{2}(V_2^2 - V_1^2) + a_2^2 - a_1^2 = 0 \tag{28}$$

and

$$\frac{a_2^2}{a_1^2} = \frac{1 + \frac{k-1}{2}\frac{V_1^2}{a_1^2}}{1 + \frac{k-1}{2}\frac{V_2^2}{a_2^2}} \tag{29}$$

The ratio of flow velocity to sonic velocity is known as the Mach number,

$$M = V/a$$

This parameter is particularly useful in compressible flow analysis. In general, if $M \leq 0.3$ the flow may be considered to be incompressible.

In terms of the Mach number

$$\frac{a_2^2}{a_1^2} = \frac{T_2}{T_1} = \frac{1 + \frac{k-1}{2}M_1^2}{1 + \frac{k-1}{2}M_2^2} \tag{30}$$

and

$$\frac{p_2}{p_1} = \left(\frac{1 + \frac{k-1}{2}M_1^2}{1 + \frac{k-1}{2}M_2^2}\right)^{\frac{k}{k-1}} \tag{31}$$

The quantity

$$p^0 = p\left(1 + \frac{k-1}{2}M^2\right)^{\frac{k}{k-1}} \tag{32}$$

is called the stagnation pressure and gives a measure of pressure energy. For incompressible flow

$$p^0 = p + \frac{1}{2g}\rho V^2 = p + q \tag{33}$$

where

$$q = \frac{1}{2g}\rho V^2 \tag{34}$$

and is the dynamic pressure. A total head tube measures stagnation pressure directly.

From Equation 31 it follows that for frictionless, adiabatic flow

$$p_2^0 = p_1^0$$

This represents another extension of the Bernoulli equation to compressible flow. Friction will cause a loss in pressure energy.

Ideal Flow through Nozzle or Orifice

The majority of low measuring systems depend upon a correlation between pressure drop, area, and quantity of flow. The basic formulas may be stated on the assumption that the flow is frictionless and adiabatic. Designating the main stream by station 1 and flow at some measuring restriction by station 2, the flow in pounds per second is

$$w = \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \tag{35}$$

or, in terms of Mach number,

$$w = A_2 M_2 \sqrt{kgp_2\rho_2} \tag{36}$$

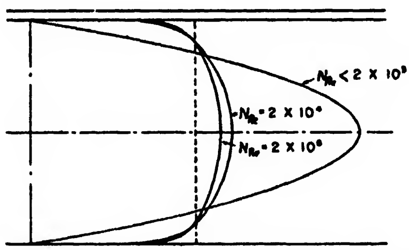

Fig. 5. Comparison of Velocity Profiles for 3 Different Reynolds Numbers but for Same Average Velocity

According to Equation 31

$$\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} = \frac{1 + \frac{k-1}{2} M_2^2}{1 + \frac{k-1}{2} M_1^2} \tag{37}$$

from which

$$M_2^2 = \frac{2}{k-1} \left(\frac{1 + \frac{k-1}{2} M_1^2}{1 - M_1^2/M_2^2} \right) \left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1 \right] \tag{38}$$

so that

$$w = A_2 \sqrt{\frac{1 + \frac{k-1}{2} M_1^2}{1 - M_1^2/M_2^2}} \sqrt{\frac{2kgp_2\rho_2}{k-1} \left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1 \right]} \tag{39}$$

If the initial velocity is sufficiently small, M_1^2 will be negligible so that

If this is computed and the figures are plotted, the curved line (partly solid and partly broken) of Fig. 6 is found. The maximum value of $\frac{p_2}{p_1}$ may be computed by differentiating w with respect to p_2 and equating the result to zero. This operation produces the formula:

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{41}$$

For air, with $k = 1.40$, $\frac{p_2}{p_1} = 0.53$.

Actually, the broken part of the curve is not attained for the flow in the nozzle. If the ratio of p_2 to p_1 is decreased from unity, the mass

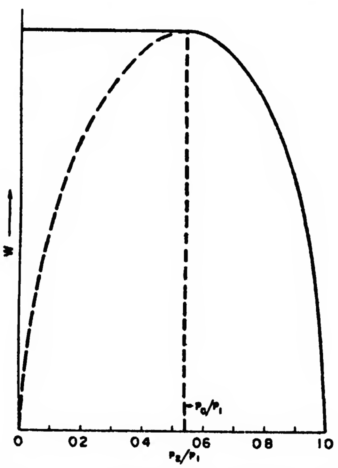

Fig. 6. Relation of Flow of Gas to Pressure Drop in a Converging Tube

rate of discharge, as well as the volume, increases from zero to a maximum, as shown by the solid section of the curve in Fig. 6; thereafter, as p_2/p_1 is decreased further, the discharge is constant, as indicated by the horizontal line. The value of p_2 at the maximum point is called the critical pressure, or p_c, and it is seen that p_c is approximately 53 per cent of p_1 when air is flowing.

To find the velocity at the critical pressure, it is assumed that the upstream velocity V_1 is so small as to be negligible. Using the subscript c to indicate conditions at the critical point, from Equation 31

$$\frac{p_c}{p_1} = \frac{1}{\left(1 + \frac{k-1}{2} M_c^2\right)^{\frac{k}{k-1}}} \tag{42}$$

or

$$M_c = \sqrt{\frac{2}{k-1}\left[\left(\frac{p_1}{p_c}\right)^{\frac{k-1}{k}} - 1\right]} \tag{43}$$

Substituting the critical pressure ratio from Equation 41 it follows that

$$M_e = 1 \tag{44}$$

or that the velocity at the throat is equal to the local sonic velocity.

In developing the working equations for orifices and nozzles, it is customary to start with the incompressible form of the flow Equation 39. In this case both M_1 and M_2 are small quantities, $\rho_1 = \rho_2$, and $(p_1 - p_2)/p_2 = \Delta p/p_2$ is small. Retaining only first order terms, it follows from Equation 36 that

$$\frac{M_1}{M_2} = \frac{A_2}{A_1} \tag{45}$$

so that

$$\sqrt{\frac{1 + \frac{k-1}{2} M_1^2}{1 - M_1^2/M_2^2}} \cong \frac{1}{\sqrt{1 - (A_2/A_1)^2}} = \frac{1}{\sqrt{1 - \beta^4}} \tag{46}$$

where $\beta = D_2/D_1$. The quantity $1/\sqrt{1 - \beta^4}$ is the velocity of approach factor as generally used, with β being the ratio of the throat or orifice diameter to the pipe diameter.

Since $\Delta p/p_2$ is small

$$\frac{2k}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right] \cong \frac{2\Delta p}{p_2} \tag{47}$$

and the mass flow is

$$w = \frac{A_2}{\sqrt{1 - \beta^4}} \sqrt{2g\rho\Delta p} \tag{48}$$

The volume flow is then

$$Q_s = A_2 \frac{1}{\sqrt{1 - \beta^4}} \sqrt{2g\Delta p/\rho} = A_2 \frac{1}{\sqrt{1 - \beta^4}} \sqrt{2gh_l} \tag{49}$$

Actual Flow Through Orifices and Nozzles

The actual rate of flow through an orifice, nozzle, or Venturi tube is rarely equal to the theoretical, and generally the actual rate is less than the theoretical. In the case of the nozzle and Venturi tube, this is due to losses from wall roughness, fluid friction, and turbulence during the expansion in the section following the throat. While wall roughness is not a factor in a sharp-edged orifice, fluid friction and turbulence are important, as is the fact that the discharge contracts to a degree variable with the ratio of outlet to inlet pressure after leaving the orifice, so that the limiting area is somewhat less than the opening in the orifice plate. Accordingly, Equation 49 must be modified by a correction factor, C. Usually, the velocity of approach factor is included with this correction factor, and, if

$$K = C \frac{1}{\sqrt{1 - \beta^4}} \tag{50}$$

then

$$Q_s = KA_2 \sqrt{2gh_l} \tag{51}$$

Multiplying by 3600 to convert from cubic feet per second to cubic feet

per hour, and converting area in square feet to diameter in inches, gives

$$Q_f = 3600\, K \frac{\pi D_2^2}{4 \times 144} \sqrt{2gh_f}$$

or

$$Q_f = 19.635\, K D_2^2 \sqrt{2gh_f} \tag{52}$$

where

Q_f = rate of flow in cubic feet per hour.
D_2 = the diameter of the orifice or nozzle throat in inches.
K = flow coefficient including correction for velocity of approach.

Equation 52 is a general equation, expressing the flow of any fluid through an orifice or nozzle. Further use of it will be made as other types of flow are discussed.

The differential loss, h_f, is in terms of feet of the fluid flowing through the orifice or nozzle. In the case of a gas flowing, where it is customary to read the differential pressure in inches of water, feet of gas must be converted to inches of water. Since dry air at 32 F and 14.7 psi absolute pressure weighs 0.0807 lb per cu ft, the weight of a cubic foot of any other kind of gas under the same conditions is 0.0807 G, where G is the specific gravity of the gas referred to air. Water weighs 62.37 lb per cu ft at 60 F. Using also the relation of 12 in. in 1 ft,

$$h_f = \frac{h_w}{12} \times \frac{62.37}{0.0807G} \tag{53}$$

in which h_w is the differential pressure in inches of water.

Also, since the gas flowing is not necessarily at 32 F and 14.7 psi, it is necessary to apply Charles' and Boyle's laws to the density of the gas and therefore

$$h_f = \frac{h_w}{12} \times \frac{62.37}{0.0807G} \times \frac{14.7}{P_f} \times \frac{T_f}{492} \tag{54}$$

in which P_f and T_f are the absolute pressure and temperature of the flowing gas. Substituting this in Equation 52, and combining the constants,

$$Q_f = 218.44 K D_2^2 \sqrt{\frac{h_w T_f}{P_f G}} \tag{55}$$

Then, to correct the value of Q_f to any other standard conditions of pressure P_b and temperature T_b, using the gas laws,

$$Q_b = Q_f \times \frac{P_f}{P_b} \times \frac{T_b}{T_f} \tag{56}$$

Equation 55 becomes

$$Q_b = 218.44 K D_2^2 \frac{T_b}{P_b} \sqrt{\frac{P_f h_w}{T_f G}} \tag{57}$$

Finally, since gases expand under the conditions of reduced pressure downstream from the orifice or nozzle, an expansion factor, Y, must be added. The final formula, then, is

$$Q_b = 218.44\,KYD_2^2\,\frac{T_b}{P_b}\sqrt{\frac{P_f h_w}{T_f G}} \tag{58}$$

In Equation 58, all the data must be observed at the time of measurement except K and Y. These must be obtained from charts, tables, or formulas, derived from or based on the results of a great many experiments, the results of which have been collected by a joint committee of the *American Gas Association* and the *American Society of Mechanical*

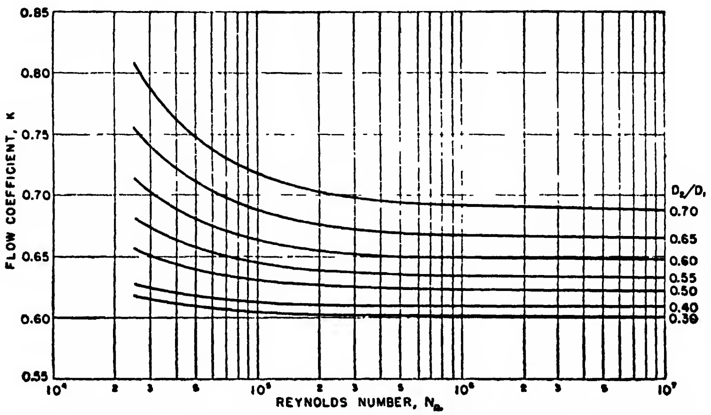

FIG. 7. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND FLANGE TAPS IN SMOOTH PIPE

NOTE: From Table 6, Bibliography [H].

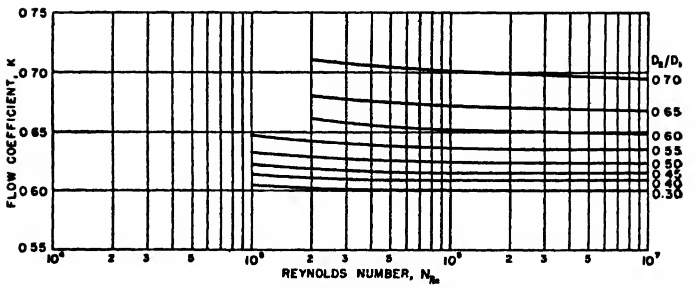

FIG. 8. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND RADIUS TAPS IN SMOOTH PIPE

NOTE: From Fig. 36d, Bibliography [K].

Engineers[2,3]. The report of the two associations gives orifice coefficients as a function of the Reynolds number and of the ratio of orifice to pipe diameter, for pipes 2 to 12 in. and 14 in. in diameter, and for four different types of pressure taps in use in the United States. The coefficients are higher for the smaller pipe sizes. This is an effect of the turbulence produced by the roughness of the pipe surface, a given roughness being relatively greater with a small pipe than with a large one.

Space does not permit presenting all the coefficient data that are available. However, if the pipe is smooth, drawn tubing, the effect of roughness is negligible, and the coefficients for the largest size of pipe apply also to smaller pipes. Figs. 7, 8, and 9 show these coefficients, N_{Re}, being the

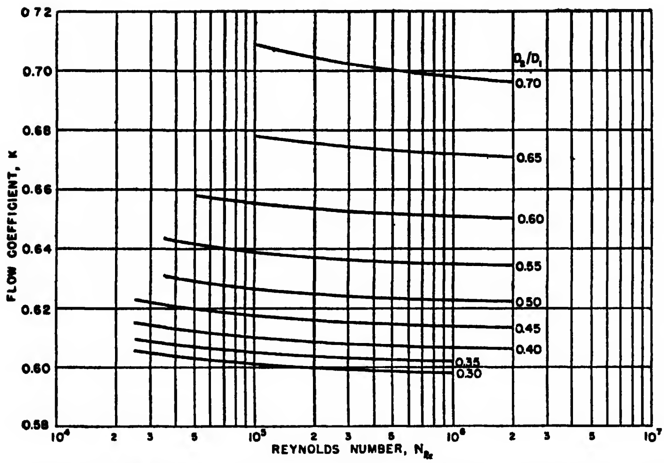

FIG. 9. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND VENA CONTRACTA TAPS, IN SMOOTH PIPE
NOTE: From Table 7, Bibliography [H].

Reynolds number referred to the diameter of the orifice or throat of the nozzle, in feet.

Pressure Taps—Location and Types

The different sets of pressure taps are called flange taps, radius taps, vena contracta taps, and pipe or full-flow taps. The relative locations of the first three of these are shown in Fig. 10, and the need for different coefficients for the different taps is indicated by the course of the change in pressure of the flowing fluid shown in the lower part of the figure. Pipe taps are located $2\frac{1}{2}$ pipe diameters upstream and 8 pipe diameters downstream, both measured from the upstream face of the orifice plate,

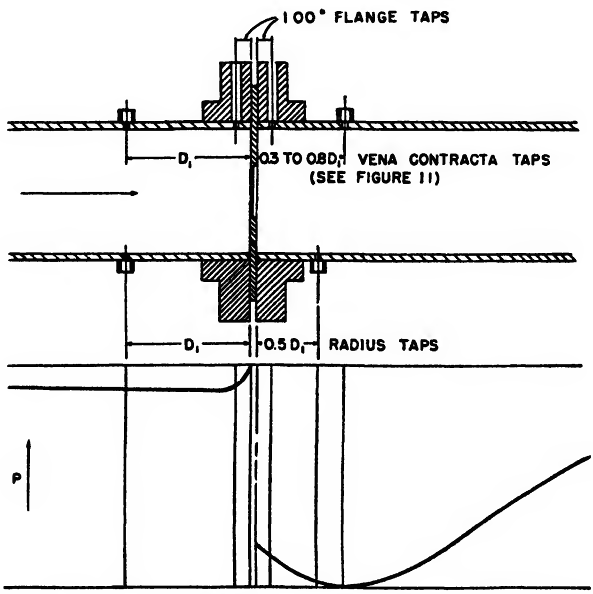

FIG. 10. RELATIVE LOCATION OF FLANGE, RADIUS AND VENA CONTRACTA TAPS

or in other words, before the orifice plate has had any effect on the flow, and after the recovery in pressure has been completed. The use of pipe or full-flow taps has been limited to the metering of natural fuel gas in certain areas. As they are not suited to use in heating and ventilating work no data for them are given in this chapter.

Still another type of pressure tap, the corner tap, is used in European practice. Pressures are taken from recesses in the flange connected to annular slits in the corners formed by the pipe wall and the orifice plate. Coefficients for these taps have been adopted by the *International Standards Association*, but are not used commercially in America.

It will be noted that the location of the downstream pressure tap of the vena contracta arrangement is variable. Vena contracta is the term applied to the minimum cross-section of the jet from the orifice, where the static pressure is at a minimum. Its location, and the location of the downstream vena contracta tap, vary with the ratio of orifice to pipe diameter, and with rate of flow, as shown in Fig. 11; the tap is generally located in accordance with the mean curve in the figure.

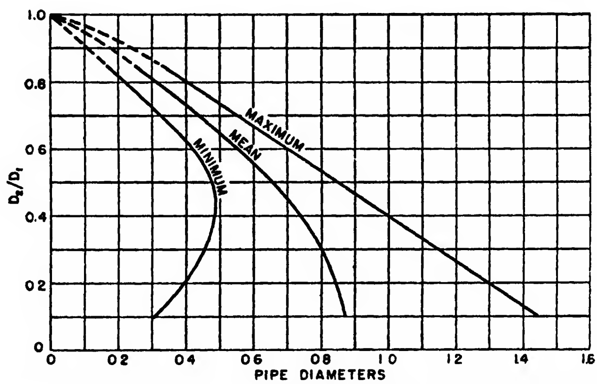

Fig. 11. Location of Vena Contracta in Relation to Ratio of Orifice to Pipe Diameter and to Rate of Flow

Expansion Factor for Gases

The expansion factor, Y, for gases (for liquids, $Y = 1$) is found from the empirical formula

$$Y = 1 - (0.41 + 0.35\beta^4)\left(\frac{p_1 - p_2}{p_1}/k\right) \tag{59}$$

This is applicable to flange, radius, and vena contracta taps.

Values of Y for air, computed from these equations, are given in Fig. 12.

Computing Orifice Discharge

With this information it is possible to compute the discharge from an orifice if the Reynolds number is known. Here an odd complication is encountered—when the value of N_{Ro} is computed, the rate of flow, which is the unknown quantity, must be used in the computation. However, it will be noted in Figs. 7, 8, and 9 that the orifice coefficient does not change greatly as N_{Ro} changes. If, then, an estimate is made of the velocity, using this in computing N_{Ro}, and if the corresponding coefficient is used in Equation 58, a value for the rate of flow will be found. Using

this velocity to compute a corrected value of N_{Re} and repeating the process, a more nearly correct value of Q_f is found. This cut-and-try method may be continued for several more cycles, but generally the first or second correction will be found sufficient.

Another method would be to use the value of K corresponding to $N_{Re} = \infty$, modifying this with a factor involving the rate of flow, determined from the temperature, and the differential and static pressures. This method is used by the *American Gas Association*[3].

STEAM FLOW MEASUREMENT

While steam may be considered as a gas, its measurement differs from that of the usual gases because of a number of factors. Equation 52 serves as the starting point. Since it is usual to measure the differential

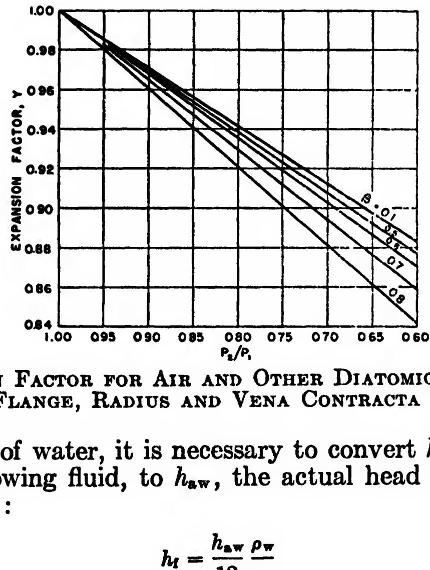

FIG. 12. EXPANSION FACTOR FOR AIR AND OTHER DIATOMIC GASES APPLICABLE TO FLANGE, RADIUS AND VENA CONTRACTA TAPS

pressure in inches of water, it is necessary to convert h_f, the head in feet in terms of the flowing fluid, to h_{aw}, the actual head in inches of water, using the equation:

$$h_f = \frac{h_{aw}}{12}\frac{\rho_w}{\rho} \tag{60}$$

where

ρ_w = the density of water at 60 F (62.37 lb per cu ft).
ρ = the density of the flowing fluid.

Substituting Equation 60 in Equation 52 gives

$$Q_f = 359.15\,KD_2^2\sqrt{\frac{h_{aw}}{\rho}} \tag{61}$$

In steam measurement, a constant head of water is maintained over each leg of the manometer by means of condensing chambers, in order to keep the heat of the steam away from the meter. As the mercury level fluctuates, the difference in head as recorded on the chart, therefore, is not that due to mercury alone, but to mercury minus an equivalent head of water. To correct for this, the equation:

$$h_{aw} = h_w\frac{12.557}{13.557} \tag{62}$$

is applied to Equation 61. The denominator in Equation 62 is the specific

gravity of mercury; and the numerator is the difference in specific gravity between mercury and water. Hence, Equation 61 becomes

$$Q_f = 345.65\, K D_2^2 \sqrt{\frac{h_w}{\rho}} \tag{63}$$

Steam is commonly measured in terms of weight, and since

$$w_h = \rho Q f \tag{64}$$

where

w_h = the rate of flow in pounds per hour.

$$w_h = 345.65\, K D_2^2 \sqrt{h_w \rho} \tag{65}$$

The expansion factor Y, and the factor P, correcting for the expansion of the orifice plate with the temperature, must then be applied, so that

$$w_h = 345.65\, KYPD_2^2 \sqrt{h_w \rho} \tag{66}$$

which is the *final form of the equation for the flow of steam through orifices.* Values of K may be obtained from Figs. 7, 8, and 9, according to the

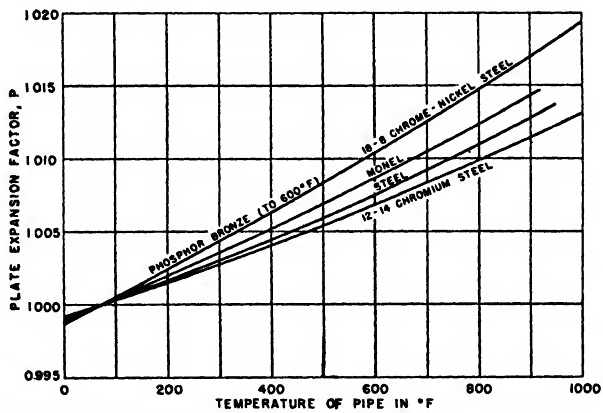

FIG. 13. VARIATION OF ORIFICE PLATE EXPANSION FACTOR, P, WITH TEMPERATURE AND MATERIAL

pressure taps used. Y may be computed from Equation 59. Fig. 13 gives values of the correction factor P, according to the temperature and the material of the orifice plate. Values of the density, ρ, may be obtained from steam tables, such as Keenan and Keyes[4], which are widely used.

METERING LIQUIDS

Orifices are also used for metering liquids, and Equation 52 serves again as a starting point for developing the working formula. Again, it is necessary to convert h_f to h_{aw}, the actual head in inches of water, by substituting Equation 60. It is also necessary to correct for the weight of the fluid above the manometer, and since this may be other than water, it is better to use an equation of more general form than Equation 62:

$$h_{aw} = h_w \, \frac{13.557 - \dfrac{\rho_f}{\rho_w}}{13.557} \tag{67}$$

where

ρ_l = the density of the fluid over the mercury in the manometer.
ρ_w = the density of water at 60 F.

Substituting Equations 60 and 67 in Equation 52 gives

$$Q_l = 44.764\, K D_2^2 \sqrt{h_w \left(\frac{1}{\rho_l} - 0.00118\right)} \tag{68}$$

Then, since liquids are generally measured in gallons instead of cubic feet, and since there are 7.4805 gal in 1 cu ft,

$$Q_w = 334.86\, K D_2^2 \sqrt{h_w \left(\frac{1}{\rho_l} - 0.00118\right)} \tag{69}$$

in which Q_w is the discharge or rate of flow, in gallons per hour.

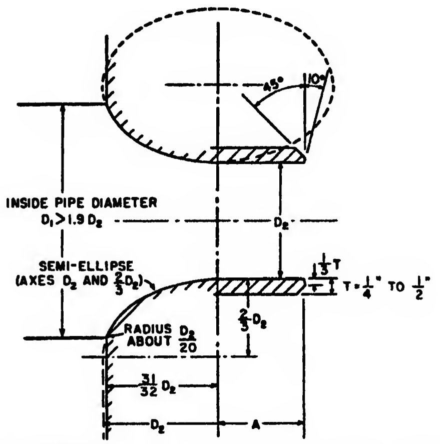

FIG. 14. SHAPE OF *ASME* LONG RADIUS NOZZLE WHEN RATIO OF THROAT TO PIPE DIAMETER IS 0.53 OR LESS

Since liquids, for practical purposes, are incompressible, no expansion factor is necessary. If circumstances demand, the factor *P* for the expansion of the orifice may be applied. Also, if it is necessary to correct the volumetric discharge to a base temperature, application of the known expansion characteristics of the liquid will enable the conversion to be made. Values of *K* again are obtainable from Figs. 7, 8, and 9, according to the type of pressure tap.

NOZZLE COEFFICIENTS AND EXPANSION FACTORS

Nozzles differ from orifices in that the flow is guided to the throat in such a way that contraction of the jet is suppressed, or, in other words, there is no vena contracta. Because of this fact, the coefficients are different from those of orifices, and are very close to unity before the velocity of approach factor is added. Also, the expansion factor may be deduced rationally, rather than empirically, as with orifices.

Two shapes of nozzles that have been under investigation by the *A.S.M.E.* for some time are shown in Figs. 14 and 15. They are referred

to as long-radius nozzles. Their contour is that of a semi-ellipse, and the contracting portion is followed by a cylindrical section of the same area as the throat. The shape shown in Fig. 14 is designed for use with ratios of throat to pipe diameter of 0.53 or less, that of Fig. 15 for ratios of 0.4 to 0.7. The most usual location of pressure taps is 1 pipe diameter upstream and ½ diameter downstream, both measured from the plane of the nozzle inlet. In addition, the *International Standards Association* has adopted still another shape of nozzle, which has a somewhat sharper approach than the *A.S.M.E.* nozzles, and which uses corner taps. Very little use of this nozzle has been made in this country.

The formulas already given for orifices apply equally to nozzles except for discharge coefficients, and for the expansion factor, when it is applied. Discharge coefficients for nozzles, as for orifices, vary with pipe size; they may either increase or decrease with decreasing size of pipes depending on the sharpness of the approach curvature of the nozzle. For the *A.S.M.E.* nozzles, they tend to decrease. Generally speaking, too, the

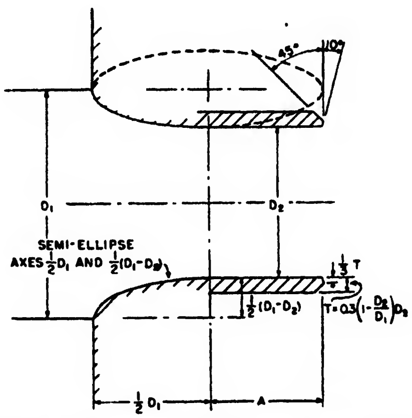

FIG. 15. SHAPE OF *ASME* LONG RADIUS NOZZLE WHEN RATIO OF THROAT TO PIPE DIAMETER IS 0.4 TO 0.7

coefficient for a given nozzle shape is higher if the finish of the surface is smoother.

Discharge coefficients, C, for pipes 2, 6, and 10 in. in diameter are given in Figs. 16, 17, and 18, as correlated by Bean, Beitler and Sprenkle[5], as functions of the diameter ratio β and the Reynolds number N_{Re} (Equation 70) referred to the diameter of the throat in feet.

$$N_{Re} = \frac{D_2 V_2 \rho_1}{\mu_1} \tag{70}$$

Coefficients from these curves must be multiplied by $\frac{1}{\sqrt{1-\beta^4}}$, the velocity of approach factor, in accordance with Equation 50, to obtain the value of K to use in the various equations.

The expansion factor for nozzles, designated by φ, is obtained from a rational formula, as already noted:

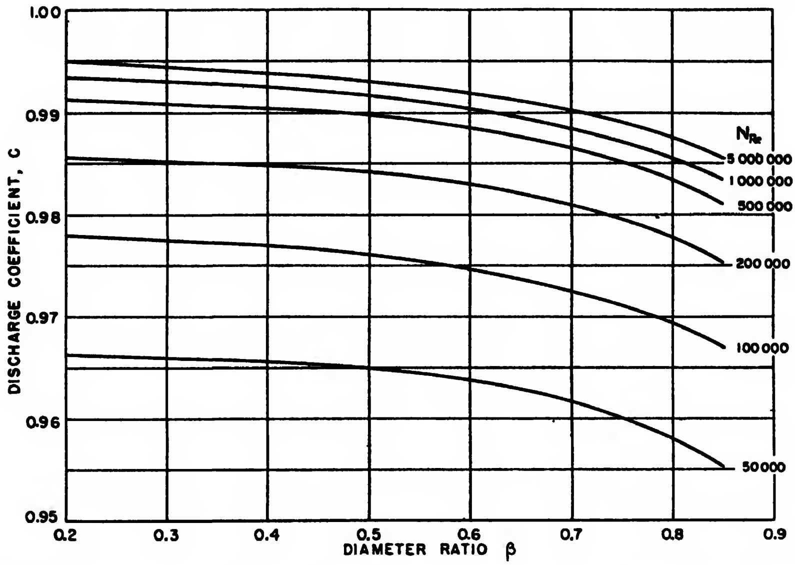

FIG. 16. RELATION OF NOZZLE DISCHARGE COEFFICIENT, C, FOR 2-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

$$\varphi = \sqrt{\left(\frac{p_2}{p_1}\right)^{2/k}\left(\frac{k}{k-1}\right)\left(\frac{1-(p_2/p_1)^{k-1/k}}{1-p_2/p_1}\right)\left(\frac{1-\beta^4}{1-\beta^4(p_2/p_1)^{2/k}}\right)} \tag{71}$$

This formula is plotted for $k = 1.40$ (air and other diatomic gases) and 1.30 (steam, carbon dioxide, natural gas) in Figs. 19 and 20, respectively.

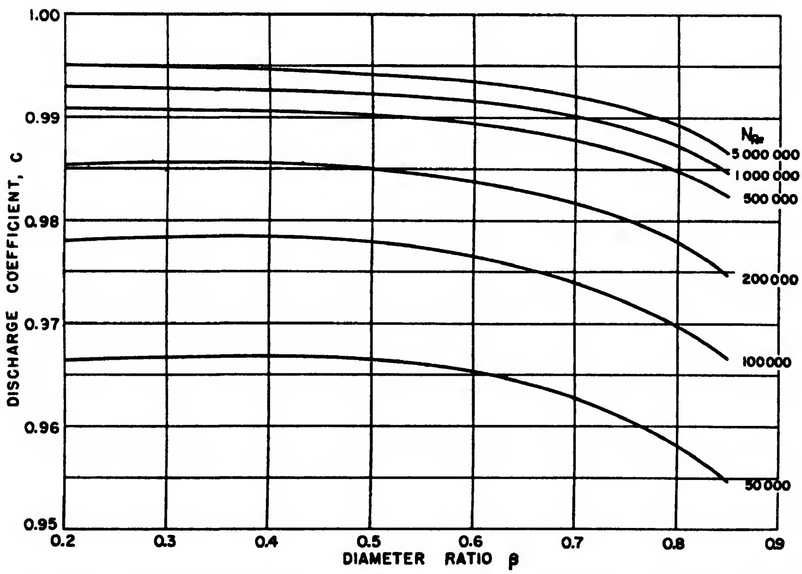

FIG. 17. RELATION OF NOZZLE DISCHARGE COEFFICIENT, C, FOR 6-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

FLOW MEASUREMENT BY PITOT TUBE

There remains one other head type meter useful in ventilating work, the Pitot tube, named for the Frenchman who discovered the principle. The Pitot tube is essentially a bent tube with its open end pointed upstream, combined with another tube with its end pointed crosswise to the flow or downstream, or connected to openings crosswise to the flow. Used with flowing liquid, the liquid will rise in each tube, but higher in the one pointed upstream. Used with a flowing gas, and the two tubes connected by a U-tube containing water, the liquid level in the U-tube will be displaced, with the lower level on the side connected to the tube pointed upstream. The tube directed upstream receives the impact pressure, which is the sum of the static and kinetic pressures, while the tube directed

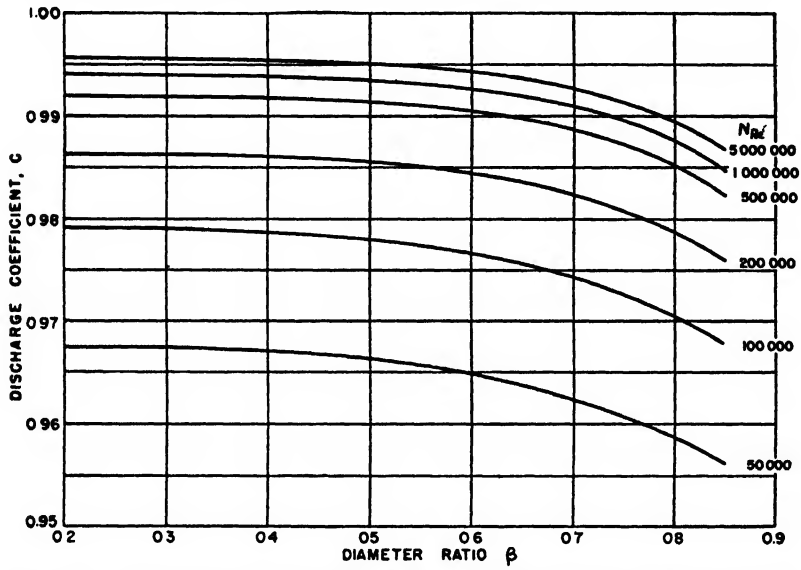

FIG. 18. RELATION OF NOZZLE DISCHARGE COEFFICIENT, C, FOR 10-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

crosswise receives only the static pressure; the difference between the two, as read on the separate tubes or on the U-tube is, of course, the kinetic pressure. The velocity is expressed as

$$V = \sqrt{2gh_t} \tag{72}$$

Application of Equation 60 serves to make the formula general, assuming that water is used in the manometer connecting the two tubes. Using this equation, and multiplying by 60 to convert feet per second to feet per minute,

$$V_m = 1096.5 \sqrt{\frac{h_{aw}}{\rho}} \tag{73}$$

in which V_m is the rate of flow in feet per minute.

It is often difficult to obtain the exact static pressure. In the usual construction of Pitot tubes, the static pressure openings are downstream from the impact pressure opening, and turbulence induced by the nose may affect the static pressure reading. If the static pressure openings point

downstream in any degree, a suction effect is produced to falsify the reading. In instruments having the static pressure opening pointed downstream, the coefficient may be as low as 0.86. Consequently, for accurate work, Pitot tubes should be calibrated, and the pertinent coefficient should be applied to Equation 72. In a sense, this coefficient is advantageous, since it results in a higher differential reading, which, in turn, enables more accurate readings at low flows.

In using Pitot tubes, it is generally necessary to make a traverse of the pipe or duct to determine the course of the velocity pattern. In a pipe, for instance, one of the profiles shown in Fig. 5 would be obtained. Near a valve or fitting, however, the profile might be quite distorted, a fact which would be revealed by the traverse. If the pipe or duct is divided

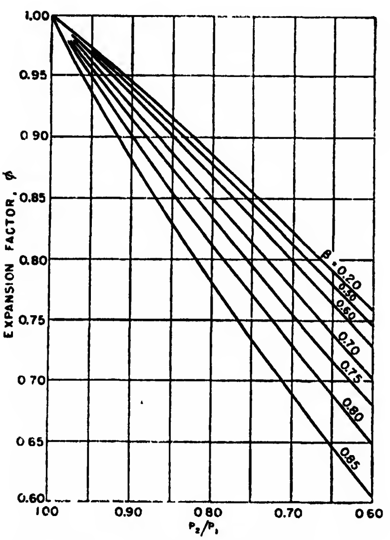

FIG. 19. RELATION OF EXPANSION FACTOR, φ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR AIR AND OTHER DIATOMIC GASES.

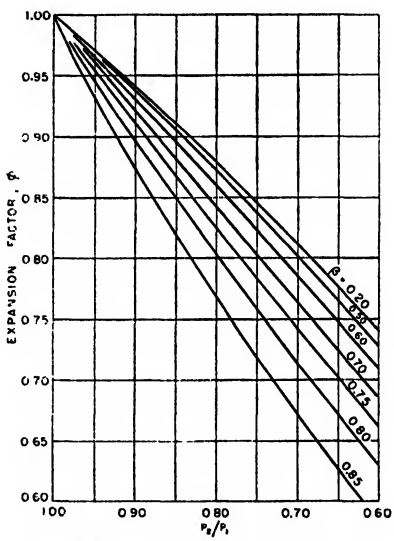

FIG. 20. RELATION OF EXPANSION FACTOR, φ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR STEAM, CARBON DIOXIDE AND NATURAL GAS.

into equal areas, and a determination of h_{aw} is made for each, the average velocity would be obtained by using the average of the square roots of h_{aw} in Equation 73.

INSTALLATION OF NOZZLES AND ORIFICES

A final note should be made of the installation of orifices and nozzles. Generally speaking, the orifice or nozzle, together with a holding arrangement, including pressure taps, is available from the manufacturer. In making the installation, the user should make certain that the flow approaching the nozzle or orifice is steady and evenly distributed, *i.e.*, with velocity profiles similar to those shown in Fig. 5. Fittings and valves, which tend to direct the flow to one side and which in some cases cause it to rotate as it advances, must be far enough upstream from the orifice or nozzle to permit the disturbed stream to straighten out to the even form before reaching the meter. Minimum conditions in the installation for avoiding trouble from fittings and valves are shown in Fig. 21. If necessary, straightening vanes may be used upstream from the orifice or nozzle at a distance of not less than 8 pipe diameters.

VARIABLE AREA FLOW METERS

In the routine measurements of many liquids and gases, satisfactory results may be obtained by the use of variable area flow meters, frequently called rotameters. In their most common form these devices consist essentially of a float which is free to move vertically in a transparent tapered tube. The fluid to be metered enters at the bottom, narrow end of the tube and moves upward, passing at some point through the annulus formed between the float and inside wall of the tube. At any particular rate of flow the float assumes a definite position in the tube, its location being indicated by means of a calibrated scale on the tube.

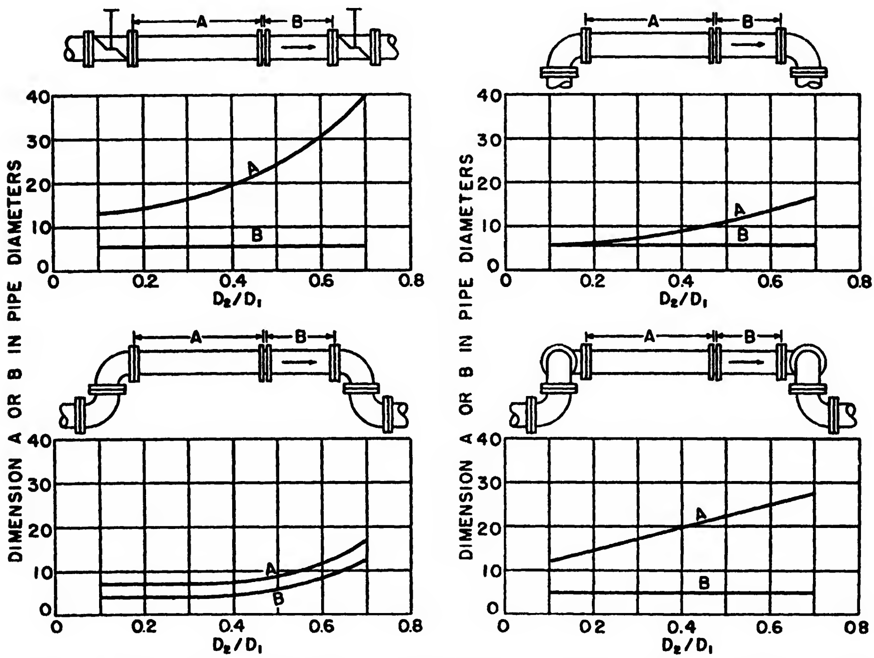

FIG. 21. MINIMUM CONDITIONS TO BE OBSERVED WHEN INSTALLING ORIFICES AND NOZZLES BETWEEN FITTINGS AND VALVES

The position of the float is established by a balance between the fluid pressure forces across the annulus and the weight of the float itself. The buoyant force which must support the float, $v_f(\rho_f - \rho)$, is balanced by the pressure difference acting on the cross section area of the float, $A_f\Delta p$, where ρ_f, A_f, v_f, are respectively the float density, float cross section area, and float volume. Accordingly the difference in head across the annulus is given by

$$h_f = \frac{\Delta p}{\rho} = \frac{v_f(\rho_f - \rho)}{A_f\rho} \tag{74}$$

The volume flow follows from equation (51) as

$$Q_s = KA_2\sqrt{2gv_f(\rho_f - \rho)/\rho A_f} \tag{75}$$

and the mass flow as

$$w = \rho Q_s = KA_2\sqrt{2gv_f(\rho_f - \rho)\cdot\rho/A_f} \tag{76}$$

The flow for any selected fluid is accordingly very nearly proportional to the area so that a convenient calibration of the tube may be obtained. The behavior of the flow coefficient, K, has been investigated[6] and the action of the flow meter as just outlined, experimentally confirmed. The flow coefficient variation for any float must be known in order to use the meter for different fluids. Some developments have been carried out in the design of the float to reduce the variation of the flow coefficient with Reynolds number, and also on float materials to reduce the dependence of mass flow calibration on fluid density.

This type of flow meter is usually furnished in standard sizes calibrated for specific fluids by the manufacturer. The compactness, reliability, and ease of installation are particularly advantageous when many measurements of essentially the same type are to be made.

LETTER SYMBOLS USED IN CHAPTER 4

β = ratio, throat or orifice diameter to pipe diameter.
μ = absolute viscosity in pounds per foot second.
μ/ρ = kinematic viscosity in square feet per second.
ρ = density of flowing fluid in pounds per cubic foot.
ρ_w = density of water at 60 F, (62.37 lb per cubic foot).
ρ_f = density of fluid over mercury in a manometer.
φ = expansion factor for nozzles.
a = velocity of sound in feet per second.
A = cross-sectional area of flow, in square feet.
C = correction factor (coefficient of discharge) for flow through orifice, nozzle or Venturi.
c_p = specific heat of gas at constant pressure.
c_v = specific heat of gas at constant volume.
D = diameter of fluid stream in feet.
d = internal diameter of pipe in feet.
e = absolute roughness of pipe surface, in feet.
f = dimensionless friction coefficient.
g = gravitational acceleration in feet per (second) (second).
g_o = gravitational conversion factor = 32.174 (pounds mass per pound force) × feet per (second) (second).
G = specific gravity of gas referred to air.
h = enthalpy, Btu per pound of fluid.
h_{aw} = loss of head in inches of water.
h_f = loss of head in feet of fluid.
h_t = total head in feet of fluid.
h_w = differential pressure in inches of water.
J = mechanical equivalent of heat = 778 foot pounds per Btu.
K = flow coefficient (correction factor), including velocity of approach correction factor, for flow through orifice, nozzle or Venturi.
k = ratio of specific heat at constant pressure to specific heat at constant volume.
L = perpendicular distance from axis of pipe in feet.
l = length of pipe in feet.
M = Mach number.
N_{Re} = Reynolds number.
P = correction factor for expansion of orifice plate with temperature.
p = pressure in pounds per square foot.
p^0 = stagnation pressure.
p_c = critical pressure.
P_b = standard pressure to which correction is to be made, pounds per square inch, absolute.
P_f = pressure of gas flowing, pounds per square inch, absolute.
Q_b = rate of flow in cubic feet per hour under standard conditions of pressure and temperature.

Q_f = rate of flow in cubic feet per hour.
Q_s = discharge rate in cubic feet per second.
Q_w = rate of flow in gallons per hour.
q = heat transferred to the fluid per pound of fluid flowing.
R = gas constant.
R_H = hydraulic radius = ratio of area of cross-section to wetted perimeter.
r = radius of pipe in feet.
s = entropy of fluid in Btu per (pound) (Fahrenheit degree)
T = temperature, Fahrenheit degrees, absolute.
T_L = standard temperature to which correction is to be made, Fahrenheit degrees absolute.
T_f = temperature of gas flowing, Fahrenheit degrees, absolute.
u = internal energy, Btu per pound of fluid.
V = velocity in feet per second.
V_c = critical velocity, feet per second.
V_{so} = velocity of sound, feet per second.
V_m = velocity in feet per minute.
v = specific volume, cubic feet per pound.
W = mechanical work in foot pounds per pound of fluid flowing.
w = weight of gas flowing, pounds per second.
w_h = weight of gas flowing, pounds per hour.
Y = expansion factor—correcting for expansion of gas under reduced downstream pressure.
z = elevation above some arbitrary datum, in feet.

REFERENCES

[1] Friction Factors for Pipe Flow, by Lewis F. Moody (*A.S.M.E. Transactions*, *66*, 1944, 671–678; Discussion, idem. *66*, 1944, 678–684); also, An Approximate Formula for Pipe Friction Factors (*Mechanical Engineering*, 69, 1947, 1005–1006).

[2] History of Orifice Meters and the Calibration, Construction, and Operation of Orifices for Metering. Report of the Joint *A.G.A.—A.S.M.E.* Committee on Orifice Coefficients (*American Society of Mechanical Engineers*, 1935).

[3] Gas Measurement Committee Report No. 2, Natural Gas Department (*American Gas Association*, 1935).

[4] Thermodynamic Properties of Steam, by Joseph H. Keenan and Frederick G. Keyes (John Wiley & Sons, Inc., 1936).

[5] Discharge Coefficients of Long Radius Flow Nozzles When Used with Pipe Wall Pressure Taps, by H. S. Bean, S. R. Beitler, and R. E. Sprenkle (*A.S.M.E. Transactions*, *63*, 1941, 439–442; Discussion, idem. *63*, 1941, 442–445).

[6] The Flow Mechanism and Performance of the Rotameter, by E. M. Schoenborn, Jr. and A. P. Colburn (*Institute of Chemical Engineers, Transactions*, 35, 1939, 359–381).

BIBLIOGRAPHY

[A] Principles of Thermodynamics, by G. A. Goodenough (Henry Holt & Co.).

[B] Principles of Engineering Thermodynamics, by Paul J. Kiefer and Milton C. Stuart (John Wiley & Sons, Inc., 1944).

[C] Fluid Mechanics, by Russell A. Dodge and Milton J. Thompson (McGraw-Hill Book Co., 1937).

[D] Fluid Mechanics, by R. C. Binder (Prentice-Hall, Inc., 1943).

[E] The Physics of Solids and Fluids, by P. P. Ewald, H. Pöschl and L. Prandtl (Blackie, 1936).

[F] A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler, Hydraulic Paper HYD-55-2 (*A.S.M.E. Transactions 55*, No. 10, 7–22, 1933; Discussion, idem., *55*, No. 10, 23–32, 1933).

[G] The Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering 55*, 1933, 497–501, 515).

[H] Fluid Meters, Their Theory and Application (*American Society of Mechanical Engineers*, 4th edition, 1937).

[I] Fluid Meters, Their Selection and Installation (*American Society of Mechanical Engineers*, 1933).

[J] The Orifice Meter for Measurement of Flow of Gases and Liquids, by Allen D. MacLean (Pittsburgh Equitable Meter Co., 1938).

[K] Flow Measurement (*American Society of Mechanical Engineers*, 1940).

[L] Pitot Tube Practice, by Edward S. Cole (*A.S.M.E. Transactions 57*, 1935, 281–294; Discussion, idem. *58*, 1936, 146–156).

[M] Pitot Tubes in Large Pipes, by Edward S. Cole and E. Shaw Cole (*A.S.M.E. Transactions*, *61*, 1939, 465–473; Discussion, idem. *61*, 1939, 473–475).

[N] Investigation of Errors of Pitot Tubes, by C. W. Hubbard (*A.S.M.E. Transactions*, *61*, 477–497; Discussion, idem. *61*, 1939, 497–506).

[O] Piping Arrangements for Acceptable Flowmeter Accuracy, by R. E. Sprenkle (*A.S.M.E. Transactions*, *67*, 345–357, 1945; Discussion, idem. *67*, 357–360, 1945).

CHAPTER 5

FUNDAMENTALS OF HEAT TRANSFER

Conduction, Convection, Radiation, Combined Convection and Radiation, Heat-Flow Resistance, Practical Heat Transfer Problems, Unit Conductances for Convection Flow Systems, Radiation Factors or Emissivities, Solutions for Steady-State Conduction Problems

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HEAT is that form of energy that is transferred by virtue of an existing temperature difference. The temperature difference is the potential which causes the transfer, the latter in turn being resisted by the thermal properties of the material combined in a single term known as the resistance. Energy exchange associated with evaporation, condensation, etc. is treated elsewhere such as in the section on *cooling tower design* in Chapter 37. The objectives of this chapter are to:

1. Describe the mechanisms and present the rate equations for the different modes of heat transfer.

2. Illustrate the application of the basic concepts to steady-state problems (temperature independent of time or a cyclic variable thereof) by means of several typical solutions of heat transfer systems.

Further applications to specific systems will be found throughout THE GUIDE.

CONDUCTION, CONVECTION AND RADIATION

Thermal conduction is the term applied to the mechanism of heat transfer whereby the molecules of higher kinetic energy transmit part of their energy to adjacent molecules of lower kinetic energy by direct molecular action. Since the temperature is proportional to the average kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The motion of the molecules is random; there is no net material flow associated with the conduction mechanism. In the case of flowing fluids, thermal conduction is significant in the region very close to a solid boundary or wall, for in this region the flow is laminar, parallel with the wall surface, and there are practically no cross currents in the direction of the heat transfer across the solid fluid boundary. In solid bodies the significant mechanism of heat transfer is always thermal conduction.

Contrasted to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion[1] in addition to conduction. This is shown schematically in Fig. 1 which exhibits transfer from a pipe wall at surface temperature t_s to a colder fluid at a bulk temperature t_f. (Bulk temperature is that which would be attained if the fluid stream were drawn off at a certain section and mixed. It is therefore slightly higher than the lowest temperature in the stream). In the *laminar* sublayer, immediately adjacent to the wall, the heat transfer occurs by thermal conduction; in the *transition* region, which is called the buffer layer, eddy mixing as well as conduction effects are significant; in the *eddy* or *turbulent* region the major fraction of the transfer occurs by eddy mixing.

In most commercial equipment the main body of the fluid is in turbulent flow, and the laminar film exists at the solid walls only, as shown in

Fig. 1. But in cases of low-velocity flow in small tubes, or with viscous liquids such as heavy oil (low Reynolds numbers), the entire flow may be laminar. In these latter cases there is no transition or eddy region.

When the fluid currents are produced by sources external to the heat transfer region, as for example by a pump, the described solid to fluid heat transfer is termed *forced convection*. In contrast, if the fluid currents are generated internally, as a result of non-homogeneous densities arising from the temperature variations, the heat transfer is termed *free convection*.

In the conduction and convection mechanisms heat is transferred as internal energy, *i.e.*, the random molecular kinetic energy associated with

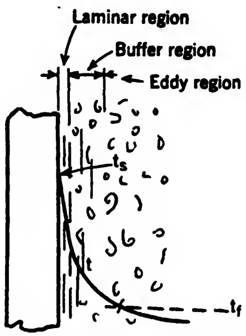

FIG. 1. THERMAL CONVECTION CONDITIONS

the material temperature. For *radiant heat transfer*, however, a change in energy form takes place from internal energy at the source to electromagnetic energy for transmission, then back to internal energy at the receiver.

The rate of heat transfer, corresponding to the three transfer mechanisms previously described, may be expressed by three rate equations.

TABLE 1. APPROXIMATE UNIT THERMAL CONDUCTIVITIES[a]
Conductivity, k = Btu per (hr) (sq ft) (F deg per in.)

MATERIAL	k	MATERIAL	k
Air	0.168	Lead	240.0
Aluminum	1416.0	Nickel	408.0
Brass (70 — 30)	720.0	Soil	2.4—12.0
Cast-Iron	336.0	Steel, mild	312.0
Copper	2640.0	Water, liquid	4.08
Glass	3.6—7.32		

[a] Thermal conductivities depend to some extent on temperature. The above magnitudes are approximate only. Refer to Heat Transmission, 2nd edition, by W. H. McAdams (McGraw-Hill Co., 1942) for additional values.

These are similar to Ohm's Law for electrical flow, the current flow through a resistance being proportional to the potential difference.

Thermal Conduction Equation

Equation 1 states symbolically that the thermal conduction per unit transfer area normal to the flow, $(dq)/(dA)$, Btu per (hour) (square foot), is proportional to the temperature gradient $(dt)/(dL)$, Fahrenheit degrees per foot. The proportionality factor is termed the *thermal conductivity*,

k, Btu per (hour) (square foot) (Fahrenheit degree per foot of thickness).

$$\frac{dq}{dA} = -k\frac{dt}{dL} \tag{1}$$

The minus sign on the right side of the equation is introduced to indicate positive transfer in the direction of decreasing temperature. Fig. 2 shows the physical significance of indicated quantities.

It should be emphasized that the thermal conductivity used should be expressed in consistent units; either using the inch or foot throughout.

Expressions of conductivity used in the heating field are usually inconsistent in this sense, in that it is customary to refer to the conductivity *per square foot* but for one inch of thickness. This custom has been adopted for the reason that wall thicknesses are usually expressed in inches, whereas if expressed in feet, decimal or fractional thicknesses would result. When dealing with flat walls no complication is involved in using the inconsistent

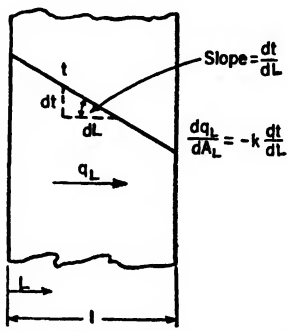

FIG. 2. THERMAL CONDUCTION IN A FLAT SLAB

expression of conductivity. However, when curved or spherical walls are considered, considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in *Btu per (hour) (square foot) (Fahrenheit degrees per one foot thickness)* are used throughout. *Conductivity values obtained from Chapter 6 or Table 1 in this chapter, must therefore be converted for use in the calculations of this chapter by dividing by 12.* As an example, the conductivity of brick listed as 5.0 in Table 2 of Chapter 6, becomes 0.42 when used in the calculations of this chapter. Also, it should be emphasized that in order to make the calculations and applications consistent in this chapter, *all dimensions of thickness must be expressed in feet.*

Thermal Convection Equation

$$\frac{dq}{dA} = h_c(t_s - t_f) \tag{2}$$

This rate equation states that the thermal convection per unit transfer area $(dq)/(dA)$, Btu per (hour) (square foot) is proportional to the temperature difference $(t_s - t_f)$ which is the temperature of the surface less that of the fluid. The particular fluid temperature to use for a given system will be noted under the discussion of that system. The proportionality factor is termed the *unit convection conductance* (sometimes called the film coefficient for convection), h_c, Btu per (hour) (square foot) (Fahrenheit degree). These convection conditions are illustrated in Fig. 1.

Table 2. Approximate Unit Conductances for Thermal Convection for Several Flow Systems[a]

Expressed in Convenient Empirical Form

Case	System	Unit Conductance Equation[b]
	Forced Convection	
1.	Longitudinal flow in cylinders, turbulent region. Fluid being heated[c].	$\frac{h_c D}{k} = 0.0225 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$ For $\left(\frac{DG}{\mu}\right) > 3000$
2.	For longitudinal air flow in cylinders case 1 reduces to[c].	$h_c = 0.0036\, G^{0.8}/D^{0.2}$ For $\left(\frac{DG}{\mu}\right) > 3000$
3.	For longitudinal water flow in cylinders case 1 reduces to[c].	$h_c = 0.00486\,(1 + 0.01t)\, \frac{G^{0.8}}{D^{0.2}}$ For $\left(\frac{DG}{\mu}\right) > 3000$
4.	Air flow normal to a single right circular cylinder.	$h_c = 0.45 \left(\frac{k}{D}\right) + 0.178\, G^{0.56} \left(\frac{k}{D}\right)^{0.44}$
5.	Air flow over staggered pipe banks.	$h_c = 0.061 \left(\frac{k}{D}\right)^{0.31} G^{0.69}$
6.	Air flow over single spheres.	$h_c = 0.040\, \frac{G^{0.52}}{D^{0.48}}$ $0 < t < 250$ F
7.	Air flow over plane surfaces.	$h_c = 1 + 0.22\, V_s$ For $V_s < 16$ fps or $h_c = 0.53\, V_s^{0.8}$ 16 fps $< V_s <$ 100 fps
8.	Air flow normal to finned cylinders.	$h_c = 6.2 \left(\frac{G}{3600}\right)^{0.8} \frac{s^{0.33}}{D^{0.52}}$ $0 < t < 250$ F
	Free Convection[d]	
9.	Single horizontal right circular cylinder in air.	$h_c = 0.23 \left(\frac{t_1 - t_2}{D}\right)^{0.25}$
10.	Vertical surfaces in air.	$h_c = 0.3\,(t_1 - t_2)^{0.25}$
11.	Top surface of horizontal plates to air.	$h_c = 0.4\,(t_1 - t_2)^{0.25}$
12.	Bottom surface of horizontal plates to air.	$h_c = 0.2\,(t_1 - t_2)^{0.25}$

[a] Heat Transmission, by W. H. McAdams.

[b] Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.

[c] These expressions are applicable to longitudinal flow in other than right circular cylinders provided the hydraulic radius is employed as the conduit dimension parameter. For non-circular cross-sections $D = 4\,R_H$.

[d] For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates increases towards 0.33. The following equations employing an exponent equal to 0.25 are applicable in the intermediate range.

The heat transmission by free or natural convection for objects surrounded by air can be conveniently expressed as in Equation 2a:

$$q_c = C\left(\frac{1}{D}\right)^{0.2}\left(\frac{1}{T_{av}}\right)^{0.181}(t_s - t_i)^{1.27} \qquad (2a)$$

where

q_c = heat transmission by convection, Btu per (square foot) (hour).
C = a constant depending upon the shape of the surface.
D = diameter of pipe or circular duct or height of vertical wall, inches. (Effect of diameter or height becomes constant at 24 in.)
T_{av} = average of wall surface and surrounding air temperature, Fahrenheit degrees absolute.
$t_s - t_i$ = temperature excess between wall surface and surrounding air, Fahrenheit degrees.

For horizontal cylinders, the value of $C = 1.016$ has been well established by various investigations. For vertical plates, the value of $C = 1.394$ has been fairly well established. Suggested values[2] of C for horizontal plates warmer than the surrounding air are 1.79 when facing upward and 0.89 when facing downward.

Problems in either forced convection or natural convection may be solved by the simple first-power equation if the convection coefficient is expressed as a unit conductance:

$$q_c = h_c\, A\,(t_1 - t_2) \qquad (2b)$$

where

q_c = heat transmission by convection, Btu per hour.
A = surface area, square feet.
$t_1 - t_2$ = temperature difference between the surface and the fluid Fahrenheit degrees.
h_c = unit conductance, from Table 2, Btu per (square foot) (hour) (Fahrenheit degree temperature difference.)

Thermal Radiation Equation

The relation shown in Equation 3 is usually applicable to systems in

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NOMENCLATURE AND DIMENSIONS FOR TABLE 2

c_p = fluid unit heat capacity at constant pressure, Btu per (pound) (Fahrenheit degree).
D = cylinder diameter, feet.
G = $3600\,V_s\rho$ = fluid mass velocity, pounds per (hour) (square foot of flow cross-section).
ρ = density, pounds per cubic foot.
h_c = unit conductance for thermal convection, Btu per (hour) (square foot) (Farhrenheit degree).
k = unit thermal conductivity of the fluid, Btu per (hour) (square foot) (Fahrenheit degree per one foot thickness).
R_H = hydraulic radius of the flow cross-section = flow cross-section area per wetted perimeter, feet.
s = fin spacing, feet.
t = average fluid film temperature, Fahrenheit degree.
$t_1 - t_2$ = temperature difference surface to main fluid, Fahrenheit degree.
V_s = fluid velocity, feet per second.
μ = fluid viscosity, pounds per (hour) (foot) = viscosity in centipoises $\times$ 2.42.

which radiant exchange takes place between the surfaces of solids, as sche-

$$q_r = \sigma A_1 F_A F_E \, (T_1^4 - T_2^4) \tag{3}$$

matically shown in Fig. 3. Gaseous and luminous radiation are not considered in this discussion. Equation 3 states that the net radiation per unit transfer area of surface 1, q_r/A Btu per (hour) (square foot), which *sees* surface 2 through a non-absorbing medium, is proportional to the difference of the fourth powers of the absolute surface temperatures $(T_1^4 - T_2^4)$. The proportionality factor $(\sigma F_A F_E)$ may be conveniently separated into three parts:

σ = the Stefan-Boltzmann radiation constant = 1730×10^{-12} Btu per (hour) (square foot) (Fahrenheit degree absolute temperature to the fourth power).

F_A = the configuration factor which is dimensionless and ≤ 1. This factor accounts for the shape and relative position of the two surfaces. The value of $F_A = 1$ may be used in the cases of large parallel planes, long concentric cylinders or smaller bodies in large enclosures.

F_E = the emissivity factor which is also dimensionless and ≤ 1. This factor ac-

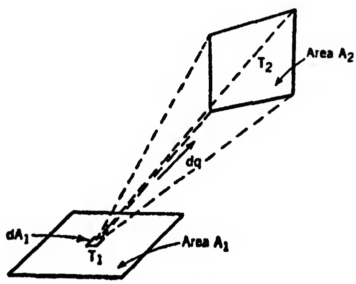


FIG. 3. RADIATION BETWEEN SURFACES

counts for the absorption and emission characteristics of the surfaces for the radiation which exists. Individual emissivities (ϵ) should be taken from Table 3 and applied, for either radiation or absorption, as follows:

a. For a small body in a large enclosure, use the emissivity of the small body only: $F_E = \epsilon_1$.

b. For large parallel planes, long concentric cylinders or large enclosed bodies, use both emissivities in the equation:

$$F_E = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$$

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 4[3] for cold surfaces as low as −39 F to warmer surfaces as high as 139 F. The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 3. For radiation table at higher temperatures, and further discussion of radiation calculations, see Chapter 31.

Combined Convection and Radiation

It should be noted that the previous equations and tables give the heat transfer by convection and by radiation computed separately. In many practical cases it is desirable to treat convection and radiation as a single combined process, using a first-power equation:

$$q_{rc} = h_{rc} \, A \, (t_1 - t_2) \tag{4}$$

where q_{rc} is the total heat flow due to radiation and convection, in Btu

per hour. Values of h_{rc}, the surface or film conductance for combined radiation and convection, are given in Chapter 6, (Table 1 and Fig. 3). Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

When dealing with the effect of operating temperatures upon the combined heat transfer of a given piece of equipment (as for instance a steam radiator), another form of equation is frequently used:

TABLE 3. RADIATION FACTORS OR EMISSIVITIES, ϵ.
For the determination of factor F_E in Equation 3

Class	Surfaces	Fraction of Black-Body Radiation		Absorptivity for Solar Radiation
		At 50–100 F	At 1000 F	
1	A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99	0.97 to 0.99	0.97 to 0.99
2	Black non-metallic surfaces such as asphalt, carbon, slate, paint, paper	0.90 to 0.98	0.90 to 0.98	0.85 to 0.98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95	0.75 to 0.90	0.65 to 0.80
4	Yellow and buff brick and stone, firebrick, fire clay	0.85 to 0.95	0.70 to 0.85	0.50 to 0.70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0.95	0.60 to 0.75	0.3 to 0.5
6	Window glass	0.90 to 0.95	....	Transparent[a]
7	Bright aluminum paint; gilt or bronze paint	0.4 to 0.6	....	0.3 to 0.5
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.2 to 0.3	0.3 to 0.5	0.4 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0.05	0.05 to 0.15	0.3 to 0.5
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04	0.05 to 0.10	0.10 to 0.40

[a]Reflects about 8 per cent

$$q_{rc} = B\,A\,(t_1 - t_2)^n \qquad (5)$$

Values of n in this equation usually range from 1.3 to 1.5 (see Chapter 25). The chief advantage of this equation is the convenience of representing heat transfer performance on logarithmic coordinates, and the factor B should be regarded as a simple constant of proportionality.

HEAT-FLOW RESISTANCE

In most of the steady-state heat transfer problems encountered in air conditioning applications, more than one of the heat transfer mechanisms are effective, and the thermal current flows through several resistances in

series or in parallel. In using the resistance concept the calculations involved are analogous to the application of Ohm's Law in electricity, *viz.*, the heat flow or thermal current is directly proportional to the thermal potential or temperature difference, and inversely proportional to the thermal resistance:

$$R \tag{6}$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \cdots + R_n \tag{7}$$

Similarly, conductance is the reciprocal of resistance, and for heat flow

TABLE 4. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]
Expressed in Btu per (square foot) (hour)

Temp F Deg	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	97.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a]*Example:* Radiation from walls of room at 32 F to surface at − 25 F for effective emissivity of 0.95 (102 − 62.3) 0.95 = 37.7 Btu per (square foot) (hour).

through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots + \frac{1}{R_n} \tag{8}$$

Practical Heat Transfer Problems

The use of these relations for resistance and conductance makes possible the solution of many practical heat transfer problems. As discussed in Chapters 6, 7 and 28, the practical analyses of heat transfer in building walls, in fin-tube coils and in pipe coverings, are usually computed by this method. The same resistance analysis may be applied to complicated steady-state *conduction* problems. Table 5 gives the resistances in six common cases of steady-state conduction.

A complete analysis by the resistance method is well illustrated by

considering the heat transfer from the air outside to the cold water inside of an insulated pipe. The temperature gradients and the nature of the resistance analysis are indicated by the two sketches of Fig. 4.

Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total transfer by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection and radiation into main cold water streams. (Radiation is not significant on the water side as liquids are sensibly opaque to radiation, although water transmits energy in the visible region). The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 4, the heat transferred for a given length N of pipe, q_{rc}, Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance, R_2, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_T, hour Fahrenheit degrees per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{9}$$

where the resultant parallel resistance R_4 is obtained from:

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from this relation. Then the heat transfer for the length of pipe (N, ft) can be established by the relation:

$$q_{rc} \text{ (Btu per hour)} = \frac{(t_0 - t_i)}{R_T} \tag{10}$$

For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N} \text{ (Btu per hour foot)} = \frac{(t_0 - t_i)}{R_T N} \tag{11}$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

$$\Delta t = R\, q_{rc}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable integration of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{12}$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 5 lists such solutions for

six different conduction systems. Table 2 in Chapter 6 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 5, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 4, for the calculation of R_2 and R_3, is case 2 in Table 5. Thus for a *1 ft length* of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of material having a conductivity of 0.025:

$$R_2 = \frac{\log_e \frac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ hr Fahrenheit degree per Btu.}$$

$$R_3 = \frac{\log_e \frac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ hr Fahrenheit degree per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material,

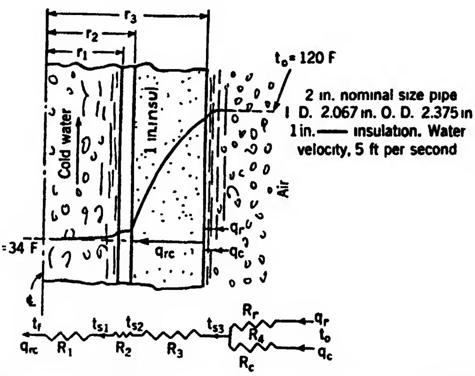


FIG. 4. HEAT TRANSFER CONDITIONS IN AN INSULATED COLD WATER LINE

R_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. The unit conductances for thermal convection, h_c, Btu per (hour) (square foot) (Fahrenheit degree), have been determined by test for many flow systems. These data may be employed to predict the conductances for *similar flow systems*. Table 2 summarizes some empirical equations expressing such test results.

For the problem under consideration (Fig. 4) case 3 of Table 2 is applicable for the calculation of the cold water side convection resistance R_1. Corresponding to the water velocity of 5 fps, the mass velocity is:

G = 5 (ft per sec) $\times$ 62.4 (lb per cu ft) $\times$ 3600 (sec per hr) = 11.2×10^5 lb per (hour) (square foot).

The inside diameter of the pipe D is $2.067/12 = 0.1725$ ft.

The average water film temperature will be estimated as 36 F (mixed mean fluid temperature of 34 F). Then case 3, Table 2 yields:

$$h_c = 0.00486(1 + 0.36) \frac{(11.2 \times 10^5)^{0.8}}{(0.1725)^{0.2}} = 650 \text{ Btu per (hr) (sq ft) (F deg).}$$

The transfer area on which this conductance is based is the inside tube

area. Associated with 1 ft length of pipe there are:

$$\pi \times \frac{2.067}{12} \times 1 = 0.542 \text{ sq ft.}$$

Thus the resistance for 1 ft of tube length is:

$$R_1 = \frac{1}{h_o \pi D \times 1} = \frac{1}{650 \times 0.542} = 2.8 \times 10^{-3} \text{ hr Fahrenheit degree per Btu.}$$

Case 9, Table 2 is applicable for calculating the free thermal convection resistance, R_c, existing between the surrounding air and the insulation. The air temperature is given as 120 F. As an approximation a 20 deg temperature difference between the air and the pipe surface will be assumed. $D = 4.375/12 = 0.364$ ft. Then case 9 yields:

$$h_c = 0.23\left(\frac{20}{0.364}\right)^{0.25} = 0.63 \text{ Btu per (hour) (square foot) (Fahrenheit degree).} \quad (13)$$

This result may not be deemed conservative inasmuch as the expression is for *still* air. If, however, the air is not still, but flows at approximately 5 mph or 7 fps the mass velocity corresponds to:

$$G = 7 \times 0.07 \times 3600 = 1770 \text{ lb air per (hour) (square foot).}$$

A magnitude of $k = 0.014$ Btu per (hour) (square foot) (Fahrenheit degree per one foot thickness) applied to case 4 yields:

$$h_c = 0.45\left(\frac{0.014}{0.364}\right) + 0.178\,(1770)^{0.56}\left(\frac{0.014}{0.364}\right)^{0.44}$$

$$= 0.017 + 2.8 = 2.8 \text{ Btu per (hour) (square foot) (Fahrenheit degree).}$$

This conductance is based on 1 sq ft of outside lagging area. Thus, since there are $\pi \times (4.375/12) = 1.14$ sq ft of outside lagging area associated with 1 ft length of pipe:

$$R_c = \frac{1}{2.8 \times 1.14} = 0.312 \text{ hr Fahrenheit degree per Btu.}$$

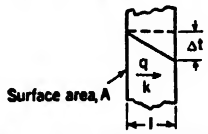
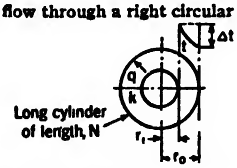
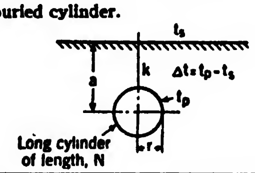
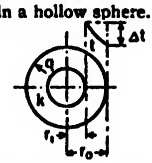
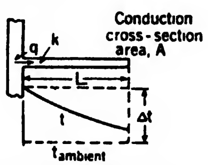
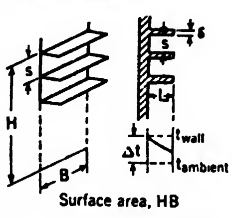
The radiation resistance, R_r, which acts in parallel with the convection resistance, R_c, for the transfer of heat to the surface of the insulation, may be calculated. For the purposes of this illustrative problem it will be assumed that the insulated pipe is exposed to (*sees*) surroundings, which exist at 120 F. Then the angle factor, F_A, is unity and for an estimated surface emissivity of 0.9 (see Table 3), $F_e = 0.9$. As a first approximation the insulation surface temperature will be estimated as 20 deg below the surroundings at 120 F. Then the radiation per degree of temperature difference, by Equation 3 (or more conveniently by Table 5) divided by the temperature difference will be:

$$h_r = \frac{(196 - 170)\,0.95}{20} = 1.17 \text{ Btu per (hour) (square foot) (Fahrenheit degree).}$$

The outside surface area of the insulation associated with 1 ft of pipe length was previously calculated as 1.14 sq ft. Thus:

$$R_r = \frac{1}{1.17 \times 1.14} = 0.75 \text{ hr Fahrenheit degree per Btu.}$$

Table 5. Solutions for Some Steady-State Thermal Conduction Problems[a, b]

No.	System	Expressions for the resistance R entering into the equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter). 	$R = \frac{L}{kA}$
2.	Radial flow through a right circular cylinder. 	$R = \frac{\log_e \frac{r_o}{r_i}}{2\pi kN}$ (See footnote *c*).
3.	The buried cylinder. 	$R = \frac{\log_e\left(\frac{a+\sqrt{a^2+r^2}}{r}\right)}{2\pi kN} = \frac{\cosh^{-1}\left(\frac{a}{r}\sqrt{1+\frac{r^2}{a^2}}\right)}{2\pi kN}$ For $\frac{a}{r} \geq 3$, a satisfactory approximation is: $R = \frac{\log_e \frac{2a}{r}}{2\pi kN} = \frac{\cosh^{-1} \frac{a}{r}}{2\pi kN}$
4.	Radial flow in a hollow sphere. 	$R = \frac{\frac{1}{r_i} - \frac{1}{r_o}}{4\pi k}$
5.	The straight fin or rod heated at one end. 	$R = \frac{m}{h_s p \tanh mL}$ (see footnotes *d* and *e*). For $ml > 2.3$, $\tanh mL \approx 1$ $m = \sqrt{h_s p/kA}$ A = conduction cross-section area. p = perimeter of cross-section A. h_s = unit conductance to the surroundings from the fin surface. k = thermal conductivity fin material. Δt = wall temperature—ambient temperature
6.	Finned surface of area HB.  Surface area, HB	$R = \frac{(s + \delta)}{h_s\left(\frac{2}{m}\tanh ml + s\right)HB}$ $m = \sqrt{\frac{h_s p}{kA}} = \sqrt{\frac{2\,h_s}{k\delta}}$ Δt defined as in Case 5 above.

[a] The dimensions to be employed in these solutions are: length of dimension p, L, r = feet; units of k = Btu per (hour) (square foot) (Fahrenheit degree for one foot thickness); units of h, Btu per (hour) (square foot) (Fahrenheit degree); units of area, A = square feet.

[b] The thermal conductivity, k, in these solutions should be taken at the average material temperature (see Table 2).

[c] $\log_e x = 2.303 \log_{10} x$.

[d] This expression can also be employed as an approximation for tapered fins or of annular fins by employing average magnitudes of A and p.

[e] tanh is the hyperbolic tangent.

The resultant resistance of R_c and R_r acting in parallel (see Fig. 4) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.312} + \frac{1}{0.75} = 4.54 \text{ Btu per (hour) (Fahrenheit degree).}$$

$$R_4 = 0.22 \text{ hr Fahrenheit degree per Btu.}$$

The over-all resistance, R_T, surroundings to cold water, is the sum of $R_1 + R_2 + R_3 + R_4 = 4.1$ hr F deg per Btu for 1 ft length of pipe. Note that the controlling resistances are R_3 and R_4 and that neglect of both R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{re}}{N} = \frac{t_o - t_f}{R_T} = \frac{120 - 34}{4.1} = 21 \text{ Btu per (hour) (foot)}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1 ft pipe length:

$$q_{re} = 21 \text{ Btu per hour.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 12 as:

$t_o - t_{s3}$ air to insulation surface $= R_4\, q_{re} = 0.22 \times 21 = 4.6$ F.
$t_{s3} - t_{s2}$ through the insulation $= R_3\, q_{re} = 3.9 \times 21 = 82$ F.
$t_{s2} - t_{s1}$ through the pipe wall $= R_2\, q_{re} = 8.5 \times 10^{-4} \times 21 = 0.02$ F.
$t_{s1} - t_f$ pipe wall to cold water $= R_1\, q_{re} = 2.8 \times 10^{-3} \times 21 = 0.06$ F.

The solution was obtained on the assumption that the air temperature and the outside temperature differed by 20 deg. In order to obtain a slightly better estimate of the rate of heat transfer the numerical solution should be repeated using the temperatures calculated from the previous listed temperature differences.

The foregoing problem serves to illustrate a general method of solving steady-state heat transfer problems. There are many problems which cannot be approximated by steady-state solutions. For instance, the problem of pipe line insulation in transient service; the behavior of automatically controlled thermoflow circuits; or the periodic absorption of solar energy by roof and wall structures during the day and nocturnal radiation to the *cold* sky at night. The transient heat transfer problem differs from the steady-state in that energy storage rates need to be considered. Thus thermal capacity in addition to resistance effects is significant. The vector sum of the thermal capacitance and resistance is the thermal impedance. It is not within the scope of this chapter to deal with these problems. There are, however, solutions available in graphical form for certain special cases. Also a general approximate method may be employed which is analogous to the treatment of capacity-resistance lumped parameter electrical circuits.

REFERENCES

[1] Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937).

[2] The Transmission of Heat by Radiation and Convection, by Griffith and Davis (*Special Report* No. 9, 1922, *Department of Scientific and Industrial Research*, His Majesty's Stationery Office, London, England).

[3] Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July, 1936, p. 782).

BIBLIOGRAPHY

Heat Transmission, by W. H. McAdams (McGraw-Hill Book Co., 2nd edition, 1942).

Industrial Heat Transfer, by Alfred Schack (John Wiley & Sons, 1933).

Calculation of Heat Transmission, by M. Fishenden and O. A. Saunders (His Majesty's Stationery Office, London, 1932).

CHAPTER 6

HEAT TRANSMISSION COEFFICIENTS OF BUILDING MATERIALS

Heat Transfer Symbols; Calculating Over-all Coefficients; Conductivity of Homogeneous Materials; Surface Conductance; Air Space Conductance; Practical Coefficients and Their Use; Computed Heat Transmission Coefficients; Roof Coefficients; Combined Ceiling and Roof Coefficients; Basement Floor, Basement Wall, and Concrete Slab Floor Coefficients; Calculating Surface Temperatures; Water Vapor and Condensation; Vapor Transmission; Condensation Control

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THE design of air conditioning or heating systems for buildings requires a knowledge of the thermal properties of the walls enclosing the space. The rate of heat flow through the walls under steady-state conditions at design temperatures is usually the basis for calculating the heat required. For a given wall under standard conditions the rate is a specific value designated as U, the *over-all coefficient of heat transmission*. It may be determined by test in a guarded hot box apparatus or it may be computed from known values of the thermal conductance of the various components. Because testing of all combinations of building materials is impracticable, the procedure and necessary data for calculation of the value of U are given in this chapter, together with tables of computed values for the more common constructions.

HEAT TRANSFER SYMBOLS

U = over-all coefficient of heat transmission (air to air); the time rate of heat flow expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference between air on the inside and air on the outside of a wall, floor, roof or ceiling). The term is applied to the usual combinations of materials in construction and also to single materials, such as window glass, and includes the surface conductance on both sides.

k = thermal conductivity; the time rate of heat flow through a homogeneous material under steady conditions through unit area per unit temperature gradient in the direction perpendicular to the area. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree per inch of thickness). Materials are considered homogeneous when the value of k is not affected by variation in thickness or size of sample within the range normally used in construction.

C = thermal conductance; the time rate of heat flow through a material from one of its surfaces to the other per unit temperature difference *between the two surfaces*. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree). The term is applied to specific materials as used which may be either homogeneous or heterogeneous.

f = film or surface conductance; the time rate of heat flow between a surface and the surrounding air. Its value is expressed in Btu per (hour) (square foot of surface) (Fahrenheit degree temperature difference). Subscripts i and o are used to differentiate between inside and outside surface conductances respectively.

a = thermal conductance of an air space; the time rate of heat flow through an air space per unit temperature difference *between the boundary surfaces*. Its value is expressed in Btu per (hour) (square foot of area) (Fahrenheit degree). The conductance of an air space is dependent on the temperature difference, the height, the depth, the position and the character of the boundary surfaces. The relationships are not linear and accurate values must be obtained by test and not by computation.

R = thermal resistance. Its value is expressed in Fahrenheit degrees per (Btu) (hour) (square foot). It may represent any of the following and must therefore be

properly described as one of the following:

$\frac{1}{U}$ = over-all or air-to-air resistance

$\frac{1}{k}$ = resistance per unit thickness (resistivity)

$\frac{1}{C}$ = resistance of a material (surface-to-surface)

$\frac{1}{f}$ = film or surface resistance

$\frac{1}{a}$ = air space resistance

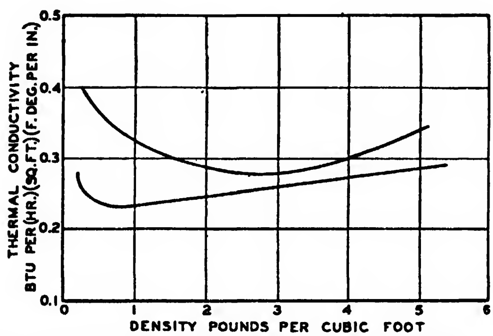


FIG. 1. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH DENSITY—FOR FIBROUS MATERIAL

CALCULATING OVER-ALL COEFFICIENTS

From Chapter 5, Equation 7, the total resistance to heat flow through a wall is equal numerically to the sum of the resistances in series.

$$R_T = R_1 + R_2 + R_3 + R + \cdots + R_n \tag{1}$$

where, $R_1 + R_2$, etc. are the individual resistances of the wall components.

R_T = total resistance.

For a wall of a single homogeneous material of conductivity k and thickness x, with surface coefficients f_i and f_o,

$$R_T = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \tag{2}$$

Then by definition, $$U = \frac{1}{R_T}$$

For a wall with air space construction and consisting of two homogeneous materials of conductivities k_1 and k_2, thicknesses x_1 and x_2 respectively, and separated by an air space of conductance a,

$$R_T = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{3}$$

and $$U = \frac{1}{R}$$

In the case of types of building materials having non-uniform or irregular sections such as hollow clay tile or concrete blocks, it is necessary to use the conductance C of the section unit as manufactured instead of a conductivity k. The resistance of the section $1/C$ is therefore substituted for x/k in Equations 2 and 3.

CONDUCTIVITIES AND CONDUCTANCES

The method of calculating the over-all coefficient of heat transmission for a given construction is comparatively simple, but accurate values of conductivities and conductances must be used to obtain satisfactory results. In addition there are sometimes parallel heat flow paths of different resistances in the same wall, and these may necessitate modification of the formula. In such cases calculated results should be checked by test measurements.

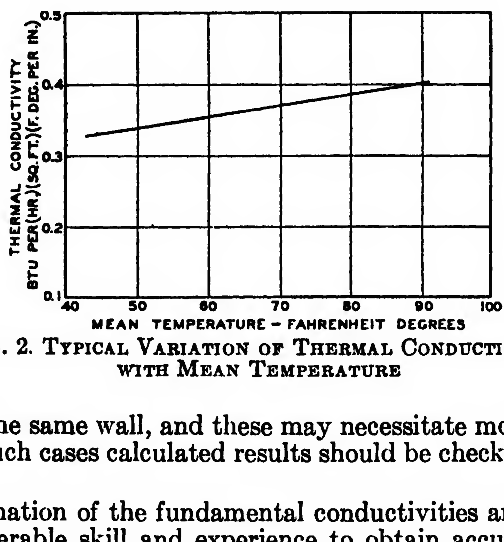


FIG. 2. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH MEAN TEMPERATURE

The determination of the fundamental conductivities and conductances requires considerable skill and experience to obtain accurate results. It is recommended that thermal conductivities of homogeneous materials be determined by means of the Guarded Hot Plate[1]. For determination of conductances, a Guarded Hot Box method[2] is generally used.

Tables 1 and 2 give conductivities and conductances which are quite generally used in calculation and which have been selected from various sources. Wherever possible the properties of the material and test conditions are given. In selecting and applying heat transmission values to any construction, caution is necessary, because coefficients for the same material may differ because of variations which occur in test methods, in the materials themselves, or in the temperature of the material when tested.

Conductivity of Homogeneous Materials

Thermal conductivity is a property of a homogeneous material and of types of building materials such as lumber, brick and stone which may be considered homogeneous. Most insulating materials, except air spaces and reflective types, are of a porous nature and consist of combinations of solid matter with small air cells. The thermal conductivity of these

Table 1. Conductances (*C*) for Surfaces and Air Spaces

All conductance values expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference).

Section A. Surface Conductances for Still Air[a]

Position of Surface	Direction of Heat Flow	Surface Emissivity $\epsilon = 0.83$	Surface Emissivity $\epsilon = 0.05$
Horizontal	Upward	1.95	1.16
Horizontal	Downward	1.21	0.44
Vertical		1.52*	0.74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

Mean Temp Fahr Deg	Conductances of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

Section C. Conductances and Resistances of Air Spaces Faced on One Surface with Reflective Insulation[c]

Location and Position of Air Space	Direction of Heat Flow	Temp[d] Diff Fahr Deg		Conductance[e] (*C*) No. of Air Spaces			Resistance[e] $\left(\frac{1}{C}\right)$ No. of Air Spaces		
		Winter	Summer	1	2	3	1	2	3
Rafter Space (8 in.)									
Horizontal	Down	45			0.10	0.07		10.00	14.29
Horizontal	Up	45			0.27	0.17		3.70	5.88
Horizontal	Down		25		0.09	0.06		11.11	16.67
Horizontal	Up		25		0.24	0.16		4.17	6.25
30 deg slope	Down	45			0.15	0.10		6.67	10.00
30 deg slope	Up	45			0.25	0.17		4.00	5.88
30 deg slope	Down		25		0.13	0.09		7.69	11.11
30 deg slope	Up		25		0.23	0.14		4.35	7.14
Stud Space (3⅝ in.)									
Vertical[f]		30		0.34			2.94		
Vertical		40			0.23	0.13		4.35	7.69
Vertical[f]			15	0.32			3.13		
Vertical			20		0.18	0.11		5.56	9.09
Vertical[g]		30		0.46			2.17		

[a] Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 513).

[b] A.S.H.V.E. Research Report No. 825—Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 165).

[c] Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S.H.V.E. Transactions, Vol. 46, 1940).

[d] Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e] These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0.05 facing each space and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f] Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space. The resistance of the small air space between the plaster base and paper was 0.43.

[g] Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 351).

* The recommended surface conductance for calculating heat losses for still air for non-reflective surfaces is 1.65 Btu. For a 15 mph wind velocity, the recommended value is 6.0 Btu. These coefficients were derived from Fig. 3 which was based on tests conducted at the University of Minnesota, and apply to vertical surfaces.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	(C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS (NON-INSULATING)	Compressed cement and asbestos sheets	123	86	2.70	....	0.37	....	(1)
	Corrugated asbestos board	20.4	110	0.48	....	2.08	....	(2)
	Pressed asbestos mill board	60.5	86	0.84	....	1.19	....	(3)
	Gypsum board—gypsum between layers of heavy paper	62.8	70	1.41	....	0.71	....	(3)
	3/8 in. gypsum board	....	....	....	3.73	....	0.27	
	1/2 in. gypsum board	....	....	....	2.82	....	0.35	
	1/2 in. gypsum board	53.5	90	....	2.60	....	0.38	(1)
FRAME CONSTRUCTION COMBINATIONS	1 in. fir sheathing and building paper	....	30	....	0.86	....	1.16	(4)
	1 in. fir sheathing, building paper and yellow pine lap siding	....	20	....	0.50	....	2.00	(4)
	1 in. fir sheathing, building paper and stucco	....	20	....	0.82	....	1.22	(4)
	Pine lap siding and building paper, siding 4 in. wide	....	16	....	0.85	....	1.18	(4)
	Yellow pine lap siding	....	....	....	1.28	....	0.78	(4)
MASONRY MATERIALS BRICK	Damp or wet	....	....	5.0[e]	....	0.20	....	(2)
	Common yellow clay brick[a]	....	....	4.8	....	0.21	....	(4)
	One tier yellow common clay brick, one tier face brick, approx. 8 in. thick[a]	....	....	....	0.77	....	1.30	(4)
CLAY TILE, HOLLOW	2 in. Tile, 1/2 in. plaster both sides	120.0	110	....	1.00	....	1.00	(2)
	4 in. Tile, 1/2 in. plaster both sides	127.0	100	....	0.60	....	1.67	(2)
	6 in. Tile, 1/2 in. plaster both sides	124.3	105	....	0.47	....	2.13	(2)
	8 in. Tile, average of 8 types (Walls No. 59, 63, 64, 66, 67, 90, 91, 92[a])	....	....	....	0.52	....	1.92	(4)
	12 in. Clay tile wall: 8 in. x 5 in. x 12 in. and 4 in. x 5 in. x 12 in.[a]	....	....	....	0.26	....	3.84	(4)
CONCRETE	Sand and gravel aggregate, various ages and mixes	....	....	11.35 to 16.36	....	0.09 to 0.06	....	(5)
	Sand and gravel aggregate	142	75	12.6	....	0.08	....	(4)
	Limestone aggregate	132	75	10.8	....	0.09	....	(4)
	Cinder aggregate	97	75	4.9	....	0.22	....	(4)
	Steam treated limestone slag aggregate[a]	74.6	75	2.27	....	0.44	....	(4)
	Pumice (Mined in California) aggregate[a]	65.0	75	2.42	....	0.41	....	(4)
	Expanded burned clay aggregate[a]	59.9	75	2.28	....	0.44	....	(4)
	Burned clay aggregate[a]	67.1	75	2.86	....	0.35	....	(4)
	Blast furnace slag aggregate	76.0	70	1.6	....	0.63	....	(3)
	Expanded vermiculite aggregate	20	90	0.68	....	1.47	....	(3)
	Expanded vermiculite aggregate	26.7	90	0.76	....	1.32	....	(3)

AUTHORITIES:

[1] U. S. Bureau of Standards, tests based on samples submitted by manufacturers.

[2] A. C. Willard, L. C. Lichty and L. A. Harding, tests conducted at the University of Illinois.

[3] J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.

[4] F. B. Rowley, et al, tests conducted at the University of Minnesota.

[5] A.S.H.V.E. Research Laboratory.

[6] E. A. Allcut, tests conducted at the University of Toronto.

[a] See Thermal Conductivity of Building Materials, by F. B. Rowley and A. B. Algren (University of Minnesota *Engineering Experiment Station Bulletin* No. 12).

[b] Heat Transmission Through Insulation as Affected by Orientation of Wall, by F. B. Rowley and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 331).

[c] The Effect of Convection in Ceiling Insulation, by G. B. Wilkes and L. R. Vianey (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 196).

[d] See A.S.H.V.E. RESEARCH REPORT No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

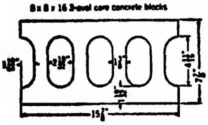
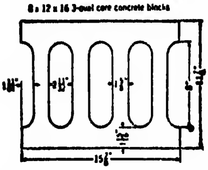
[e] See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[f] See BMS13, U. S. Department of Commerce, National Bureau of Standards, Washington, D. C.

[g] Roofing, 0.15 in. thick (1.34 lb per square foot), covered with gravel (0.83 lb per square foot), combined thickness assumed 0.25.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS —(Continued) Concrete—(Continued)	Expanded vermiculite aggregate	35	90	0.86	....	1.16	......	(3)
	Expanded vermiculite aggregate	50	90	1.10	....	0.91	......	(3)
	Concrete plank	76	75	2.5	....	0.40	......	(3)
	Cellular concrete	40.0	75	1.06	....	0.94	......	(3)
	Cellular concrete	50.0	75	1.44	....	0.69	......	(3)
	Cellular concrete	60.0	75	1.80	...	0.56	......	(3)
	Cellular concrete	70.0	75	2.18	....	0.46	......	(3)
8 In. Concrete Blocks 8 x 8 x 16 3-oval core concrete blocks 	8 in. three oval core, sand and gravel aggregate[a]	126.4	40	......	0.90	......	1.11	(4)
	8 in. three oval core, crushed limestone aggregate[a]	134.3	40	......	0.86	......	1.16	(4)
	8 in. three oval core, cinder aggregate[a]	86.2	40	......	0.58	......	1.73	(4)
	8 in. three oval core, burned clay aggregate[a]	67.7	40	......	0.50	......	2.00	(4)
	8 in. three oval core, expanded blast furnace slag aggregate[a]	......	40	......	0.49	......	2.04	(4)
12 In. Concrete Blocks 8 x 12 x 16 3-oval core concrete blocks 	12 in. three ova core, sand and gravel aggregate[a]	124.9	40	......	0.78	......	1.28	(4)
	12 in. three oval core, cinder aggregate[a]	86.2	40	......	0.53	......	1.88	(4)
	12 in. three oval core, burned clay aggregate[a]	76.7	40	......	0.47	......	2.13	(4)
Gypsum	3 in. solid gypsum partition tile[a]	......	...	2.41	...	0.42	....	(4)
	3 in. three cell gypsum partition tile[a]	......	...	......	0.74	......	1.35	(4)
	4 in. three cell gypsum partition tile[a]	......	...	......	0.60	......	1.67	(4)
	87½ per cent gypsum, 12½ per cent wood chips	51.2	74	1.66	....	0.60	......	(4)
	Gypsum plaster	......	....	3.30	...	0.30	......	
PLASTERING MATERIALS	Gypsum plaster, ⅝ in. thick	......	73	......	8.80	......	0.11	(4)
	Cement plaster	......	....	8.00	....	0.13	......	(2)
	Wood, lath and plaster, total thickness ¾ in.	......	70	......	2.50	......	0.40	(4)
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	0.85	...	1.18	......	(3)
	Insulating plaster 0.9 in. thick applied to ⅜ in. gypsum board	54.0	75	......	1.07	......	0.93	(3)
ROOFING	Asbestos shingles	65.0	75	......	6.0	......	0.17	(3)
	Asphalt, composition or prepared	70.0	75	......	6.5	......	0.15	(3)
	Asphalt shingles	70.0	75	......	6.5	......	0.15	(3)
	Built-up roofing, bitumen or felt, gravel or slag surfaced[g]	......	....	1.33	....	0.75	......	(2)
	Slate	......	....	10.00	...	0.10	.. .	
	Wood shingles	......	....	......	1.28	......	0.78	
WOODS	Balsa	20.0	90	0.58	....	1.72	......	(1)
	Balsa	8.8	90	0.38	....	2.63	......	(1)
	Balsa	7.3	90	0.33	....	3.03	......	(1)
	California redwood, 0 per cent moisture[a]	28.0	75	0.70	....	1.43	......	(4)
	Cypress	28.7	86	0.67	....	1.49	......	(1)
	Douglas fir, 0 per cent moisture[a]	34.0	75	0.67	....	1.49	......	(4)
	Eastern hemlock, 0 per cent moisture[a]	30.0	75	0.76	....	1.32	......	(4)
	Long leaf yellow pine, 0 per cent moisture[a]	40.0	75	0.86	....	1.16	......	(4)
	Mahogany	34.3	86	0.90	..	1.11	......	(1)

See footnotes on first page of Table 2.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
WOODS—(Continued)	Hard maple, 0 per cent moisture[a]	46.0	75	1.05	....	0.95	......	(4)
	Maple	44.3	86	1.10	....	0.91	......	(1)
	Maple, across grain	40.0	75	1.20	....	0.83	......	(3)
	Norway pine, 0 per cent moisture[a]	32.0	75	0.74	....	1.35	......	(4)
	Red cypress, 0 per cent moisture[a]	32.0	75	0.79	....	1.27	......	(4)
	Red oak, 0 per cent moisture[a]	48.0	75	1.18	....	0.85	......	(4)
	Short leaf yellow pine, 0 per cent moisture[a]	36.0	75	0.91	....	1.10	......	(4)
	Soft elm, 0 per cent moisture[a]	34.0	75	0.88	....	1.14	......	(4)
	Soft maple, 0 per cent moisture[a]	42.0	75	0.95	....	1.05	......	(4)
	Sugar pine, 0 per cent moisture[a]	28.0	75	0.64	....	1.56	......	(4)
	Virginia pine	34.3	86	0.96	....	1.04	......	(1)
	West coast hemlock, 0 per cent moisture[a]	30.0	75	0.79	....	1.27	......	(4)
	White pine	31.2	86	0.78	....	1.28	......	(1)
	Yellow pine	...	.	1.00	....	1.00	......	(3)
	Sawdust, various	12.0	90	0.41	....	2.44	......	(1)
	Shavings, various from planer	8.8	90	0.41	....	2.44	......	(1)
	Shavings, from maple beech and birch (coarse)	13.2	90	0.36	....	2.78	......	(1)
INSULATING MATERIALS Blanket and Bat Insulations[a]	Chemically treated wood fibers held between layers of strong paper	3.62	70	0.25	....	4.00	......	(3)
	Eel grass between strong paper	4.60	90	0.26	....	3.85	......	(1)
	Eel grass between strong paper	3.40	90	0.25	....	4.00	......	(1)
	Flax fibers between strong paper	4.90	90	0.28	....	3.57	......	(1)
	Chemically treated hog hair between kraft paper	5.76	71	0.26	....	3.85	......	(3)
	Chemically treated hog hair between kraft paper and asbestos paper	7.70	71	0.28	....	3.57	......	(3)
	Hair felt between layers of paper	11.00	75	0.25	....	4.00	......	(3)
	Kapok between burlap or paper	1.00	90	0.24	....	4.17	......	(1)
	Stitched and creped expanding fibrous blanket	1.50	70	0.27	....	3.70	......	(3)
	Paper and asbestos fiber with emulsified asphalt binder	4.2	94	0.28	....	3.57	......	(1)
	Cotton insulating bat	0.875	72	0.24	....	4.17	......	(3)
	Cotton fibers							
	Short Staple Linters, Fireproofed	6.25	90	0.25	....	4.00	......	(1)
	Short Staple Linters, Fireproofed	4.50	90	0.24	....	4.17	......	(1)
	Short Staple Linters, Fireproofed	2.45	90	0.24	....	4.17	......	(1)
	Short Staple Linters, Fireproofed	1.60	90	0.26	....	3.85	......	(1)
	Short Staple Linters, Fireproofed	0.85	90	0.29	....	3.45	......	(1)
	Short Staple Linters, Fireproofed	0.65	90	0.30	....	3.33	......	(1)
	Felted cattle hair	13.00	90	0.26	....	3.84	......	(1)
	Felted cattle hair	11.00	90	0.26	....	3.84	......	(1)
	Felted hair and asbestos	7.80	90	0.28	....	3.57	......	(1)
	Ground paper between two layers, each ⅜ in. thick made up of two layers of kraft paper (sample ¾ in thick)	12.1	75	....	0.40	......	2.50	(4)
	Mineral Wool	4.5	....	0.27	....	3.70	......	
Reflective	See Table 1, Section C	......	....	......	....	......	......	
Insulating Board	Made from sugar cane fiber	13.5	70	0.33	....	3.03	......	(3)
	Made from corn stalks	15.00	71	0.33	....	3.03	......	(3)
	Made from exploded wood fibers	17.90	78	0.32	....	3.12	......	(4)
	Made from hard wood fibers	15.20	70	0.32	....	3.12	......	(3)
	Made from wood fiber	15.90	72	0.33	....	3.03	......	(3)
	Made from wood fiber	15.00	70	0.33	....	3.03	......	(3)
	Made from wood fiber	......	52	0.33	....	3.03	......	(6)
	Made from wood fiber	8.50	72	0.29	....	3.45	......	(3)
	Made from wood fiber	15.20	....	0.33	....	3.03	......	(3)
	Made from wood fiber	16.90	90	0.34	....	2.94	......	(1)

See footnotes on first page of Table 2.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Concluded

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
INSULATING MATERIALS—Continued								
Insulating Board—Continued	Made from licorice root	16.1	81	0.34	....	2.94	......	(3)
	½ in. insulating boards without special finish[f] (eleven samples)	16.5 to 21.8	90	0.33 to 0.40	....	3.03 to 2.50	......	(1)
	1 in. insulating board[e]	13.2	....	0.34	....	2.94	......	(4)
Loose Fill Type	Made from ceiba fibers	1.90	75	0.23	....	4.35	......	(3)
	Made from ceiba fibers	1.60	75	0.24	....	4.17	......	(3)
	Fibrous material made from dolomite and silica	1.50	75	0.27	....	3.70	......	(3)
	Fibrous material made from slag	9.40	103	0.27	....	3.70	......	(1)
	Redwood bark	8.00	90	0.31	....	3.22	......	(1)
	Redwood bark	5.00	75	0.26	....	3.84	......	(3)
	Glass wool fibers 0.0003 in. to 0.006 in. in diameter	1.50	75	0.27	....	3.70	......	(3)
	Granular insulation made from combined silicate of lime and alumina	4.20	72	0.24	....	4.17	......	(3)
	Expanded vermiculite	......	....	0.48	....	2.08	......	(1)
	Expanded vermiculite, particle size—−3 + 14	6.2	....	0.32	....	3.12	......	(3)
	Regranulated cork about ⅛ in. particles	8.10	90	0.31	....	3.22	......	(1)
	Hand applied granular mineral wool 2 in. to 6 in. thick; horizontal position[b]. No covering	6.05 to 7.13	....	0.30 to 0.33	....	3.33 to 3.03	......	(4)
	4 in. machine blown granular mineral wool, horizontal position[b]. No covering	5.74	....	0.30	....	3.33	......	(4)
	Rock wool	10.0	90	0.27	....	3.70	......	(1)
Slab Insulations	Corkboard, no added binder	14.0	90	0.34	....	2.94	......	(1)
	Corkboard, no added binder	10.6	90	0.30	....	3.33	......	(1)
	Corkboard, no added binder	7.0	90	0.27	....	3.70	......	(1)
	Corkboard, no added binder	5.4	90	0.25	....	4.00	......	(1)
	Corkboard[e]	8.7	....	0.29	....	3.45	......	(4)
	Corkboard, asphaltic binder	14.5	90	0.32	....	3.12	......	(1)
	Chemically treated hog hair with film of asphalt	10.0	75	0.28	....	3.57	......	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.8	70	0.30	....	3.33	......	(3)
	Made from 85 per cent magnesia and 15 per cent asbestos	19.3	86	0.51	....	1.96	......	(1)
	Made from shredded wood and cement	24.2	72	0.46	....	2.17	......	(3)
	Made from shredded wood and cement[e]	29.8	....	0.77	....	1.30	......	(4)

See footnotes on first page of Table 2.

materials will vary with density, mean temperature, size of fibers or particles, degree and extent of bond between particles, moisture present, and the arrangement of fibers or particles within the material.

The effect of density upon conductivity (at constant mean temperature) is illustrated for two fibrous materials in Fig. 1. It will be noted that for each there is an optimum density for lowest conductivity. Typical variation of conductivity with mean temperature is shown in Fig. 2.

Surface Conductance

The surface conductance of a wall is the combined heat transfer to or from the wall by radiation, convection and conduction. Each of the three portions making up the total may vary independently of the others, thus affecting the total conductance. The heat transfer by radiation between two surfaces is controlled by the character of the surfaces (emissivity), the temperature difference between them, and the solid angle through which

they *see* each other. The heat transfer by convection and conduction is controlled by the roughness of the surface, by air movement and temperature difference between the air and the surface.

The importance of the effect of temperature of *surrounding surfaces* on the surface conductance due to the effect on radiation is illustrated in Table 3, which applies to a vertical surface at 80 F, with ambient air at 70 F and effective emissivity equal to 0.83[3].

In many cases, because the heat resistance of the internal parts of the wall is high compared with the surface resistance, the surface factors are of minor importance. In other cases, *e.g.* single glass windows, the surface resistances constitute almost the entire resistance and are therefore very important. In a building heated by convection there is only a slight difference between the temperatures of the interior wall surface and the surroundings, but if the building is heated by radiant panels there may be a greater difference[4]. (See also Chapter 31.)

The convection part of the surface conductance coefficient is affected markedly by air movement. This is illustrated by Fig. 3, which shows

TABLE 3. VARIATION IN SURFACE CONDUCTANCE COEFFICIENT WITH DIFFERENT TEMPERATURES OF SURROUNDING SURFACE

SURROUNDING SURFACE TEMPERATURE	75 F	70 F	69 F	60 F	50 F
Convection—Btu per (hr) (sq ft)	6.6	6.6	6.6	6.6	6.6
Radiation—Btu per (hr) (sq ft)	4.4	8.6	9.6	17.0	24.9
Total—Btu per (hr) (sq ft)	11.0	15.2	16.2	23.6	31.5

the surface conductances for different materials at a mean temperature of 20 F and for wind velocities up to 40 mph. These include the radiation portion of the coefficient, which for ordinary building materials under these conditions would be constant at about 0.7 Btu.

Due to these variations for different conditions the selection of surface conductance coefficients for a practical building becomes a matter of judgment. In calculating the over-all heat transmission coefficients for the walls, etc. of Tables 5 to 18, 1.65 has been selected as an *average inside surface conductance* and 6.0 as an *average outside surface conductance* for a 15-mile wind. These values apply only to ordinary building materials and should not be used for bright metal surfaces having a low emissivity.

In special cases, where surface conductance coefficients become important factors in the over-all rates of heat transfer, more selective coefficients may be required. The surface conductance values given in Table 1, **Section A** are based on recent tests and are for still air conditions and surface emissivities of 0.83 and 0.05 respectively, and may be used where it is desirable to differentiate between horizontal and vertical surfaces or where coefficients applicable to low-emissivity surfaces are required.

Air Space Conductance

The transfer of heat across an air space involves the boundary surfaces as well as the intervening air, consequently the factors influencing surface conductance play an important part in determining the conductance of

the air space. The coefficients given for air space conductance represent the total conductance from surface to surface.

The radiation portion of the coefficient is affected by the difference in temperature between the boundary surfaces and by their respective emissivities and is practically independent of depth. The convection and conduction transfer is controlled by depth and shape of the air space, the roughness of the boundary surfaces, the mean temperature and the direction of heat flow. For air spaces usually employed in building construction, the radiation and convection factors vary independently of each other.

Table 1, **Section B** gives experimentally-determined conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc.,

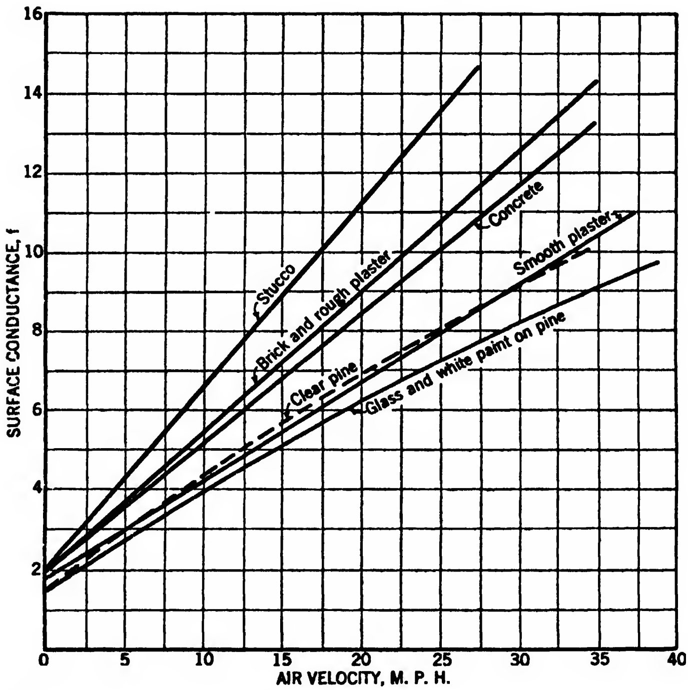


Fig. 3. Curves Showing Relation Between Surface Conductances for Different Surfaces at 20 F Mean Temperature

having emissivity coefficients of 0.8 or higher, and having extended parallel surfaces perpendicular to the direction of heat flow. The conductances decrease as the depth is increased but change only slightly for spaces greater than $\frac{3}{4}$ in. Air space tests reported by Wilkes and Peterson gave conductance values for air spaces of $3\frac{5}{8}$ in. depth having boundary surfaces with emissivity values of 0.83 as follows[5].

Vertical	1.17
Horizontal (heat flow upward)	1.32
Horizontal (heat flow downward)	0.94

Since, in buildings, the same constructions may be used for conditions where the direction of heat flow may be in one direction or its opposite,

and since much of the construction involves vertical air spaces, an average value of 1.10 Btu per (hour) (square foot) (Fahrenheit degree temperature difference) was chosen for use in calculating the over-all coefficients in Tables 5 to 18 wherever air spaces $\frac{3}{4}$ in. or more in depth were involved.

If one or both boundary surfaces of an air space are faced with metals which have low emissivity surfaces, the radiant heat transfer will be greatly reduced in comparison with that occurring from surfaces of ordinary building materials. Table 1, **Section C** gives conductances and resistances of air spaces *bounded by one reflective surface* with an emissivity of 0.05. These values include heat transferred both by radiation and convection, but the radiation component is relatively small for the test conditions.

When reflective materials are installed with single or multiple air spaces, the position (vertical, horizontal or inclined) of the material and the direction of heat flow must be taken into consideration. For example, the resistance to *upward* heat flow is about one-third the resistance to *downward* heat flow in a horizontal position (Table 1, **Section C**). The difference between the conductance through vertical air spaces and that through horizontal and sloping air spaces with upward heat flow is considerably less. For upward heat flow it is recommended that a value of 0.46 be used for the conductance of horizontal or sloping air spaces bounded on one side by reflective materials having an emissivity of approximately 0.05. The same conductance value is also recommended for similar vertical air spaces.

When considering heat transfer to and from reflective surfaces in building construction, the emissivity should be known. This can be determined directly for the long wave length radiation corresponding to average room and wall temperatures. The possibility of change in emissivity with time of exposure due to surface coatings, chemical action, deposition of dust, etc. must be considered in selecting a material for use[6].

PRACTICAL COEFFICIENTS AND THEIR USE

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making actual tests. In Table 2 coefficients are given for a group of materials which have been selected from tests by various authorities. Since there is some variation in the resulting values due to variations in materials and in test conditions, average values for the usual conditions encountered in building practice have been selected and listed in Table 4. These coefficients were used in the calculation of overall coefficients given in Tables 5 to 18. These tables constitute typical examples of combinations frequently used, but any special constructions not given can be computed by the use of the conductivity values in Table 4 and the fundamental heat transfer formulas.

Caution

The user should realize that the average conductivity and conductance values given in Tables 2 or 4 do not necessarily apply to all products of the same general description. In using these values judgment should be exercised with regard to the extent to which the product (either as received or as applied) will comply with the tabulated values. Exact conductivities or conductances for specific materials should be obtained from the maker.

Insulating Materials

In order to determine the benefit derived from the addition of insulating materials to a given construction, the over-all coefficient of heat transmis-

TABLE 4. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 5 TO 18

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

MATERIAL	DESCRIPTION	CONDUCTIVITY OR CONDUCTANCE		RESISTANCE	
		(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$
AIR SPACES					
Bounded by ordinary materials	Vertical[a], ¾ in. or more in width	......	1.10	......	0.91
Bounded by aluminum foil	Vertical[a], ¾ in. or more in width	......	0.46	......	2.17
EXTERIOR FINISHES (Frame Walls)					
Brick Veneer	4 in. thick (nominal)	......	2.27	......	0.44
Stucco (1 in.)		12.50	......	0.08	......
Wood Shingles		......	1.28	......	0.78
Yellow Pine Lap Siding		......	1.28	......	0.78
INSULATING MATERIALS					
Aluminum Foil	See Air Spaces	......	......	......	......
Bats and Blankets	Made from mineral or vegetable fiber or animal hair, enclosed or open	0.27	......	3.70	......
Corkboard	Pure, no added binder	0.30	......	3.33	......
Insulating Board	Vegetable fiber	0.33	......	3.03	......
Mineral Wool	Fiber made from rock, slag or glass	0.27	......	3.70	......
Vermiculite	Expanded	0.48	......	2.08	......
INTERIOR FINISHES					
Composition Wallboard	⅜ in. to ½ in. thick	0.50	......	2.00	......
Gypsum Plaster		3.30	......	0.30	......
Gypsum Board (⅜ in.)	Plain or decorated	......	3.70	......	0.27
Gypsum Lath (⅜ in.) and Plaster	Plaster thickness assumed ½ in.	......	2.4	......	0.42
Insulating Board (½ in.)	Plain or decorated	......	0.66	......	1.52
Insulating Board Lath (½ in.) and Plaster	Plaster thickness assumed ½ in.	......	0.60	......	1.67
Insulating Board Lath (1 in.) and Plaster	Plaster thickness assumed ½ in.	......	0.31	......	3.18
Metal Lath and Plaster	Plaster thickness assumed ¾ in.	......	4.40	......	0.23
Plywood (⅜ in.)	Plain or decorated	......	2.12	......	0.47
Wood Lath and Plaster		......	2.50	......	0.40
MASONRY MATERIALS					
Brick	Adobe, assumed 4 in. thick	......	0.89	......	1.12
Brick	Common, assumed 4 in. thick	......	1.25	......	0.80
Brick	Face, assumed 4 in. thick	......	2.30	......	0.43
Cement Mortar		12.00	......	0.08	......
3 in. Clay tile (hollow)		......	1.28	......	0.78
4 in. Clay tile (hollow)		......	1.00	......	1.00
6 in. Clay tile (hollow)		......	0.64	......	1.57
8 in. Clay tile (hollow)		......	0.60	......	1.67
10 in. Clay tile (hollow)		......	0.58	......	1.72
12 in. Clay tile (hollow)		......	0.40	......	2.50
16 in. Clay tile (hollow)		......	0.31	......	3.23
Concrete	Light weight aggregate[b]	2.50	......	0.40	......
Concrete	Sand and gravel aggregate	12.00	......	0.08	......
3 in. Concrete blocks	Hollow, cinder aggregate	......	1.28	......	0.78
4 in. Concrete blocks	Hollow, cinder aggregate	......	1.00	......	1.00
8 in. Concrete blocks	Hollow, gravel aggregate	......	1.00	......	1.00
12 in. Concrete blocks	Hollow, gravel aggregate	......	0.80	......	1.25
8 in. Concrete blocks	Hollow, cinder aggregate	......	0.60	......	1.66
12 in. Concrete blocks	Hollow, cinder aggregate	......	0.53	......	1.88
8 in. Concrete blocks	Hollow, light weight aggregate[b]	......	0.50	......	2.00
12 in. Concrete blocks	Hollow, light weight aggregate[b]	......	0.47	......	2.13
Gypsum fiber concrete	87½ per cent gypsum and 12½ per cent wood chips	1.66	......	0.60	......
3 in. Gypsum tile	Hollow	......	0.61	......	1.64
4 in. Gypsum tile	Hollow	......	0.46	......	2.18
Stucco		12.50	......	0.08	......
Tile and Terrazzo	For flooring	12.00	......	0.08	......
Stone		12.50	......	0.08	......

[a] Conductance values for horizontal air spaces depend on whether the heat flow is upward or downward, but in most cases it is sufficiently accurate to use the same values for horizontal as for vertical air spaces.

[b] Expanded slag, burned clay or pumice.

TABLE 4. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 5 TO 18—Concluded

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

MATERIAL	DESCRIPTION	Conductivity or Conductance		Resistance	
				Per Inch Thickness	For Thickness Listed
		(k)	(C)	$\left(\frac{1}{k}\right)$	$\left(\frac{1}{C}\right)$
ROOFING MATERIALS					
Asbestos Shingles		....	6.00	....	0.17
Asphalt Shingles		....	6.50	....	0.15
Built-up Roofing	Assumed thickness ⅜ in.	....	3.53	....	0.28
Heavy Roll Roofing		....	6.50	....	0.15
Slate	Assumed ½ in.	10.00	20.00	0.10	0.05
Wood Shingles		....	1.28	....	0.78
SHEATHING					
Gypsum (½ in.)		....	2.82	....	0.35
Insulating Board (²⁵⁄₃₂ in.)		....	0.42	....	2.37
Plywood (⁵⁄₁₆ in.)		....	2.56	....	0.39
Fir or Yellow Pine (1 in.)	Actual thickness ²⁵⁄₃₂ in.	....	1.02	....	0.98
Fir, plus building paper	Actual thickness ²⁵⁄₃₂ in.	....	0.86	....	1.16
SURFACES					
Still air	Ordinary non-reflective materials, vertical	....	1.65	....	0.61
15 mph wind velocity	Ordinary non-reflective materials, vertical	....	6.00	....	0.17
WOODS					
Fir sheathing (1 in.) building paper and Yellow Pine lap siding		....	0.50	....	2.00
Maple or Oak		1.15	....	0.87	....
Yellow Pine or Fir		0.80	....	1.25	....

sion U_i of the insulated construction may be compared with the corresponding coefficient U without insulation. Attention is called to the necessity of applying the insulating material in accordance with the manufacturer's specification. The engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of proper evaluation, or improper installation may lead to unsatisfactory results.

Computed Heat Transmission Coefficients

Computed over-all heat transmission coefficients of many common types of building construction are given in Tables 5 to 18, inclusive, each coefficient being identified by a serial number except in Table 18. For example, the coefficient U of a brick veneer, frame wall with wood sheathing and ½-inch of plaster on gypsum lath is 0.27 (Wall No. 28-C in Table 5) and with 2-inches of blanket or bat insulation the coefficient would be 0.097 (No. 49-B in Table 6).

In the analysis of any wall construction for the purpose of calculating the over-all coefficient of heat transmission U, it is first necessary to determine the paths of heat flow; that is, whether they are parallel or series, or a combination of both. This is in accordance with the basic laws of heat transfer, which state that in parallel flow the conductances are additive while in series flow the resistances are additive. Likewise, in order to determine the total resistance for the wall, the conductance must be known.

The importance of this analysis cannot be over emphasized. This is especially true in wall constructions in which there are parallel paths of heat flow and one path has a high heat transfer while others have a low heat transfer.

The method of making this calculation can best be shown by the following example and Fig. 4. As this wall was tested by the hot box method

at the University of Minnesota a direct comparison can be made between calculated and tested values.

Example 1. Calculate the coefficient of heat transmission U for wall as shown in Fig. 4. Wall construction consists of two 4-inch concrete walls separated by a $2\frac{1}{2}$ in. space filled with insulation; $\frac{1}{4}$-in. diameter metal tie rods are imbedded a distance of 1-in. in each 4 in. concrete wall and spaced 9 in. vertically and 12 in. horizontally. Values of k are: insulation 0.30, concrete 12.00, tie rods 400.00.

In Fig. 4 the following paths of heat flow from plane A to plane F will be noted:

1. From A to B: One path through 3 in. of concrete.
2. From B to C: Two paths; (a) through 1 in. of tie rod and (b) through 1 in. of concrete.
3. From C to D: Two paths; (a) through $2\frac{1}{2}$ in. of tie rod and (b) through $2\frac{1}{2}$ in. of insulation.
4. From D to E: Two paths; (a) through 1 in. of tie rod and (b) through 1 in. of concrete.
5. From E to F: One path through 3 in. of concrete.

It will be noted that items 2 and 4 are paths of similar flow and could be treated as one. If equilibrium or steady state heat transfer is assumed, there will exist a tem-

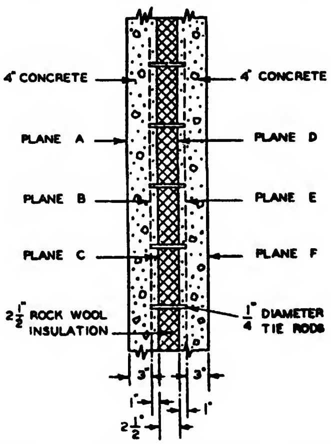


Fig. 4. Section of Concrete Wall Having Steel Tie Rods and Insulation

perature difference between the metal tie rod and the concrete and also between the metal tie rod and the insulating material. The rate of heat transfer between these materials is dependent upon their conductivity values and the temperature difference. As the conductivity of the metal tie rods is considerably higher than that of the concrete or insulating material, it cannot be assumed that the same rate of heat transfer takes place for all parallel paths. Likewise, an appreciable error would be made by assuming that no heat transfer takes place between the metal tie rod and the surrounding materials. Although the pattern of the isotherms is unknown, the following method of calculation does partially take into account the heat flow between the metal tie rods and its bounding materials.

Parallel Flow. The conductances through the areas of parallel heat flow may be determined as follows:

1. The area of each $\frac{1}{4}$-in. diameter tie rod is 0.0036 square feet and as the tie rods are spaced 9 inches vertically, and 12 inches horizontally, there will be $0.00036 \times \frac{4}{3} =$ 0.00048 square feet of tie rod to each square foot of wall area. Then from plane B to plane C, the conductance C_1 is

$$C_1 = \frac{0.00048}{1.0} \times \frac{400}{1.0} + \frac{0.99952}{1.0} \times \frac{12}{1.0} = 0.192 + 11.994 = 12.186$$

2. For Tie Rod and Insulation from plane C to plane D the conductance C_2 is

$$C_2 = \frac{0.00048}{1.0} \times \frac{400}{2.5} + \frac{0.99952}{1.0} \times \frac{0.30}{2.5} = 0.077 + 0.120 = 0.197$$

3. For Tie Rod and Concrete from plane D to plane E the conductance C_3 is

$$C_3 = \frac{0.00048}{1.0} \times \frac{400}{1.0} + \frac{0.99952}{1.0} \times \frac{12}{1.0} = 0.192 + 11.994 = 12.186$$

Series Flow. After the conductance values have been determined, the total resistance and U value for the wall can be determined as follows:

$$R_T = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + \frac{x_2}{k_2} + \frac{1}{f_0}$$

$$R_T = \frac{1}{1.65} + \frac{3.0}{12.0} + \frac{1}{12.186} + \frac{1}{0.197} + \frac{1}{12.186} + \frac{3.0}{12.0} + \frac{1}{6.0}$$

$$R_T = 0.606 + 0.250 + 0.0821 + 5.076 + 0.0821 + 0.250 + 0.167 = 6.513$$

$$U = \frac{1}{R_T} = \frac{1}{6.513} = 0.153 \text{ Btu per (hr) (sq ft) (F deg).}$$

The Hot Box test value, from University of Minnesota, for this wall corrected for a 15 mph wind velocity was $U = 0.150$ Btu per (hr) (sq ft) (F deg). The error between the calculated and test values would be

$$\frac{0.153 - 0.150}{0.150} \times 100 = 2 \text{ per cent.}$$

If the effect of the tie rods were omitted from the calculations, the overall U value would be 0.103. Although the percentage of area occupied by the tie rods per square foot of wall area is $\frac{0.0048}{1.0} \times 100 = 0.048$ per cent the error between the calculated and test values would be

$$\frac{0.150 - 0.103}{0.103} \times 100 = 45 \text{ per cent.}$$

In making the calculations for values of U shown in Tables 5 to 18, the following conditions have been assumed:

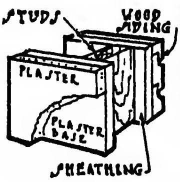
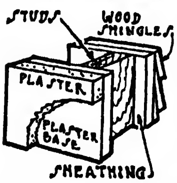
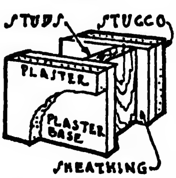
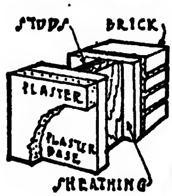
Equilibrium or steady-state heat transfer, eliminating effects of heat capacity.
Surrounding surfaces at ambient air temperatures.
Exterior wind velocity of 15 mph.
Surface emissivity of ordinary building materials = 0.83.
No correction for position or direction of heat flow. (Average coefficients used).
Air spaces are ¾ in. or more in width.
Variation of conductivity with mean temperature neglected.
Corrections for framing made on basis of parallel heat flow through 2 × 4 in. (nominal) studs, 16 in. on centers, the framing covering 15 per cent of wall area.
Actual thicknesses of lumber assumed to be as follows:

Nominal	*Actual*
1 in. (S-2-S)	25/32 in.
1½ in. (S-2-S)	1 5/16 in.
2 in. (S-2-S)	1⅝ in.
2½ in. (S-2-S)	2⅛ in.
3 in. (S-2-S)	2⅝ in.
4 in. (S-2-S)	3⅝ in.
Finish flooring, (maple or oak)	1 3/16 in.

Table 5. Coefficients of Transmission (U) of Frame Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

No Insulation Between Studs[a] (See Table 6)

Exterior Finish	Interior Finish	Type of Sheathing: Gypsum (½ in. thick) A	Ply-wood (5/16 in. thick) B	Wood[f] (25/32 in. thick) Bldg. Paper C	Insulating Board (25/32 in. thick) D	Wall Number
Wood Siding (Clapboard) 	Metal Lath and Plaster[b]	0.33	0.32	0.26	0.20	1
	Gypsum Board (3/8 in.) Decorated	0.32	0.32	0.26	0.20	2
	Wood Lath and Plaster	0.31	0.31	0.25	0.19	3
	Gypsum Lath (3/8 in.) Plastered[c]	0.31	0.30	0.25	0.19	4
	Plywood (3/8 in.) Plain or Decorated	0.30	0.30	0.24	0.19	5
	Insulating Board (½ in.) Plain or Decorated	0.23	0.23	0.19	0.16	6
	Insulating Board Lath (½ in.) Plastered[c]	0.22	0.22	0.19	0.15	7
	Insulating Board Lath (1 in.) Plastered[c]	0.17	0.17	0.15	0.12	8
Wood[d] Shingles 	Metal Lath and Plaster[b]	0.25	0.25	0.21	0.17	9
	Gypsum Board (3/8 in.) Decorated	0.25	0.25	0.21	0.17	10
	Wood Lath and Plaster	0.24	0.24	0.20	0.16	11
	Gypsum Lath (3/8 in.) Plastered[c]	0.24	0.24	0.20	0.16	12
	Plywood (3/8 in.) Plain or Decorated	0.24	0.24	0.20	0.16	13
	Insulating Board (½ in.) Plain or Decorated	0.19	0.19	0.17	0.14	14
	Insulating Board Lath (½ in.) Plastered[c]	0.19	0.18	0.17	0.13	15
	Insulating Board Lath (1 in.) Plastered[c]	0.14	0.14	0.13	0.11	16
Stucco 	Metal Lath and Plaster[b]	0.43	0.42	0.32	0.23	17
	Gypsum Board (3/8 in.) Decorated	0.42	0.41	0.31	0.23	18
	Wood Lath and Plaster	0.40	0.39	0.30	0.22	19
	Gypsum Lath (3/8 in.) Plastered[c]	0.39	0.39	0.30	0.22	20
	Plywood (3/8 in.) Plain or Decorated	0.39	0.38	0.29	0.22	21
	Insulating Board (½ in.) Plain or Decorated	0.27	0.27	0.22	0.18	22
	Insulating Board Lath (½ in.) Plastered[c]	0.26	0.26	0.22	0.17	23
	Insulating Board Lath (1 in.) Plastered[c]	0.19	0.19	0.16	0.14	24
Brick Veneer[e] 	Metal Lath and Plaster[b]	0.37	0.36	0.28	0.21	25
	Gypsum Board (3/8 in.) Decorated	0.36	0.36	0.28	0.21	26
	Wood Lath and Plaster	0.35	0.34	0.27	0.20	27
	Gypsum Lath (3/8 in.) Plastered[c]	0.34	0.34	0.27	0.20	28
	Plywood (3/8 in.) Plain or Decorated	0.34	0.33	0.27	0.20	29
	Insulating Board (½ in.) Plain or Decorated	0.25	0.25	0.21	0.17	30
	Insulating Board Lath (½ in.) Plastered[c]	0.24	0.24	0.20	0.16	31
	Insulating Board Lath (1 in.) Plastered[c]	0.18	0.18	0.15	0.13	32

[a] Coefficients not weighted; effect of studding neglected.
[b] Plaster assumed ¾ in. thick.
[c] Plaster assumed ½ in. thick.
[d] Furring strips (1 in. nominal thickness) between wood shingles and all sheathings except wood.
[e] Small air space and mortar between building paper and brick veneer neglected.
[f] Nominal thickness, 1 in.

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) OF FRAME WALLS WITH INSULATION BETWEEN FRAMING[a, b]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

COEFFICIENT WITH *NO* INSULATION BETWEEN FRAMING	COEFFICIENT WITH INSULATION BETWEEN FRAMING				NUMBER
	MINERAL WOOL OR VEGETABLE FIBERS IN BLANKET OR BAT FORM[c] (Thickness below)			3⅝ IN. MINERAL WOOL BETWEEN FRAMING[d]	
	1 IN.	2 IN.	3 IN.		
	A	B	C	D	
0.11	0.078	0.063	0.054	0.051	33
0.12	0.083	0.067	0.056	0.053	34
0.13	0.088	0.070	0.058	0.055	35
0.14	0.092	0.072	0.061	0.057	36
0.15	0.097	0.075	0.062	0.059	37
0.16	0.10	0.078	0.064	0.060	38
0.17	0.10	0.080	0.066	0.062	39
0.18	0.11	0.082	0.067	0.063	40
0.19	0.11	0.084	0.069	0.065	41
0.20	0.12	0.086	0.070	0.066	42
0.21	0.12	0.088	0.072	0.067	43
0.22	0.12	0.089	0.073	0.068	44
0.23	0.12	0.091	0.074	0.069	45
0.24	0.12	0.093	0.075	0.070	46
0.25	0.13	0.094	0.076	0.071	47
0.26	0.13	0.096	0.077	0.072	48
0.27	0.14	0.097	0.078	0.073	49
0.28	0.14	0.098	0.079	0.073	50
0.29	0.14	0.10	0.080	0.075	51
0.30	0.14	0.10	0.080	0.075	52
0.31	0.14	0.10	0.081	0.076	53
0.32	0.15	0.10	0.082	0.077	54
0.33	0.15	0.10	0.083	0.077	55
0.34	0.15	0.10	0.083	0.078	56
0.35	0.15	0.11	0.084	0.078	57
0.36	0.15	0.11	0.085	0.079	58
0.37	0.16	0.11	0.085	0.080	59
0.38	0.16	0.11	0.086	0.080	60
0.39	0.16	0.11	0.086	0.081	61
0.40	0.16	0.11	0.087	0.082	62
0.41	0.16	0.11	0.087	0.082	63
0.42	0.16	0.11	0.088	0.082	64
0.43	0.17	0.11	0.088	0.082	65
0.44	0.17	0.11	0.089	0.083	66

[a] This table may be used for determining the coefficients of transmission of frame constructions with the types and thicknesses of insulation indicated in Columns A to D inclusive between framing. Columns A, B and C may be used for walls, ceilings or roofs with only one air space between framing but are not applicable to ceilings with *no* flooring above. (See Table 11.) Column D is applicable to walls only. *Example:* Find the coefficient of transmission of a frame wall consisting of wood siding, $\frac{25}{32}$ in. insulating board sheathing, studs, gypsum lath and plaster, with 2 in. blanket insulation between studs. According to Table 5, a wall of this construction with *no insulation* between studs has a coefficient of 0.19 (Wall No. 4D). Referring to Column B above, it will be found that a wall of this value with 2 in. blanket insulation between the studs has a coefficient of 0.084.

[b] Coefficients corrected for 2 x 4 framing, 16 in. on centers—15 per cent of surface area.

[c] Based on one air space between framing.

[d] No air space.

Coefficients for frame construction are corrected for the effect of framing where such correction would increase the coefficients, but not where the correction would decrease the coefficients[7].

It should be noted that the effects of poor workmanship in construction and installation have an increasingly greater percentage effect on heat transmission as the coefficient becomes numerically smaller. Failure to meet design estimates may be caused by lack of proper attention to exact compliance with specifications, and a factor of safety may be employed as a precaution when it is judged desirable.

Roof Coefficients

Computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Values for roofs containing Spanish and French clay roofing tile are assumed the same as for slate roofs. Values for pitched roofs in Table 16 apply where the roof is over a heated attic or top floor so that the heat passes directly through the roof structure, including any interior finish material.

Combined Ceiling and Roof Coefficients

If the attic space between the ceiling and roof is unheated, the combined coefficient from room air below the ceiling to exterior air can be calculated from the following formula:

$$R_T = \frac{1}{U_{ce}} + \frac{1}{nU_r} \qquad (4)$$

and

$$U = \frac{1}{R_T} \qquad (5)$$

where

U = combined coefficient to be used with ceiling area
R_T = total resistance of ceiling and roof
U_{ce} = coefficient of transmission of ceiling
U_r = coefficient of transmission of roof
n = ratio of roof area to ceiling area

It should be noted that the over-all coefficient U should be multiplied by the ceiling area to determine heat loss and not by the roof area. Values of U_r and U_{ce} should be calculated using a value of 2.2 (the reciprocal of one-half the air space resistance) rather than 1.65 for the conductances of surfaces facing the attic, since the attic is equivalent to an air space.

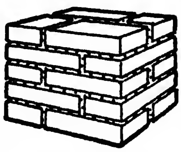
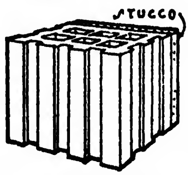
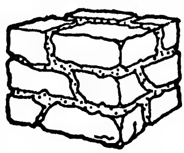
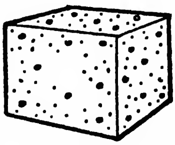
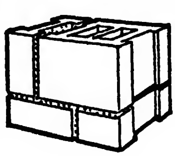
If the attic contains windows, dormers and vertical wall spaces and if their area is small compared to that of the roof, they may be considered part of the roof area. For accuracy, the sum of the coefficients of each individual section multiplied by its percentage of the total area should be used as U_r. Where attic wall areas are large it is preferable to estimate the attic temperature as illustrated in Chapter 14 and calculate the heat loss through the ceiling by multiplying the value of U_{ce} for the ceiling by the difference in temperature above and below the ceiling.

Basement Floor, Basement Wall and Concrete Slab Floor Coefficients

The heat transfer through basement walls and floors to the ground is dependent on the temperature difference between the air within and that

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Type of Masonry		Thickness of Masonry Inches	Interior Finish (Plus Insulation Where Indicated): Plain Walls—No Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[f] Furred[h]	Gypsum Board (⅜ in.) Decorated—Furred[h]	Gypsum Lath (⅜ in.) Plastered[g]—Furred[h]	Insulating Board (½ in.) Plain or Decorated—Furred[h]	Insulating Board Lath (½ in.) Plastered[g]—Furred[h]	Insulating Board Lath (1 in.) Plastered[g]—Furred[h]	Gypsum Lath[g] Plastered Plus 1 in. Blanket Insulation—Furred[h]	Wall Number
			A	B	C	D	E	F	G	H	I	
Solid[a] Brick		8	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	67
		12	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	68
		16	0.28	0.27	0.21	0.21	0.20	0.17	0.16	0.13	0.12	69
Hollow[b] Tile (Stucco Exterior Finish)		8	0.40	0.37	0.27	0.27	0.26	0.20	0.20	0.15	0.13	70
		10	0.39	0.37	0.27	0.27	0.26	0.20	0.19	0.15	0.13	71
		12	0.30	0.28	0.22	0.22	0.21	0.17	0.17	0.13	0.12	72
		16	0.24	0.24	0.19	0.19	0.18	0.15	0.15	0.12	0.11	73
Stone[c]		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	74
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	75
		16	0.49	0.45	0.31	0.31	0.29	0.22	0.22	0.16	0.14	76
		24	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	77
Poured Concrete[d]		6	0.79	0.71	0.42	0.41	0.39	0.27	0.26	0.19	0.16	78
		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	79
		10	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	80
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	81
Hollow Concrete Blocks		Gravel Aggregate										
		8	0.56	0.52	0.34	0.34	0.32	0.24	0.23	0.17	0.15	82
		12	0.49	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	83
		Cinder Aggregate										
		8	0.41	0.39	0.28	0.28	0.27	0.21	0.20	0.15	0.13	84
		12	0.38	0.36	0.26	0.26	0.25	0.20	0.19	0.15	0.13	85
		Light Weight Aggregate[e]										
		8	0.36	0.34	0.26	0.25	0.24	0.19	0.19	0.15	0.13	86
		12	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	87

[a] Based on 4 in. hard brick and remainder common brick.

[b] The 8 in. and 10 in. tile figures are based on two cells in the direction of heat flow. The 12 in. tile is based on three cells in the direction of heat flow. The 16 in. tile consists of one 10 in. and one 6 in. tile each having two cells in the direction of heat flow.

[c] Limestone or sandstone.

[d] These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.

[e] Expanded slag, burned clay or pumice.

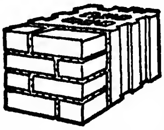
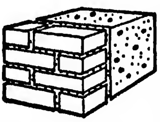
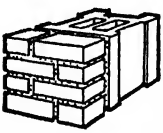
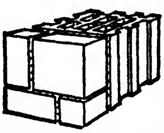
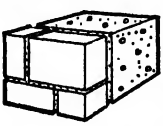
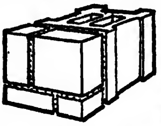
[f] Thickness of plaster assumed ¾ in.

[g] Thickness of plaster assumed ½ in.

[h] Based on 2 in. furring strips; one air space.

Table 8. Coefficients of Transmission (U) of Brick and Stone Veneer Masonry Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Typical Construction	Facing	Backing	Interior Finish (Plus Insulation Where Indicated): Plain Walls—no Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[e]—Furred[g]	Gypsum Board (⅜ in.). Decorated—Furred[g]	Gypsum Lath (⅜ in.) Plastered[f]—Furred[g]	Insulating Board (½ in.) Plain or Decorated—Furred[g]	Insulating Board Lath (½ in.) Plastered[f]—Furred[g]	Insulating Board Lath (1 in.) Plastered[f]—Furred[g]	Gypsum Lath Plastered[f] Plus 1 in. Blanket Insulation—Furred[g]	Wall Number
			A	B	C	D	E	F	G	H	I	
	4 in. Brick Veneer[a]	6 in. Hollow Tile[b]	0.35	0.34	0.25	0.25	0.24	0.19	0.18	0.14	0.13	88
		8 in. Hollow Tile[b]	0.34	0.32	0.25	0.24	0.23	0.19	0.18	0.14	0.13	89
		6 in. Concrete	0.59	0.54	0.35	0.35	0.33	0.24	0.23	0.17	0.15	90
		8 in. Concrete	0.54	0.50	0.33	0.33	0.31	0.23	0.23	0.17	0.15	91
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.44	0.41	0.29	0.29	0.28	0.21	0.21	0.16	0.14	92
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	93
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	94
	4 in. Cut Stone Veneer[a]	6 in. Hollow Tile[b]	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	95
		8 in. Hollow Tile[b]	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	96
		6 in. Concrete	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	97
		8 in. Concrete	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	98
		6 in. Concrete Blocks[c] (Gravel Aggregate)	0.47	0.44	0.30	0.30	0.29	0.22	0.21	0.16	0.14	99
		6 in. Concrete Blocks[c] (Cinder Aggregate)	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	100
		6 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.32	0.30	0.23	0.23	0.22	0.18	0.17	0.14	0.12	101

[a] Calculations based on ½ in. cement mortar between backing and facing except in the case of the concrete backing which is assumed to be poured in place.

[b] The hollow tile figures are based on two air cells in the direction of heat flow.

[c] Hollow concrete blocks.

[d] Expanded slag, burned clay or pumice.

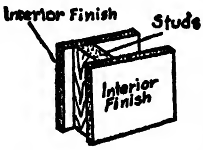
[e] Thickness of plaster assumed ¾ in.

[f] Thickness of plaster assumed ½ in.

[g] Based on 2 in. furring strips; one air space.

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF FRAME PARTITIONS OR INTERIOR WALLS[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



INTERIOR FINISH	SINGLE PARTITION (Finish on one side only of studs)	DOUBLE PARTITION (Finish on both sides of studs) — No insulation between studs	DOUBLE PARTITION (Finish on both sides of studs) — 1 in. Blanket[d] between studs. One air space.	Partition Number
	A	B	C	
Metal Lath and Plaster[b]	0.69	0.39	0.16	1
Gypsum Board (3/8 in.) Decorated	0.67	0.37	0.16	2
Wood Lath and Plaster	0.62	0.34	0.15	3
Gypsum Lath (3/8 in.) Plastered[c]	0.61	0.34	0.15	4
Plywood (3/8 in.) Plain or Decorated	0.59	0.33	0.15	5
Insulating Board (1/2 in.) Plain or Decorated	0.36	0.19	0.11	6
Insulating Board Lath (1/2 in.) Plastered[c]	0.35	0.18	0.11	7
Insulating Board Lath (1 in.) Plastered[c]	0.23	0.12	0.082	8

[a] Coefficients not weighted; effect of studding neglected.

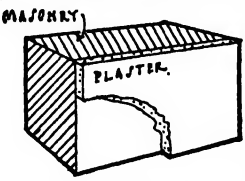
[b] Plaster assumed 3/4 in. thick.

[c] Plaster assumed 1/2 in. thick.

[d] For partitions with other insulations between studs refer to Table 6, using values in Column B of above table in left hand column of Table 6. *Example:* What is the coefficient of transmission (U) of a partition consisting of gypsum lath and plaster on both sides of studs with 2 in. blanket between studs? *Solution:* According to above table, this partition with no insulation between studs (No. 4B) has a coefficient of 0.34. Referring to Table 6, it will be found that a wall having a coefficient of 0.34 with no insulation between studs, will have a coefficient of 0.10 with 2 in. of blanket insulation between studs (No. 56B).

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY PARTITIONS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



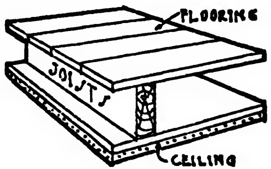
TYPE OF PARTITION		Thickness of Masonry (Inches)	TYPE OF FINISH — No Finish (Plain walls)	TYPE OF FINISH — Plaster One Side	TYPE OF FINISH — Plaster Both Sides[a]	Partition Number
			A	B	C	
Hollow Clay Tile		3	0.50	0.47	0.43	9
		4	0.45	0.42	0.40	10
Hollow Gypsum Tile		3	0.35	0.33	0.32	11
		4	0.29	0.28	0.27	12
Hollow Concrete Tile or Blocks	Cinder Aggregate	3	0.50	0.47	0.43	13
		4	0.45	0.42	0.40	14
	Light Weight Aggregate[b]	3	0.41	0.39	0.37	15
		4	0.35	0.34	0.32	16
Common Brick		4	0.50	0.46	0.43	17

[a] 2 in. solid plaster partition, U = 0.53.

[b] Expanded slag, burned clay or pumice.

Table 11. Coefficients of Transmission (U) of Frame Construction Ceilings and Floors

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference between the air on the two sides) and are based on still air (no wind) conditions on both sides.



Type of Ceiling	Insulation Between, or on Top of, Joists (No Flooring Above)												With Flooring[g] (On Top of Ceiling Joists)		Number
	None	Insulating Board on Top of Joists		Blanket or Bat Insulation[f] Between Joists[a]			Vermiculite Insulation Between Joists[a]			Mineral Wool Insulation Between Joists[a]			Single Wood Floor[b]	Double Wood Floor[c]	
		1/2 In.	1 In.	1 In.	2 In.	3 In.	2 In.	3 In.	4 In.	2 In.	3 In.	4 In.			
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
No Ceiling		0.37	0.24										0.45	0.34	1
Metal Lath and Plaster[d]	0.69	0.26	0.19	0.19	0.12	0.093	0.18	0.14	0.11	0.12	0.093	0.077	0.30	0.25	2
Gypsum Board (3/8 in.) Plain or Decorated	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.13	0.10	0.12	0.092	0.077	0.30	0.24	3
Wood Lath and Plaster	0.62	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	4
Gypsum Lath (3/8 in.) Plastered[e]	0.61	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	5
Plywood (3/8 in.) Plain or Decorated	0.59	0.24	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.23	6
Insulating Board (1/2 in.) Plain or Decorated	0.36	0.19	0.15	0.16	0.10	0.082	0.14	0.12	0.097	0.10	0.082	0.069	0.22[h]	0.19[h]	7
Insulating Board Lath (1/2 in.) Plastered[e]	0.35	0.19	0.15	0.15	0.10	0.081	0.14	0.11	0.096	0.10	0.081	0.068	0.21	0.18	8
Insulating Board Lath (1 in.) Plastered[e]	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.084	0.089	0.072	0.061	0.16	0.14	9

[a] Coefficients corrected for framing on basis of 15 per cent area, 2 in. x 4 in. (nominal) framing, 16 in. on centers.

[b] 25/32 in. yellow pine or fir.

[c] 25/32 in. pine or fir sub-flooring plus 13/16 in. hardwood finish flooring.

[d] Plaster assumed 3/4 in. thick.

[e] Plaster assumed 1/2 in. thick.

[f] Based on insulation in contact with ceiling and consequently no air space between.

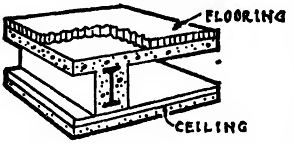
[g] For coefficients for constructions in Columns M and N (except No. 1) with insulation between joists, refer to Table 6. *Example:* The coefficient for No. 3-N of Table 11 is 0.24. With 2 in. blanket insulation between joists, the coefficient will be 0.093. (See Table 6.) (Column D of Table 6 applicable only for 3 5/8 in. joists.)

[h] For 25/32 in. insulating board sheathing applied to the underside of the joists, the coefficient for *single* wood floor (Column M) is 0.18 and for *double* wood floor (Column N) is 0.16. For coefficients with insulation between joists, see Table 6.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.

TYPE OF CEILING



TYPE OF CEILING	THICKNESS OF CONCRETE[g] (INCHES)	TYPE OF FLOORING: No Flooring (Concrete Bare) A	Tile[a] or Terrazzo Flooring on Concrete B	⅛ In. Asphalt Tile[b] Directly on Concrete C	Parquet[c] Flooring In Mastic on Concrete D	Double Wood Floor on Sleepers[d] E	NUMBER
No Ceiling	3	0.68	0.65	0.66	0.45	0.25	1
	6	0.59	0.56	0.58	0.41	0.23	2
	10	0.50	0.48	0.49	0.36	0.22	3
½ in. Plaster Applied to Underside of Concrete	3	0.62	0.59	0.60	0.43	0.24	4
	6	0.54	0.52	0.53	0.39	0.22	5
	10	0.46	0.44	0.45	0.34	0.21	6
Metal Lath and Plaster[e]—Suspended or Furred	3	0.38	0.37	0.37	0.30	0.19	7
	6	0.35	0.34	0.35	0.28	0.18	8
	10	0.32	0.31	0.32	0.26	0.17	9
Gypsum Board (⅜ in.) and Plaster[f]—Suspended or Furred	3	0.36	0.35	0.35	0.28	0.19	10
	6	0.33	0.32	0.33	0.27	0.18	11
	10	0.30	0.29	0.30	0.24	0.17	12
Insulating Board Lath (½ in.) and Plaster[f] Suspended or Furred	3	0.25	0.24	0.25	0.21	0.15	13
	6	0.23	0.23	0.23	0.20	0.15	14
	10	0.22	0.21	0.22	0.19	0.14	15

[a] Thickness of tile assumed to be 1 in.

[b] Conductivity of Asphalt Tile assumed to be 3.1.

[c] Thickness of wood assumed to be $\frac{13}{16}$ in.; thickness of mastic, ⅛ in. (k = 4.5). Col. D may also be used for concrete covered with carpet.

[d] Based on $\frac{25}{32}$ in. yellow pine or fir sub-flooring and $\frac{13}{16}$ in. hardwood finish flooring with an air space between sub-floor and concrete.

[e] Thickness of plaster assumed to be ¾ in.

[f] Thickness of plaster assumed to be ½ in.

[g] For other thicknesses of concrete, interpolate.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING

U = 0.10[a] Btu per (hour) (square foot) (Fahrenheit degree temperature difference between the ground and the air over the floor).

[a] Until more complete data are available, it is recommended that a coefficient of 0.10 be used for all types of concrete floors on the ground, with or without insulation. For basement wall below grade, use the same average coefficient (0.10). A lower ground temperature should, however, be used for walls than floors as explained in Chapter 14. For further data see A.S.H.V.E. RESEARCH REPORT No. 1213—Heat Loss Through Basement Walls and Floors, by F. C. Houghton, S. I. Taimuty, Carl Gutberlet and C. J. Brown (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 369).

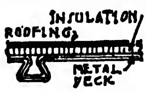
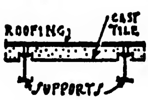
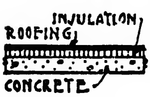
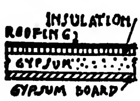
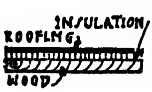
of the ground, on the material constituting the wall or floor, and on the conductivity of the surrounding earth. The conductivity of the earth will vary with local conditions and is usually unknown. Tests[8] at the A.S.H.V.E. Research Laboratory indicate a heat flow of approximately 2.0 Btu per (hour) (square foot) through an uninsulated concrete basement floor with a temperature difference of 20 F between ground temperature and the air temperature 6 in. above the floor. Based on this result, a coefficient of 0.10 Btu per (hour) (square foot) (Fahrenheit degree difference) is recommended for calculation where it is desirable to allow for the small basement floor heat loss, *e.g.* for heated basements.

For basement walls the same coefficient may be used, but due to closer proximity to the surface of the ground, the temperature difference for winter design conditions will be greater than for the floor. The test results

Table 14. Coefficients of Transmission (*U*) of Flat Roofs Covered with Built-up Roofing. No Ceiling—Under Side of Roof Exposed

(See Table 15 for Flat Roofs with Ceilings)

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Type of Roof Deck	Thickness of Roof Deck (Inches)	No Insulation	Insulation on Top of Deck (Covered with Built-Up Roofing)							Number
			Insulating Board (Thickness Below)				Corkboard (Thickness Below)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck[a] 		1.06	0.39	0.24	0.18	0.14	0.23	0.17	0.13	
Precast Cement Tile 	1⅝ in.	0.84	0.37	0.24	0.17	0.14	0.22	0.16	0.13	2
Concrete 	2 in.	0.82	0.36	0.24	0.17	0.14	0.22	0.16	0.13	3
	4 in.	0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	4
	6 in.	0.65	0.33	0.22	0.16	0.13	0.21	0.15	0.12	5
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in.	0.38	0.24	0.18	0.14	0.12	0.17	0.13	0.11	6
	3½ in.	0.31	0.21	0.16	0.13	0.11	0.15	0.12	0.10	7
Wood[c] 	1 in.	0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	8
	1½ in.	0.37	0.24	0.17	0.14	0.11	0.17	0.13	0.11	9
	2 in.	0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	10
	3 in.	0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	11

[a] Coefficients of transmission of bare corrugated iron (no roofing) is 1.50 Btu per (hour) (square foot of projected area) (Fahrenheit degree difference in temperature) based on an outside wind velocity of 15 mph.

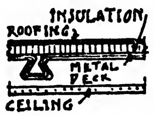
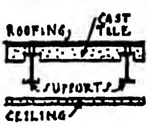
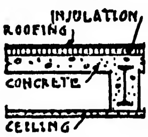
[b] 87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board.

[c] Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 15. COEFFICIENTS OF TRANSMISSION (U) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. WITH LATH AND PLASTER CEILINGS[a]

(See Table 14 for Flat Roofs with No Ceilings)

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	NO INSULATION	INSULATION ON TOP OF DECK (COVERED WITH BUILT-UP ROOFING)							NUMBER
			INSULATING BOARD (Thickness Below)				CORKBOARD (Thickness Below)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck 		0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11	12
Precast Cement Tile 	1⅝ in.	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11	13
Concrete 	2 in.	0.42	0.26	0.19	0.14	0.12	0.18	0.14	0.11	14
	4 in.	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	15
	6 in.	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	16
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in.	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097	17
	3½ in.	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091	18
Wood[c] 	1 in.	0.31	0.21	0.16	0.13	0.11	0.15	0.12	0.10	19
	1½ in.	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095	20
	2 in.	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092	21
	3 in.	0.18	0.14	0.12	0.10	0.087	0.11	0.095	0.082	22

[a] Calculations based on metal lath and plaster ceilings, but coefficients may be used with sufficient accuracy for gypsum lath or wood lath and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

[b] 87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board.

[c] Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 16. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)	WOOD SHINGLES (ON 1 x 4 WOOD STRIPS[c] SPACED 2 IN. APART)				ASPHALT SHINGLES OR ROLL ROOFING (ON SOLID WOOD SHEATHING)[c]				SLATE OR TILE[b] (ON SOLID WOOD SHEATHING)[c]				NUMBER
	INSULATION BETWEEN RAFTERS				INSULATION BETWEEN RAFTERS				INSULATION BETWEEN RAFTERS				
	None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			
		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.	
	A	B	C[a]	D[a]	E	F	G[a]	H[a]	I	J	K[a]	L[a]	
No Ceiling Applied to Rafters	0.48[f]	0.15	0.10	0.081	0.52[f]	0.15	0.11	0.084	0.55[f]	0.16	0.11	0.085	1
Metal Lath and Plaster[d]	0.31	0.14	0.10	0.081	0.33	0.15	0.10	0.083	0.34	0.15	0.10	0.083	2
Gypsum Board (3/8 in.) Decorated	0.30	0.14	0.10	0.080	0.32	0.15	0.10	0.082	0.33	0.15	0.10	0.083	3
Wood Lath and Plaster	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	4
Gypsum Lath (3/8 in.) Plastered[e]	0.29	0.14	0.10	0.079	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	5
Plywood (3/8 in.) Plain or Decorated	0.29	0.14	0.099	0.079	0.30	0.14	0.10	0.081	0.31	0.15	0.10	0.081	6
Insulating Board (1/2 in.) Plain or Decorated	0.22	0.12	0.090	0.072	0.23	0.12	0.091	0.074	0.24	0.13	0.092	0.074	7
Insulating Board Lath (1/2 in.) Plastered[e]	0.22	0.12	0.088	0.072	0.22	0.12	0.090	0.073	0.23	0.12	0.091	0.074	8
Insulating Board Lath (1 in.) Plastered[e]	0.16	0.10	0.078	0.064	0.17	0.10	0.079	0.065	0.17	0.10	0.080	0.066	9

[a] Coefficients corrected for framing on basis of 15 per cent area, 2 in. x 4 in. (nominal), 16 in. on centers.
[b] Figures in Columns I, J, K and L may be used with sufficient accuracy for rigid asbestos shingles on wood sheathing. Layer of slater's felt neglected.
[c] Sheathing and wood strips assumed 25/32 in. thick.
[d] Plaster assumed 3/4 in. thick.
[e] Plaster assumed 1/2 in. thick.
[f] No air space included in 1-A, 1-E or 1-I; all other coefficients based on one air space.

TABLE 17. COMBINED COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a] AND HORIZONTAL CEILINGS—BASED ON CEILING AREA[b]

Coefficients are expressed in Btu per (hour) (square foot of ceiling area) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

CEILING COEFFICIENT[f] (FROM TABLE 11)	TYPE OF ROOFING AND ROOF SHEATHING: Wood Shingles on Wood Strips[d]			Asphalt Shingles[c] or Roll Roofing on Wood Sheathing[e]			NUMBER
	No Roof Insulation (Rafters Exposed) (U_r = 0.48)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.22)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.16)	No Roof Insulation (Rafters Exposed) (U_r = 0.53)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.23)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.17)	
	A	B	C	D	E	F	
0.10	0.085	0.073	0.066	0.087	0.074	0.067	19
0.11	0.092	0.078	0.07	0.094	0.079	0.071	20
0.12	0.099	0.082	0.074	0.10	0.083	0.075	21
0.13	0.11	0.087	0.078	0.11	0.088	0.079	22
0.14	0.11	0.091	0.081	0.11	0.093	0.083	23
0.15	0.12	0.096	0.084	0.12	0.097	0.086	24
0.16	0.13	0.10	0.087	0.13	0.10	0.089	25
0.17	0.13	0.10	0.090	0.13	0.10	0.092	26
0.18	0.14	0.11	0.093	0.14	0.11	0.095	27
0.19	0.14	0.11	0.095	0.15	0.11	0.098	28
0.20	0.15	0.11	0.098	0.15	0.12	0.10	29
0.21	0.15	0.12	0.10	0.16	0.12	0.10	30
0.22	0.16	0.12	0.10	0.17	0.12	0.11	31
0.23	0.16	0.12	0.10	0.17	0.12	0.11	32
0.24	0.17	0.13	0.11	0.18	0.12	0.11	33
0.25	0.17	0.13	0.11	0.18	0.13	0.11	34
0.26	0.18	0.13	0.11	0.19	0.13	0.11	35
0.27	0.18	0.13	0.11	0.19	0.13	0.12	36
0.28	0.19	0.14	0.12	0.19	0.14	0.12	37
0.29	0.19	0.14	0.12	0.20	0.14	0.12	38
0.30	0.20	0.14	0.12	0.20	0.14	0.12	39
0.34	0.21	0.15	0.12	0.22	0.15	0.13	40
0.35	0.22	0.15	0.13	0.22	0.15	0.13	41
0.36	0.22	0.15	0.13	0.23	0.15	0.13	42
0.37	0.23	0.15	0.13	0.23	0.16	0.13	43
0.45	0.25	0.17	0.13	0.26	0.17	0.14	44
0.59	0.29	0.18	0.14	0.30	0.19	0.15	45
0.61	0.29	0.18	0.15	0.31	0.19	0.15	46
0.62	0.30	0.19	0.15	0.31	0.19	0.15	47
0.67	0.31	0.19	0.15	0.33	0.20	0.16	48
0.69	0.31	0.19	0.15	0.33	0.20	0.16	49

[a] Calculations based on ⅓ pitch roof (n 1.2) using the following formula:

$U = \frac{U_r \times U}{}$ U combined coefficient to be used with ceiling area.
U_r coefficient of transmission of the roof.
coefficient of transmission of the ceiling.
the ratio of the area of the roof to the area of the ceiling.

[b] Use ceiling area (not roof area) with these coefficients.

[c] Coefficients in Columns D, E and F may be used with sufficient accuracy for tile, slate and rigid asbestos shingles on wood sheathing.

[d] Based on 1 x 4 in. strips spaced 2 in. apart.

[e] Sheathing assumed 25/32 in. thick.

[f] Values of U_{ce} to be used in this column may be selected from Table 11.

Table 18. Coefficients of Transmission (U) of Doors, Windows, Skylights and Glass Block Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in the temperature between the air inside and outside of the door, window, skylight or wall) and are based on an outside wind velocity of 15 mph.

Section A. Windows and Skylights		Single	Double	Triple
	U	1.13[ac]	0.45[ae]	0.281[ae]

Section B. Solid Wood Doors[bc]	Nominal Thickness Inches	Actual Thickness Inches	U Exposed Door	U^d With Glass Storm Door
	1	25/32	0.69	0.42
	1 1/4	1 1/16	0.59	0.38
	1 1/2	1 5/16	0.52	0.35
	1 3/4	1 3/8	0.51	0.35
	2	1 5/8	0.46	0.32
	2 1/2	2 1/8	0.38	0.28
	3	2 5/8	0.33	0.25

Section C. Hollow Glass Block Walls	Description	U Still Air Both Sides	U Still Air Inside 15 mph Outside
	Smooth surface glass blocks 7 3/4 x 7 3/4 x 3 7/8 in. thick	0.40	0.49
	Ribbed surface glass blocks 7 3/4 x 7 3/4 x 3 7/8 in. thick	0.38	0.46

[a] See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[b] Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$.

[c] It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per (hour) (square foot) (degree difference between inside and outside air temperatures).

[d] These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors.

[e] Air spaces assumed to be ¾ in. or more in width.

indicate a unit area heat loss, at mid-height of the basement wall, approximately twice that of the same floor area.

For concrete slab floors laid in contact with the ground at grade level, recent tests[9] indicate that for small floor areas (equal to that of a house 25 ft square) the heat loss may be calculated as proportional to the length of exposed edge rather than total area. This amounts to 0.81 Btu per (hour) (lineal foot of exposed edge) (Fahrenheit degree difference between the inside air temperature and the average outside air temperature). It should be noted that this may be appreciably reduced by insulating the edges of the floor from the abutting wall.

CALCULATING SURFACE TEMPERATURES

In many heating and cooling load calculations it is necessary to determine the inside surface temperature or the temperature of the surfaces within the structure. As the resistance of any path of heat flow is expressed in Fahrenheit degrees per (Btu) (Hour) (Square Foot) the resistances through any two paths of heat flow would be proportional to the temperature drop through these paths and can be expressed as follows:

$$\frac{R_1}{R_2} = \frac{(t_i - t_x)}{(t_i - t_o)} \qquad (6)$$

where

R_1 = the resistance from the inside air to any point in the structure at which the temperature is to be determined.

R_2 = the over-all resistance of the wall from inside air to outside air.
t_i = inside air temperature
t_x = temperature to be determined
t_o = outside air temperature

Example 2. Determine the inside surface temperature for a wall having an over-all coefficient of heat transmission U = 0.25, inside air temperature 70 F, outside air temperature −20 F. Solution:

$$R_1 = \frac{1}{f_i} = \frac{1}{1.65} = 0.606$$

$$R_2 = \frac{1}{U} = \frac{1}{0.25} = 4.00$$

Then, by Equation 1

$$\frac{0.606}{4.00} = \frac{70 - t_x}{70 - (-20)}$$

$$t_x = 56.4 \text{ F.}$$

The same procedure can be used for determining the temperature at any point within the structure.

WATER VAPOR AND CONDENSATION

Water vapor is an important factor in the design and construction of many types of buildings and in processes where controlled air conditions are essential. It must often be considered in the construction of residences, or public buildings located in cold climates and, to a lesser extent, in those located in warm climates. It is extremely important to consider the moisture problem in the construction of cold storage and low temperature rooms. Manufacturing processes which require special humidity often require buildings designed with consideration for the effect of moisture on the building. There are, likewise, many processes which in themselves create moisture problems which become the major consideration in either the construction of the building or the method of plant operation.

These water vapor problems, being present to a greater or lesser extent in the majority of heating, cooling and air conditioning processes, make it necessary to understand the laws governing water vapor and its relation to air conditioning processes as well as its effect on different types of structures.

Water Vapor

The theory governing water vapor is well known and yet it is too often overlooked or given scant consideration in the construction of buildings and the layout of air conditioning processes. Water vapor is present in all air; it occupies the space; and has the same properties that it would have if the air were not present. It is steam at low pressure and temperature. Thus, in an air vapor mixture at 80 F, the density of the water vapor may be 0.00158 lb per cu ft providing that it is saturated and the vapor pressure would be 1.0323 inches of mercury. These are the same conditions that would be obtained in a cubic foot of saturated steam at 80 F, and it is spoken of as 100 per cent relative humidity air. If this same volume of air contained only one-half of the original moisture or 0.00079 lb per cu ft, it would be only 50 per cent saturated, or the relative humidity would be 50.1 per cent. In the first case, the vapor pressure would be 1.0323 inches of mercury and in the second case, it would be 50.1 per cent of this or 0.5172. In the first case, the vapor would be saturated and the dew point,

or condensing temperature, would be 80 F. In the second case, the vapor would be superheated and the dew point, or condensing temperature, would be about 60.2 F. When the vapor in a space is cooled down either by contact with cold surfaces or otherwise to a temperature below its dew point temperature, some of the vapor will be condensed and form either free water or frost depending upon the temperature. In condensing, the vapor will give up heat and be deposited as either free water or frost depending upon the condensing temperature.

Surface Condensation

If water vapor comes in contact with surfaces of materials which have temperatures below its dew point temperature, condensation will take place. This process is seen in the accumulation of moisture on surface of a cold glass of water, or on cold water pipes. In cold storage systems con-

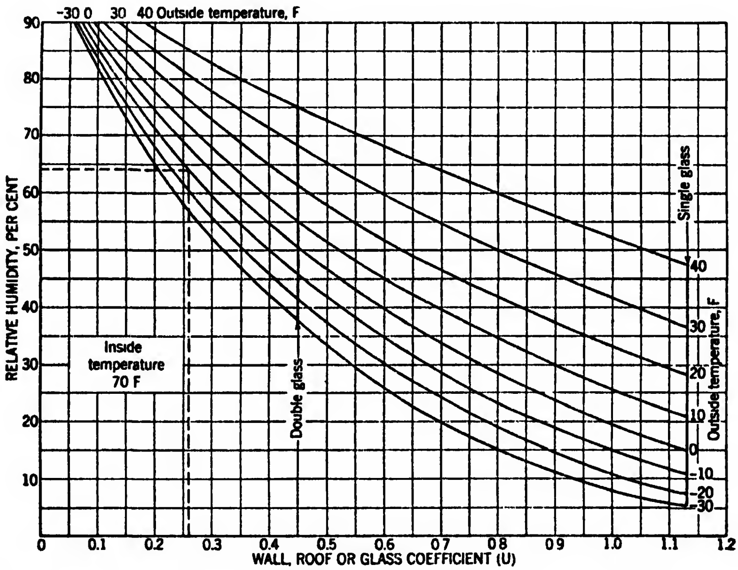


Fig. 5. Permissible Relative Humidities for Various Transmission Coefficients

densation occurs on the cooling surfaces. In the winter condensation collects on interior surfaces of windows, cold closet walls or attic surfaces, and sometimes it occurs within cold sections of the structure. The extent of condensation in these places depends upon the surface temperature of the material and the dew point temperature of the vapor in contact with these materials.

In residences and public buildings the surface condensation problem is usually more important from the viewpoint of its nuisance and deteriorating effect on the structure than it is from the standpoint of addition to the cooling load. In cold storage plants and refrigerating processes, it often has a material effect on the structure, the cooling load and the operating efficiency.

For residences and other similar buildings, condensation is usually dependent upon surface temperatures and upon the dew point temperature of the air in contact with these surfaces. For any set of temperature and

humidity conditions, there is a definite relation between the condensation possibilities and the insulation of exposed parts of the structure. Limiting maximum relative humidities for walls, roofs or glass, having transmission coefficients up to 1.2 Btu for outside temperatures, from −30 F to 40 F, and for 70 F inside temperature may be obtained from Figure 5.

Vapor Transmission Through Materials

The condensation of moisture within buildings is not limited to visible surfaces such as wall surfaces and glass surfaces. Vapor will pass through certain materials very readily and may penetrate into exterior or cold walls and come in contact with material within these structures having a temperature below the dew point temperature of the vapor and thus form moisture or frost within the wall. This moisture tends to accumulate over long periods of time without being observed. It is this accumulation of interior and unobserved condensation that causes the greatest difficulty in many long-range processes. The property of a material to transmit vapor is known as its *vapor permeability*. The theory covering vapor transmission through materials leads to the following formula:

$$W = \mu A\ (P_1 - P_2) \tag{7}$$

where

W = total moisture vapor flow, grains per hour through the wall.
μ = permeability, grains per (hour) (square foot) (unit vapor pressure differential).
A = area of the wall, square feet.
P_1 = vapor pressure on the humid side of the wall, and
P_2 = vapor pressure on the other side of the wall, both in units consistent with the pressure units of the transmission coefficient.

The over-all moisture transfer coefficient for a wall consisting of a combination of several materials in series may be calculated by combining the permeabilities (μ_1, μ_2, μ_3, etc.) of the individual materials according to the formula:

$$\mu = \frac{1}{\frac{1}{\mu_1} + \frac{1}{\mu_2} + \frac{1}{\mu_3} + \cdots \frac{1}{\mu_n}} \tag{8}$$

In the application of Equation 8 it is assumed that the permeability is directly proportional to the vapor pressure drop between two different planes, and that the resistance to vapor is additive for several materials in series. This theory may apply so long as the vapor remains in the vapor state. In most cases, however, there is a change in temperature throughout the structure and the vapor may change to a liquid or even a solid, and thus completely change the mechanism by which it is transferred through the material. Furthermore, many materials are hygroscopic and vapor is absorbed somewhat in the proportion to the relative humidity and not directly proportional to the pressure of the vapor in contact with the material. A further point to be considered is that the vapor pressure or dew point temperature drop per degree of temperature drop is much greater in high temperature than it is in low temperature ranges. Due to the uncertainties as to the exact mechanism for the transfer of vapor through various types of structures, the ***application of a theory which parallels the theory of heat transmission should be used with caution.***

There are several methods for determining the vapor permeability of materials. While a lengthy discussion of these methods cannot be under-

taken here, it may be said that there is not complete agreement in the results obtained by the different methods nor regarding a single standard to be used. Notwithstanding the uncertainties as to the theory and the lack of complete agreement as to test methods for measuring vapor permeabilities, the requirements are well understood and the relative values of certain types of materials have been established with sufficient precision to indicate their vapor permeability. Some values which may be used as a guide are given in Table 19.

TABLE 19. PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR

GROUP	MATERIAL	PERMEABILITY GRAINS PER (SQ FT) (HR) (INCH HG)
1[a]	Plaster base and plaster, ¾ in.	14.7
	Fir sheathing, ¾ in.	2.9
	Waterproof paper[b]	49.1
	Pine lap siding	4.9
	Paint film	3.4
	Sugar cane fiberboard, ¾ in.	12.5
	Brick masonry, 4 in.	1.1
2[c]	Foil-surfaced reflective insulation, double-faced	0.08 to 0.13
	Roll roofing—smooth, 40 to 65 lb per roll 108 sq ft	0.13 to 0.17
	Duplex or laminated papers, 30-30-30	1.37 to 2.58
	Duplex or laminated papers, 30-60-30	0.52-0.86
	Duplex paper, coated with metallic oxides	0.52-1.29
	Insulation backup paper, treated	0.86-3.42
	Plaster, wood lath	11.00
	Plaster, 3 coats of lead and oil	3.68 to 3.84
	Plaster, 2 coats of aluminum paint	1.15
	Plaster, fiberboard or gypsum lath	19.73 to 20.57
	Plywood, ⅜ in., 5-ply Douglas fir	2.67 to 2.74
	Plywood, 2 coats of asphalt paint	0.43
	Plywood, 2 coats of aluminum paint	1.29
	Gypsum lath with metallic aluminum backing	0.09-0.39
	Insulating lath and sheathing, board type	25.68 to 34.27
	Insulating sheathing, surface-coated	3.03 to 4.36
	Insulating cork blocks, 1 in.	6.19
	Mineral wool, unprotected, 4 in.	29.07
	Sheathing paper, asphalt impregnated, glossy	0.17-2.05

[a] Calculating Vapor and Heat Transfer Through Walls, by L. G. Miller (*Heating and Ventilating* 35, No. 11, 56, November, 1938).
[b] Light weight slaters felt used to keep rain from drifting through. Not used as a vapor barrier.
[c] How to Overcome Condensation in Building Walls and Attics, by L. V. Teesdale (*Heating and Ventilating*, Vol. 36, No. 4, April, 1939).

Water-proofed building papers are listed in Federal Specifications UU-P-147, May 24, 1948, according to water vapor resistance required as:

Class A. For uses where a high degree of water-vapor resistance is required.

Class B. For uses where only a moderate degree of water-vapor resistance or high water resistance is required.

Class C. For uses where only a moderate degree of water resistance is required.

Class D. For uses where high permeability to water vapor is required.

Detail requirements in these specifications are given as follows:

Class A paper shall have a minimum tensile strength in each direction of either 35 pounds per inch width or 20 pounds per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 24 hours and a maximum water-vapor permeability of 4 grams per square meter per 24 hours.

Class B paper shall have a minimum tensile strength in each direction of either 35 pounds per inch width or 20 pounds per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 16 hours, and a maximum water-vapor permeability of 6 grams per square meter per 24 hours.

Class C paper shall have a minimum tensile strength in each direction of either 35 pounds per inch width or 20 pounds per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 8 hours.

Class D paper shall have a minimum tensile strength in each direction of 20 pounds per inch width. The paper shall have a minimum water resistance of 10 minutes and a minimum water-vapor permeability of 35 grams per square meter per 24 hours.

The method of test used to determine permeability as specified is the following:

The test specimen having an area of at least 50 square centimeters, shall be sealed on the mouth of a dish containing calcium chloride. The seal shall be made with wax composed of 60 per cent refined amorphous wax and 40 per cent of refined crystalline paraffin wax. The dish shall be exposed to an atmosphere of 73 F $\pm$ 3.5 deg and 50 $\pm$ 2 per cent relative humidity until a constant rate of gain in the weight of the dish is attained. The average constant weight of gain for at least four test specimens shall be reported as the water-vapor permeability of the material in terms of grams per square meter, per 24 hours. Both sides of the material, in equal number, shall be exposed towards the calcium chloride.

Surface Condensation Control

Since surface condensation is caused by water vapor coming into contact with the surfaces having temperatures below its dew point temperature, the obvious remedy is, first, to reduce as far as practicable the dew point temperatures of the surrounding vapors, and, second, to increase the temperatures of the surfaces with which these vapors may come in contact. The control of the dew point is usually an operating problem. It may be lowered by giving attention to source of the moisture and eliminating it before it comes in contact or mixes with the air in the space. It may also be effectively reduced by ventilation or by some moisture absorption process. The control of vapor formation and its elimination from the space as soon as possible is one of the first requirements in most condensation problems. It is often the complete remedy.

The temperatures of the surfaces with which the vapor comes in contact may be increased by adding insulation to outside walls, by double glazing of windows, by circulation of warmer air over the surface, or perhaps by direct heating of the surfaces. The most expedient method of overcoming surface condensation difficulty will depend upon special conditions surrounding the problem. This is a construction rather than an operating problem.

Control of Condensation Within Structure

Since condensation within the structure is really surface condensation transferred to the interior parts of the structure, the same precautions as to humidity control should be observed as for surface condensation. In addition to this, however, the structure must be built to prevent vapor from getting to the interior sections of a wall. A wall which is apt to have a cold interior section should be constructed with a vapor-resisting material on its warm surface.

There are many types of materials and methods of construction which may be used to vapor proof the interior surfaces of cold walls. Vapor resistant membrane materials are often built into the wall near the warm surface. In wood frame walls they may be applied to the inside surface of the studs. They are sometimes attached to the warm side of insulating materials or they may be applied on the cold side of plaster base materials. There are several types of vapor resistant papers in combination with metal foils which may be used. To be effective these barriers should have a reasonably high resistance to the passage of vapor and should be so applied that they are continuous and unbroken.

The interior construction of the wall may also be made of vapor resistant

material, or some vapor resistant coating may be applied to the inner or warm surface of the wall.

In applying vapor resistance to a wall, there are certain fundamental principles which should be followed. First, the vapor barrier should be placed as near to the warm surface of the wall as practicable. Second, it should be continuous with no direct openings through the barrier. If membrane barriers are used back of the plaster of interior finish, the joints should be lapped over some solid framing member and not between the studs or in similar places. Usually a two-inch lap over a framing member will make a sufficiently tight joint when the interior finish is applied. Such a lap, however, without backing would not be adequate. All openings for electrical fixtures and joints around window and door casings should be carefully sealed.

Since, in applying vapor barriers the primary purpose is to prevent water vapor from entering the warm side of the wall the barrier in order to be effective must be placed near the warm side and all joints must be sufficiently tight to prevent direct leakage of the vapors. The limiting permeability for a material which may be considered as a barrier, will depend upon the requirements. For ordinary residential work, it has generally been considered that a material having a permeability of one grain of moisture per (sq ft) (hr) (in. Hg of vapor pressure) difference across the barrier is adequate. There are cases, however, in residential construction where a barrier having a permeability of 1.00 would not be sufficient and there are also many industrial applications in which a very much higher vapor resistance is required. The best time to vapor proof a building is during its construction. After the building is completed the remedies are limited largely to operational control and surface treatment of the structure.

Ventilation of Structure

Condensation difficulties may often be eliminated by lowering the dew point temperature or the relative humidity by ventilation. It is much more practicable to apply ventilation in open spaces than it is in interior parts of the structure. For a wall construction it is far better to seal the warm surface so that the vapor cannot enter, than it is to try to ventilate the vapor out of the wall once it has entered. Wherever possible, it is preferable to eliminate the moisture at its source rather than to rely on ventilation.

Condensation on the interior surface of cold attic walls may be eliminated by ventilation. However, in new construction and in other places where practicable, it is far better to use vapor barriers and other means to prevent the vapors from entering the attic space. Ventilation is often uncertain in its effect and, furthermore, it is a source of some heat loss. Where ventilation is used for attics or other parts of a building, precautions must be taken to see that the air is adequately distributed throughout the space to be ventilated. No fixed amount can be given for the ventilation required but for the ordinary home with gravity attic ventilation, the inlet and outlet openings should be well distributed and the total area of each should be one quarter square inch per square foot of floor. These openings should be distributed with due regard to the type of construction, outside wind velocities, and all factors which affect the circulation of air. The conditions are so varied that no hard and fast rules can be set down which will cover all cases. The best defense against condensation on attic walls and other similar surfaces is to prevent the vapors from entering these spaces. Ven-

tilation is a precaution but not the best direct solution of most condensation problems.

REFERENCES

[1] Standard Method of Test for Thermal Conductivity by Means of the Guarded Hot Plate, sponsored by A.S.H.V.E., *A.S.T.M.*, *A.S.R.E.*, and *N.R.C.* and approved as a Tentative Code by A.S.H.V.E. and *A.S.T.M.* in 1942 (*A.S.T.M.* designation C-177-42T, Approved 1945).

[2] Heat Transmission Through Building Materials, by F. B. Rowley and A. B. Algren (University of Minnesota, *Engineering Experiment Station Bulletin* No. 8, p. 11).

[3] Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 513).

[4] Radiation Corrections for Basic Constants Used in the Design of All Types of Heating Systems, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 213).

[5] Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 351).

[6] Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 109).

[7] Effect of Studs and Joists on Heat Flow Through Frame Walls and Ceilings, by Paul D. Close (*Heating, Piping and Air Conditioning*, October, 1943, p. 529).

[8] A.S.H.V.E. RESEARCH REPORT No. 1213—Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 369).

[9] Measurements of Heat Losses from Slab Floors, by R. S. Dill, Wm. C. Robinson and H. E. Robinson (*National Bureau of Standards*, Building Materials and Structures Report BMS 103).

[10] Methods of Moisture Control and Their Application to Building Construction, by F. B. Rowley, A. B. Algren and C. E. Lund (University of Minnesota *Engineering Experiment Station Bulletin* No. 17).

Moisture Migration: A Survey of Theory and Existing Knowledge, by P. F. McDermott (*Refrigerating Engineering*, August 1941, p. 103).

[11] Permissible Relative Humidities in Humified Buildings, by Paul D. Close (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1939, p. 766).

[12] Condensation Within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 95).

BIBLIOGRAPHY

A.S.H.V.E. RESEARCH REPORTS:

No. 852—Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 123).

No. 895—Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 301).

No. 914—Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 33).

No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

No. 964—The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 329).

No. 996—Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 413).

No. 1026—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 33).

No. 1048—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Robert Lander (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 33).

Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 285).

Condensation of Moisture and Its Relation to Building Construction and Operation, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 231).

A Theory Covering the Transfer of Vapor Through Materials, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 545).

Thermal Conductivity of Wood, by J. D. MacLean (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 323).

The Specific Heat of Thermal Insulating Materials, by G. B. Wilkes and C. O Wood (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 493).

Heat Loss Studies in Four Identical Buildings to Determine the Effect of Insulation, by D. B. Anderson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 471).

Simultaneous Heat and Vapor Transfer Characteristics of an Insulating Material, by F. G. Hechler, E. R. McLaughlin and E. R. Queer (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 505).

Comparative Resistance to Vapor Transmission of Various Building Materials, by L. V. Teesdale (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 124).

The Diffusion of Water Vapor Through Various Building Materials, by J. D. Babbitt (*Canadian Journal of Research*, Vol. 17, February, 1939, p. 15).

Effect of Ceiling Insulation upon Summer Comfort, by T. D. Phillips (*National Bureau of Standards*, Report BMS52, July 1, 1940).

Moisture Condensation in Building Walls, by H. W. Wooley (*National Bureau of Standards*, Report BMS63, December 14, 1940).

Heat Transmission Through Building Materials, by F. B. Rowley and A. B. Algren (University of Minnesota *Engineering Experiment Station Bulletin* No. 8).

Building Insulation, by P. D. Close (*American Technical Society*, Chicago, 1945).

CHAPTER 7

PERFORMANCE OF AIR HEATING AND COOLING COILS

Performance of Heating and Dry Cooling Coils, Over-all Coefficient of Heat Transfer, Performance of Dehumidifying Coils, External Film Coefficient, Internal Film Coefficient, Determining Size of Cooling Coil

THE surfaces described in this chapter are for heating or cooling an air stream under the conditions of forced convection. Such surfaces may be made up of a number of banks of tubes assembled in the field or the entire assembly may be factory constructed. They may be made of either bare or finned tubing, but regardless of their construction they are generally referred to as *coils*. A description of the types, application, and selection of such coils is given in Chapter 25. Therefore, this chapter is confined to the theoretical considerations which affect the calculation of the performance of these coils.

HEATING AND DRY COOLING COILS

The performance of heating and dry cooling coils depends in general upon:

1. The over-all coefficient of heat transfer from the fluid within the coil to the air it heats or cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$q_t = U \times (MTD) \times A \times N \qquad (1)$$

where

q_t = total heat transferred by the coil, Btu per (hour) (square foot of coil face area).

U = over-all coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degree temperature difference between the fluid within the coil and the air flowing over the coil).

MTD = mean temperature difference, Fahrenheit degrees, between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference.)

A = external surface area of the given coil, square feet per (square foot of coil face area) (row of coil depth).

N = number of rows of coil depth.

Over-all Coefficient of Heat Transfer

Of all factors affecting the performance of heating or dry cooling coils, the over-all coefficient of heat transfer is the most difficult to determine as

it is influenced by several factors which depend upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the over-all heat transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference).
2. The coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.
3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference).

These three individual coefficients acting in series result in an over-all coefficient of heat transfer in accordance with the basic laws given in Chapters 5 and 6. For a *bare pipe* coil the over-all coefficient of heat transfer, whether for heating or for cooling (without dehumidification), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{h_r} + \frac{L}{k} + \frac{1}{h_a}} \qquad (2)$$

where

U = over-all coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and fluid within the coil).

h_r = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference between that surface and the average fluid temperature).

h_a = film coefficient of heat transfer between air and the external surface of the coil, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between the mass of air and the external surface).

k = conductivity of material from which the bare pipe is constructed, Btu per (hour) (square foot) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L/k in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L/k in typical bare pipe heating or cooling coils seldom exceeds 1 to 2 per cent of the over-all coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{h_a}} \qquad (3)$$

For finned coils the formula[1] for the over-all coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\dfrac{R}{h_r} + \dfrac{1}{\eta h_a}} \tag{4}$$

in which the term η, called the *fin efficiency*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface. In the discussions which follow, coefficients h_r and ηh_a will be considered separately, and also various ways of combining them will be outlined.

The performances of all heating and dry cooling coils are influenced by

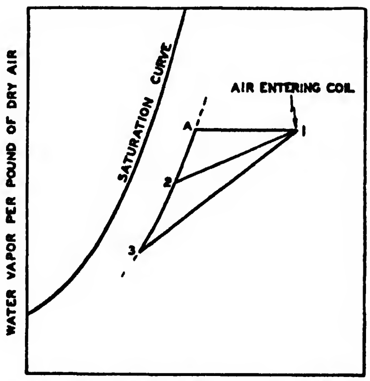


FIG. 1. PERFORMANCE OF DEHUMIDIFYING COIL

these same factors. But, when cooling coils operate wet or act as dehumidifying coils, the performance cannot be predicted on the basis of over-all coefficients and an analysis must be made on the basis of individual film coefficients as will be explained.

PERFORMANCE OF DEHUMIDIFYING COILS

When a cooling coil operates with a surface temperature which is below the dew-point of the air entering the coil, moisture is condensed and the air leaves the coil with a humidity ratio lower than it had when it entered the coil. To understand the performance of surface coils under such conditions, assume that air enters a cooling coil at conditions corresponding to point 1 in Fig. 1. As long as the surface temperature of the coil is above the dew-point, the air is cooled without dehumidification, and its condition leaving the coil will be somewhere on line 1-A. Its exact position on this line depends on the air velocity and the external film coefficient as well as upon the surface temperature. When the surface temperature just equals the dew-point, the air leaves with conditions represented by point A. If the surface temperature is below the dew-point, condensation takes place, and the air has a final condition somewhere along the line A-2-3 which is a

line at a constant horizontal distance from the saturation curve. It should be understood that the line 1-A-2-3 is not intended to represent the path of the condition of the air as it passes through the coil from row to row. It is simply the path traced by the exit air conditions as the surface temperature is gradually reduced with other conditions remaining constant[2].

In the process of dehumidification, since heat is being transferred to the coil surface by two different mechanisms, (convection and condensation), it is evident that an over-all coefficient of heat transfer cannot be determined by the same method used for heating and for dry cooling coils. However, if it is assumed that the sensible heat transfer of a dehumidifying coil is unaffected by the presence of moisture on its surface, Equation 5 may be obtained to express this part of the heat transfer in terms of the external film coefficient and the surface temperature.

$$q_s = h_a \times A \times N \times (MTD_a) \tag{5}$$

where

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).
t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.
t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.
t_s = average temperature of coil external surface, Fahrenheit degrees.
MTD_a = logarithmic mean temperature difference between air and coil surface =

$$\frac{t_1 - t_2}{\log_e \dfrac{t_1 - t_s}{t_2 - t_s}}$$

If Equation 5 is combined with another equation expressing sensible heat transfer in terms of mass velocity and temperature difference, the variables may be arranged in the following form (which is useful for the solution of dehumidification problems and for the determination of h_a from test data):

$$\frac{h_a AN(t_1 - t_2)}{\log_e \dfrac{t_1 - t_s}{t_2 - t_s}} = 0.243G(t_1 - t_2)$$

or,

$$\frac{h_a AN}{0.243G} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \tag{6}$$

where

0.243 = specific heat of humid air, Btu per (pound) (Fahrenheit degree).
G = air mass velocity, pounds per (hour) (square foot of coil face area).

An examination of Fig. 1 will reveal that when t_s is at the dew-point of the entering air:

$$\frac{t_1 - t_s}{t_a - t_s} = \frac{t_1 - t_{dp1}}{t_a - t_{dp1}}$$

and when t_s is below the dew-point:

$$\frac{t_1 - t_s}{t_2 - t_s} = \frac{t_1 - t_{dp1}}{t_2 - t_{d2}}$$

Therefore, Equation 6 may be written in its most useful form as:

$$\frac{h_a AN}{0.243G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \tag{7}$$

where

t_s = minimum dry-bulb possible without dehumidification, Fahrenheit degrees.
t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.
t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

This equation may be used to establish a line as A-2-3 for a given coil if h_a is known for the coil, or it *may be used to determine h_a from test data for the purpose of rating coils.* The use of this equation for coil selection is illustrated in Example 1 at the end of the chapter. Equation 7 is also important as a means of determining the external film coefficient.

External Film Coefficient

While formulas have been developed expressing the film coefficient h_a for air passing parallel to a plane surface, they cannot be used directly for fins on tubes because of air turbulence and because of the temperature gradient prevalent from the edge of a fin to its center. It is therefore necessary to make tests to evaluate the combined term ηh_a. The term, ηh_a, will be written merely h_a in this discussion as there is no necessity for separately evaluating η and because values of h_a are usually applied only to the particular coils for which tests are made.

The air side coefficient, h_a, of a coil of particular dimensions is an exponential function of the mass velocity of the air:

$$h_a = Z\,G^n \tag{8}$$

where

h_a = film coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and average surface temperature).
G = air mass velocity, pounds per (hour) (square foot of coil face area).
Z and n = constants which depend upon both air turbulence and surface arrangement.

Evaluation of constants Z and n may be accomplished through the use of test data in Equation 7 which gives values of h_a directly from the results of any *wet coil test.* If h_a, calculated in this manner, is plotted against values of G which prevailed during the tests a straight line should result on logarithmic coordinates. The slope of this line is the value of n. The value of Z may then be determined by direct substitution in Equation 8.

For finned coils of different designs, values of Z and n are extremely variable, depending on the particular design and arrangement of the coil surface. Therefore, it is desirable that these constants be determined directly from test data for each type of coil surface.

Internal Film Coefficient

The internal film coefficient, h_r which appears in Equation 3, is evaluated in various ways, depending upon the nature of the fluid, and whether the fluid is changing state.

When evaporating refrigerants are used in tubes, the temperature of the

fluid is fairly constant, being affected principally by pressure drop through the tubes, by superheat of the evaporated refrigerant, and by the presence of oil in solution. To obtain maximum coil capacity it is necessary to keep the pressure drop through the tubes at a minimum, to keep the superheat as low as possible without carrying liquid back to the compressor, and to arrange for good separation and return of oil to the compressor. Another important factor is the removal of gas to keep the tube surface flooded with liquids as much as possible. The internal film coefficient is markedly increased by heavy heat loads, because the increased turbulence and gas velocity cause good contact of the liquid with the tubes. Values of h_r usually lie between 150 and 450. For rating of dehumidifying coils, satisfactory results are obtainable by first determining the average external surface temperature from Equation 7, and then using the difference between the external film temperature and the refrigerant for evaluating h_r in Equation 9.

$$h_r = \frac{q_t}{\frac{AN}{R}(t_s - t_r)} \tag{9}$$

where

h_r = internal film coefficient of heat transfer, Btu per (hour) (square foot of internal tube surface) (Fahrenheit degree).

t_r = average refrigerant temperature, Fahrenheit degrees.

The term $(t_s - t_r)$ is commonly written Δt. To evaluate h_r by this method the same tests that were required to determine h_a may be used.

When water is the cooling medium in tubes, the rate of heat transfer is a function of its velocity, which influences the number of contacts of the water molecules with the tube surface, per unit of time. Increased water velocity and reduced tube diameter cause increased heat transfer. Heat transfer is also greater at higher temperatures of the water. The basic formula for the film coefficient of heat transfer for flow of water in smooth tubes is as follows:

$$h_r = 1.5(t + 100)\frac{V^{0.8}}{D^{0.2}} \tag{10}$$

where

V = water velocity, feet per second.

D = internal diameter of tube, inches.

t = average water temperature, Fahrenheit degrees.

Equation 10 should not be used when Reynolds Number is less than 2000.

Since, in the case of finned tubes using water as a refrigerant, test values of h_r based on the calculated surface temperature for the entire coil may be lower than those obtained by use of Equation 10, actual test results are preferred if available.

When saturated steam is condensed in the tubes of coils, the film coefficient h_r varies from 1000 to 2000, depending on freedom from air in the steam, and upon good drainage of the tubes. The coefficient is fairly constant for a particular coil, giving values of Δt that are directly proportional to q_t. However, if water coil test results are analyzed on a row-by-row basis good agreement with Equation 10 will result.[3]

The use of turbulence promoters increases the value of h_r for liquids in tubes at the expense of pressure drop. The increase obtained depends upon the type of turbulence promoter and the rate of flow. No general statement can be made regarding their use and it is best to refer to detailed papers on this subject for further information.[3,4]

Determining Size of Cooling Coil

To illustrate the use of individual film coefficients in coil calculations, the procedure for selecting the proper size cooling coil and for determining exit air condition, coil surface temperature, total coil load and refrigerant temperature is outlined in Example 1.

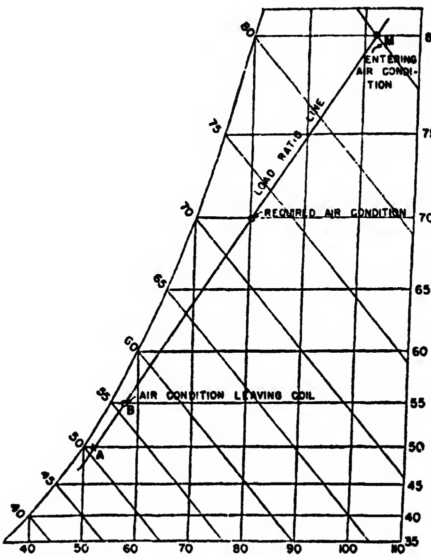


FIG. 2. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

Example 1. An industrial application requires the cooling of a certain quantity of air from a condition of 102 F dry-bulb and 85 F wet-bulb to a final condition of 80.5 F dry-bulb and 73 F wet-bulb. The air velocity across the coil is to be 400 fpm and coil data are as follows: h_a = 10.7 at 400 fpm, h_r = 325, external surface area = 15 sq ft per (square foot of face) (row of coil depth), ratio of external surface area to internal surface area = 15.

Solution. (1) Lay out the problem psychrometry as indicated in Fig. 2 and note that the minimum horizontal distance between the load ratio line and the saturation curve is 1.8 F dry-bulb at point A Fig. 2. This means that $t_2 - t_{dp2}$ in Equation 7 must not be less than 1.8. Therefore, Equation 7 should be solved for N to determine the proper number of rows to be used for the coil.

$$\frac{h_a AN}{0.243G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{102 - 80}{1.8} = \log_e 12.22 = 2.5$$

Then substituting values for h_a, A, and G, N may be found as follows:

$$\frac{10.7 \times 15N}{0.243 \times 1740} = 2.5 \text{ from which, } N = 6.58$$

(2) This establishes the maximum whole number of coil rows that can be used as 6 and it is now possible to determine the actual location of the exit air conditions from Equation 7 by solving for the actual value of $t_2 - t_{dp2}$ for a 6 row coil.

$$\frac{10.7 \times 15 \times 6}{0.243 \times 1740} = \log_e \frac{102 - 80}{t_1 - t_{dp2}} = 2.275$$

This establishes values of 9.78 for $\frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = R$ and 2.25 for $t_2 - t_{dp2}$.

(3) Next, the exit air condition at 57.3 F dry-bulb and 56 F wet-bulb as shown at B, is found by locating a point on the load ratio line at a horizontal distance of 2.25 dry-bulb degrees from the saturation curve.

(4) The surface temperature may now be found from Equation 7 which may also be written as:

$$t_s = \frac{Bt_2 - t_1}{B - 1}$$

where

$$B = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

$$t_s = \frac{9.78 \times 57.3 - 102}{8.78} = 52.3$$

(5) The total coil load may be calculated from the enthalpy difference across the coil and the air quantity using the weight of dry air instead of the weight of the mixture.

$$q_t = G_a\ (h_1 - h_2)$$

$$= 1700\ (49.24 - 23.77) = 43{,}200 \text{ Btu per (hr) (sq ft of face area)}$$

where

G_a = weight of dry air per (hour) (square foot of coil face area).

h_1 = enthalpy of air vapor mixture entering coil, Btu per pound of dry air.

h_2 = enthalpy of air vapor mixture leaving coil, Btu per pound of dry air.

(6) The refrigerant temperature may be found from Equation 9

$$\frac{43{,}200}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 22.1$$

Therefore, $t_r = (52.3 - 22.1) = 30.2$.

Thus a coil 6 rows deep, operating at a refrigerant temperature of 30.2 F and a face velocity of 400 fpm, is required; and it will carry a total load of 43,200 Btu per (hour) (square foot of face area). The air conditions leaving the coil are too low for the conditions of the problem and therefore it is necessary to by-pass air at the entering condition to obtain the desired result of 80.5 F dry-bulb and 73 F wet-bulb.

Although the preceding solution is satisfactory, it may be more desirable in some cases to use a higher refrigerant temperature and employ reheat to obtain the desired load ratio. Such a solution is shown in Fig. 3. In this case the coil load ratio line intersects the saturation curve and therefore a coil of any depth may be selected.

If a coil depth of 6 rows is maintained, the exit air conditions for the coil are indicated at point B Fig. 3 as 72.3 F dry-bulb and 70.8 F wet-bulb and the surface temperature will be:

$$t_s = \frac{9.78 \times 72.3 - 102}{8.78} = 69.0$$

The coil load will be: $q_t = 1700\ (49.24 - 34.66) = 24{,}800$ Btu per (hour) (square foot of face area) and the refrigerant temperature will be found from Equation 9:

$$\frac{24{,}800}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 12.7$$

Therefore, $t_r = 69.0 - 12.7 = 56.3$.

Thus, for the case where reheat is used, a coil 6 rows deep operating at a refrigerant temperature of 56.3 F is required. The total coil load will be 24,800 Btu per (hour)

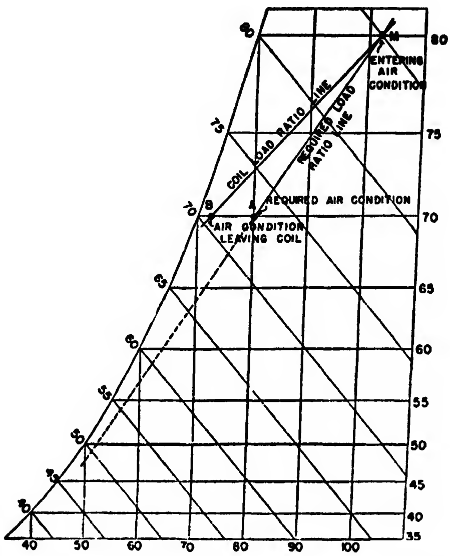


FIG. 3. PSYCHROMETRIC LAYOUT FOR COIL SELECTION USING REHEAT

(square foot of face area) but the actual effective load will be less by the amount of reheat required. Therefore, for a given load, a larger coil and more refrigerating capacity are required when reheat is used.

LETTER SYMBOLS USED IN CHAPTER 7

η = fin efficiency.

A = external area of coil, square feet per (square foot of coil face area) (row of coil depth).

$t_1 - t_{dp1}$

D = internal diameter of tube, inches.

G = air mass velocity, pounds per (hour) (square foot of coil face area).

G_a = dry air mass velocity, pounds dry air per (hour) (square foot of coil face area).

h_1 = enthalpy of air-vapor mixture entering coil, Btu per pound of dry air.

h_2 = enthalpy of air-vapor mixture leaving coil, Btu per pound of dry air.

h_a = film coefficient of heat transfer between air and external coil surface, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and coil).

h_i = film coefficient of heat transfer between fluid and internal coil surface, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature between fluid and surface).

k = conductivity of pipe or tube material, Btu (square foot) (hour) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

MTD = abbreviation—mean temperature difference between fluid in coil and air passing over coil, Fahrenheit degrees.
NOTE: *MTD*—usually logarithmic mean.

MTD_a = logarithmic mean temperature difference between air and coil surface.

N = number of rows of coil depth.

n = a constant, exponent of G in Equation 8, obtained by plotting, on logarithmic coordinates, G against values of h_a. The value of n is the slope of the line.

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).

q_t = total heat transferred by coil, Btu per (hour) (square foot of face area).

R = ratio between external and internal surface of tube.

t = average water temperature, Fahrenheit degrees.

t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.

t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.

t_a = minimum dry-bulb temperature possible without dehumidification, Fahrenheit degrees.

t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.

t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

t_r = average refrigerant temperature, Fahrenheit degrees.

t_s = average temperature of external surface of coil, Fahrenheit degrees.

$\Delta t = t_s - t_r$.

U = over-all coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degrees temperature difference between fluid in coil and air flowing over coil).

V = water velocity, feet per second.

Z = a constant for use in Equation 8 obtained by plotting on logarithmic coordinates G against values of h_a.

NOTE: Numerical subscripts refer to condition entering and leaving respectively.

REFERENCES

[1] Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall (*Refrigerating Engineering*, October, 1935, p. 211).

[2] Performance of Surface-Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. G. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 523).

[3] The Effect of Turbulence Promoters on Heat Transfer Coefficients for Water Flowing in Horizontal Tubes, by L. G. Seigel (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1946, p. 111).

[4] Maximum Rate of Heat Transfer with Minimum Loss of Energy, by Z. Nagoaka and A. Watanabe (Proceedings, *International Congress on Refrigeration*, 7th Congress, Vol. 3, No. 16, pp. 221-245, 1937).

CHAPTER 8

AIR LEAKAGE

Causes of Infiltration, Infiltration Due to Wind Pressure, Infiltration Through Walls, Window and Door Leakage, Crack Method, Air Change Method, Infiltration Due to Temperature Difference, Sealing of Vertical Openings

THE air leakage which takes place through various apertures in buildings must be considered in heating and cooling calculations, and properly evaluated. This *infiltration* as it is sometimes designated takes place through cracks around doors and windows, through solid walls and through fireplaces and chimneys. Although the latter sources of leakage may be considerable, they are often neglected on the assumption that dampers would be closed during periods of extreme cold weather or else that the fireplace will be in use at such times and will therefore contribute to the heat supplied and lessen the heating load.

CAUSES OF INFILTRATION

The displacement of heated air in buildings by unheated outside air is due to two causes, namely, (1) the pressure exerted by the wind and (2) the difference in density of outside and inside air because of differences in temperature. The former is generally referred to as *infiltration* and the latter as *stack* or *chimney effect.*

In either case an exact estimate of the amount of infiltration under design conditions is difficult to make. The complicating factors include (1) variations in building construction particularly as to width of crack or size of openings through which air leakage takes place, (2) the variations in wind velocity and direction, (3) the exposure of the building with respect to air leakage openings and with respect to adjoining buildings, (4) the variations in outside temperatures which influence the chimney effect, (5) the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors, and (6) the influence of a planned air supply and the related outlet vents. Tight construction is essential for preventing large heat loss due to infiltration.

INFILTRATION DUE TO WIND PRESSURE

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages, owing to this building up of pressure, will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure which are greater or lower than prevalent wind pressures. In such designs, if the rate at which air is specified to be introduced to or removed from the enclosure by posi-

tive means exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

Infiltration Through Walls

Data on infiltration through brick and frame walls are given in Table 1[1]. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building,

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall[b]	Plain	2	4	8	12	19	23
	Plastered[c]	0.02	0.04	0.07	0.11	0.16	0.24
13 in. Brick Wall[b]	Plain	1	4	7	12	16	21
	Plastered[c]	0.01	0.01	0.03	0.04	0.07	0.10
	Plastered[d]	0.03	0.10	0.21	0.36	0.53	0.72
Frame Wall, with lath and plaster[e]		0.03	0.07	0.13	0.18	0.23	0.26

[a] The values given in this table are 20 per cent less than test values to allow for building up of pressur, in rooms and are based on test data reported in the papers listed in chapter footnotes.
[b] Constructed of porous brick and lime mortar—workmanship poor.
[c] Two coats prepared gypsum plaster on brick.
[d] Furring, lath, and two coats prepared gypsum plaster on brick.
[e] Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and three coats gypsum plaster.

with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them. The infiltration indicated in Fig. 1 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

Window and Door Leakage

There are two methods of estimating air leakage through window and door cracks, namely, (1) the crack method and (2) the air change method.

The crack method is generally regarded as being more accurate than the purely arbitrary air change method, provided the variables such as crack width and clearance, can be properly evaluated.

Crack Method

The crack method is based on known air leakage factors for various types of windows and widths of crack and clearance. The wind velocity and length of crack are also considered when the crack method is employed. The amount of infiltration for various types of windows is given in Table 2[2]. The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the

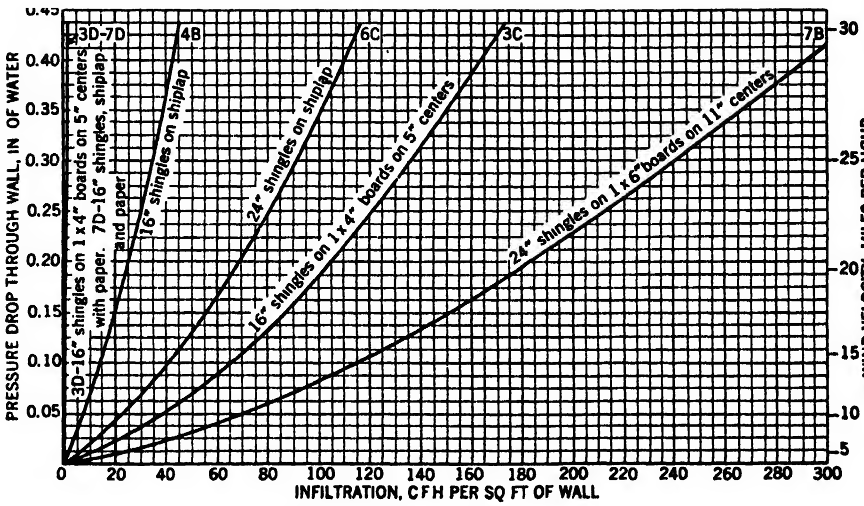


FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

sash thickness is considered as the clearance. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. All of the window crack in any given room is not necessarily used in estimating the infiltration heat loss by the crack method. The length of crack to be selected in any given case depends on the number of exposed sides as explained in Chapter 14.

Values of leakage shown in Table 2 for the average double-hung wood window were determined by using, on nine windows tested in the laboratory, the average measured crack and clearance of a large number of windows found in a field survey. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3	8	14	20	27	35
	Around frame in masonry wall—calked[b]	1	2	3	4	5	6
	Around frame in wood frame construction[b]	2	6	11	17	23	30
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood frame leakage[d]	7	21	39	59	80	104
	Ditto, weatherstripped[d]	4	13	24	36	49	63
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance[e] Includes wood frame leakage[d]	27	69	111	154	199	249
	Ditto, weatherstripped[d]	6	19	34	51	71	92
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in. crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed in chapter footnotes.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and non-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represent a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architecturally projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of

TABLE 3. INFILTRATION THROUGH 72-INCH REVOLVING DOOR AND 36-INCH SWINGING DOOR[a,b]

(Cubic Feet per Person per Passage)

USAGE	FREELY-REVOLVING DOOR	DOOR EQUIPPED WITH BRAKE
Infrequent	75	60
Average	60	50
Heavy	40	40
36-Inch Swinging Door	100	

[a] These figures are based on the assumption that there is no wind pressure and that swinging doors are in use in one wall only. Any swinging doors in other walls should be kept closed to insure air conditioning in accordance with these recommended standards.

[b] From Application Engineering Standards for Air Conditioning for Comfort. *Air Conditioning & Refrigerating Machinery Association, Inc.*, Washington, D. C. Used by permission.

sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows[3]. By applying storm sash to poorly fitted windows, a reduction in leakage of 50 per cent may be obtained, the effect so far as air leakage is concerned being roughly equivalent to that obtained by the installation of weatherstrips.

Door Leakage

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3

represent current engineering practice. Some tests of infiltration through swinging and revolving doors have been reported.[4]

Air Change Method

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. Where it is not possible to determine or pre-determine with accuracy the width of crack or clearance of windows, or where other sources of air leakage cannot readily be evaluated, as is often the case, the use of the air change method may be justified.

The values in Table 4 may be used with reasonable accuracy for residences and are the requirements for each room. The total infiltration allowance for the entire building should be one-half the sum of the infiltration allowances of the individual rooms, since whatever air enters on the windward side generally leaves the building on the leeward side and the infiltration requirements therefore do not exist simultaneously on all sides or in all rooms. An allowance of one air change per hour for all sources of air leakage for the entire volume may be considered average for a well constructed residence.

Table 4. Air Changes Taking Place under Average Conditions Exclusive of Air Provided for Ventilation[a]

Kind of Room or Building	Number of Air Changes taking Place per Hour	Kind of Room or Building	Number of Air Changes taking Place per Hour
Rooms, 1 side exposed	1	Rooms with no windows or outside doors	½ to ¾
Rooms, 2 sides exposed	1½	Entrance Halls	2 to 3
Rooms, 3 sides exposed	2	Reception Halls	2
Rooms, 4 sides exposed	2	Bath Rooms	2
		Stores	1 to 3

[a]For rooms with weatherstripped windows or storm sash, use ⅔ these values, where applicable.

The air leakage for vestibules due to opening and closing of doors is sometimes based on the air change method, even though the air leakage estimates for other rooms are based on the crack method. Except for vestibules and reception halls, it is not advisable to attempt to apply the air change method to factories and industrial and commercial buildings because of wide variations in the type and percentage of fenestration which is the principal source of air leakage in such buildings.

INFILTRATION DUE TO TEMPERATURE DIFFERENCE

The air exchange due to temperature difference, inside to outside, is a chimney effect, causing air to enter through openings at lower levels and to leave at higher levels[5]. Although it is not appreciable in low buildings, this loss should be considered in tall, single story buildings with openings near the ground level and near the ceiling. Also in tall, multi-story buildings it may be a considerable item unless the sealing between various floors and rooms is quite perfect.

In tall buildings, temperature difference or chimney effect will produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Further-

more, the chimney effect is reduced in multi-story buildings by the partial isolation of floors, thereby preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone*[6] is located at mid-height of a building, and that the temperature difference is 70 F, Equations 1 and 2 may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$V_e = \sqrt{V^2 - 1.75a} \tag{1}$$

$$V_e = \sqrt{V^2 + 1.75b} \tag{2}$$

where

V_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
V = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height, feet.
b = distance if *below* mid-height, feet.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and, in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

Infiltration and Air for Combustion

Infiltration in residences normally supplies the air required for combustion by fuel burning appliances, but in some residences weatherstripping,

sealing and caulking may reduce infiltration to the point that special openings must be provided to supply adequate air to the heating appliances.

REFERENCES

[1] A.S.H.V.E. Research Report No. 786—Infiltration Through Plastered and Unplastered Brick Walls, by F. C. Houghten and Margaret Ingels (A.S.H.V.E. Transactions, Vol. 33, 1927, p. 377). No. 826—Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 183). No. 851—Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 99. No. 868—Air Infiltration Through Various Types of Wood Frame Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 397).

[2] A.S.H.V.E. Research Reports No. 686—Air Leakage, by F. C. Houghten and C. C. Schrader (A.S.H.V.E. Transactions, Vol. 30, 1924, p. 105). No. 704—Air Leakage Around Window Openings, by C. C. Schrader (A.S.H.V.E. Transactions, Vol. 30, 1924, p. 313). No. 803—Air Leakage on Metal Windows in a Modern Office Building, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. Transactions, Vol. 34, 1928, p. 321). No. 815—Air Leakage Through a Pivoted Metal Window, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. Transactions, Vol. 34, 1928, p. 519). No. 817—Effect of Frame Calking and Storm Sash on Infiltration Around and Through Windows, by W. M. Richtmann and C. Braatz (A.S.H.V.E. Transactions, Vol. 34, 1928, p. 547). No. 909—Air Infiltration Through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 571). The Weathertightness of Rolled Section Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. Transactions, Vol. 34, 1928, p. 527). Pressure Differences Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 83). Air Infiltration Through Steel Framed Windows, by D. O. Rusk, V. H. Cherry and L. Boelter (A.S.H. V.E. Transactions, Vol. 39, 1933, p. 169).

[3] Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo (A.S.H.V.E. Transactions, Vol. 42, 1936, p. 87).

[4] The Infiltration Problem of Multiple Entrances, by A. M. Simpson and K. B. Atkinson (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1936, p. 345). Infiltration Characteristics of Entrance Doors, by A. M. Simpson (*Refrigerating Engineering*, June, 1936).

[5] A.S.H.V.E. Research Reports No. 994—Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 387). No. 1069—Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 437). Flue Action in High Buildings, by H. L. Alt (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, May, 1932, p. 376). Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 377).

[6] Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. Transactions, Vol. 32, 1926, p. 59).

CHAPTER 9

NATURAL VENTILATION

Wind Forces, Temperature Difference Forces, Heat Removal, Effect of Unequal Openings, Combined Wind and Temperature Forces, Types of Openings, Windows, Doors, Skylights, Roof Ventilators, Principles of Control, Stacks, General Rules, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings. The natural forces available for moving air into, through, and out of buildings are: (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside a building. The air movement may be caused by either of these forces acting alone or by a combination of the two, depending upon atmospheric conditions, building design and location. The ventilating results obtained will vary, from time to time, due to variation in the velocity and direction of the wind and the temperature difference. The arrangement, location, and control of the ventilating openings should be such that the two forces act cooperatively rather than in opposition.

WIND FORCES

In considering the use of natural wind forces for producing ventilation, account must be taken of: (1) average wind velocity, (2) prevailing wind direction, (3) seasonal and daily variations in velocity and direction, and (4) local wind interference by nearby buildings, hills or other obstructions of similar nature.

Values are given in Table 3, Chapter 15 for the average wind velocities for the months June to September in various localities throughout the United States, while Table 1, Chapter 14, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While the tables give no average velocities below 5 mph, there will be times when the velocity is lower, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the velocity falls below one-half of the average for many hours per month. Consequently, if the natural ventilating system is designed for wind velocities of one-half of the average seasonal velocity, it should prove satisfactory in almost every case.

Equation 1 may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings to produce given results:

$$Q = EAV \qquad (1)$$

where

Q = air flow, cubic feet per minute.
A = free area of inlet openings, square feet.
V = wind velocity, feet per minute, = miles per hour $\times$ 88.
E = effectiveness of openings. (E should be taken at 0.50 to 0.60 for perpendicular winds and 0.25 to 0.35 for diagonal winds[1].)

The accuracy of the results obtained by the use of Equation 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edged orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less and, if unusually well placed, the flow will be slightly more than that given by the

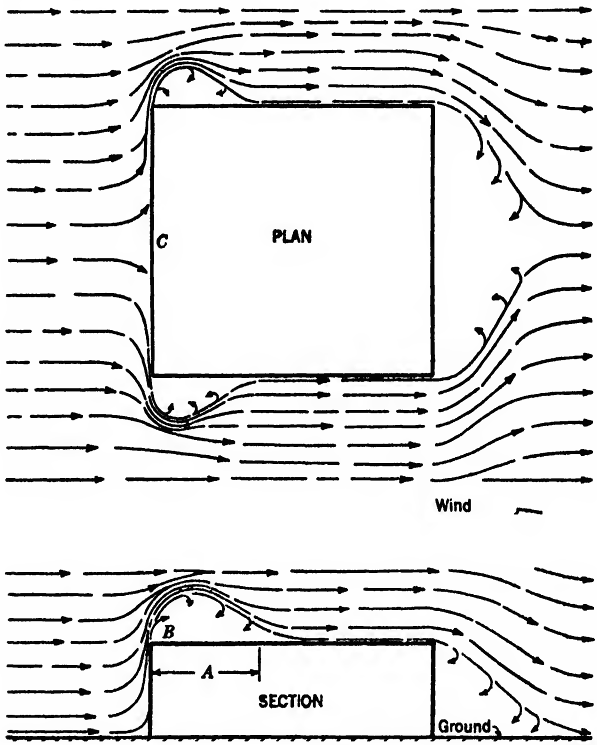


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (A—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the five places listed:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. On the sides adjacent to the windward face where low pressure areas occur.
4. In a monitor on the side opposite from the wind.
5. In roof ventilators or stacks.

TEMPERATURE DIFFERENCE FORCES[2]

The stack effect produced within a building when the outdoor temperature is lower than the indoor temperature is due to the difference in weight of the warm column of air within the building and cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4A\sqrt{h(t - t_o)} \tag{2}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
h = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height h, Fahrenheit degrees.
t_o = temperature of outdoor air, Fahrenheit degrees.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of Equation 3.

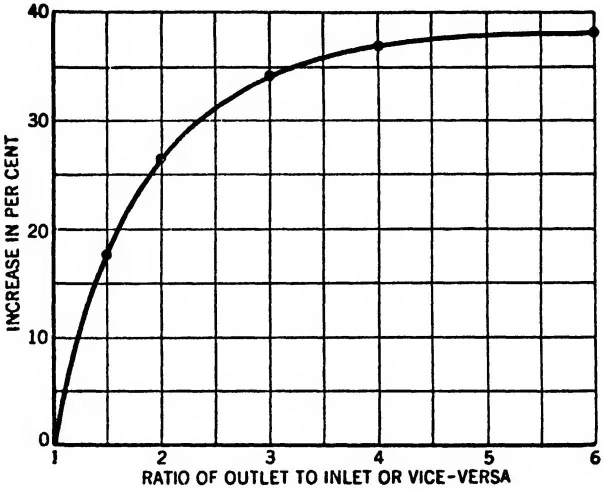


FIG. 2. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER ANOTHER

$$H = 0.0175\,Q\,(t - t_o) \qquad (3)$$

H = heat removed, Btu per minute.
Q = air flow, cubic feet per minute.
$t - t_o$ = inside-outside temperature difference, Fahrenheit degrees.

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the preceding equations are based on this condition. Increasing outlets over inlets, or vice-versa, will increase the air flow, but not in proportion to the added area. When solving problems having an unequal distribution of openings, use the smaller area, either inlet or outlet, in the equations and add the increase as determined from Fig. 2.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations for determining the air flow due to temperature difference and wind have already been given. It must be remembered that when

both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The flow through any opening is proportional to the square root of the sum of the heads acting on that opening.

When the two heads are about equal in value and the ventilating openings are operated so as to coordinate them, the total air flow through the building is about 10 per cent greater than that produced by either head acting independently under conditions ideal to it. This percentage decreases rapidly as one head increases over the other and the larger will predominate.

The wind velocity and direction, the outdoor temperature, or the indoor distribution, cannot be predicted with certainty, and refinement in calculations is not justified; consequently, a simplified method can be used.

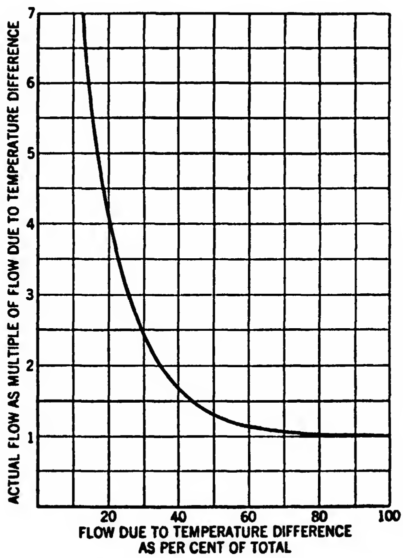


Fig. 3. Determination of Flow Caused by Combined Forces of Wind and Temperature Difference

This may be done by using the equations and calculating the flows produced by each force separately under conditions of openings best suited for coordination of the forces. Then, by determining, as a percentage, the ratio of the flow produced by temperature difference to the sum of the two flows, the actual flow due to the combined forces can be approximated from Fig. 3.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per pound is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Desired summer temperature difference is 10 deg and the prevailing wind is 8 mph perpendicular to the long dimension. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution for Temperature Difference Only. The heat $H = \frac{15 \times 7.75 \times 18{,}000}{60} =$ 34,875 Btu per minute.

By Equation 3, the air flow required to remove this heat with an average temperature difference of 10 deg is:

$$Q = \frac{H}{0.0175(t - t_o)} = \frac{34{,}875}{0.0175 \times 10} = 199{,}286 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 2 the inlet (or outlet) opening area should be:

$$A = \frac{Q}{9.4\sqrt{h(t - t_o)}} = \frac{199{,}286}{9.4\sqrt{30 \times 10}} = 1224 \text{ sq ft.}$$

The flow per square foot of inlet or outlet would be 199,286 ÷ 1224 = 163 cfm with all windows open.

Solution for Wind Only. With 1,224 sq ft of inlet openings distributed around the sidewalls, there will be about 410 sq ft in each long side and 202 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 612 sq ft on each side. With the wind perpendicular to the long side, there will be 410 sq ft of opening in its path for inflow and 612 in the lee side of the monitor for outflow with the windward side closed. The air flow, as calculated by Equation 1, will be:

$$Q = 0.60 \times 410 \times 704 = 173{,}200 \text{ cfm.}$$

This gives 17.3 air changes per hour, which should be more than ample when there is no heat to be removed.

Solution to Combined Heads. Since the windward side of the monitor is closed when the wind is blowing, the flow due to temperature difference must be calculated for this condition, using Fig. 2. This chart shows that when inlets are twice the size of the outlets, in this case 1,224 sq ft in the sidewalls and 612 sq ft in the monitor, the flow will be increased 26.5 per cent over that produced by equal openings. Using the smaller opening and the flow per square foot obtained previously, the calculated amount for this condition will be:

$$612 \times 163 \times 1.265 = 126{,}200 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 126,200 =	42 per cent.
Wind	= 173,200 =	58 per cent.
Total	299,400 =	100 per cent.

From Fig. 3, it is determined that when the flow, due to temperature difference, is 42 per cent of the total, the actual flow, due to the combined forces, will be about 1.6 times that calculated for temperature difference alone, or 201,920 cfm.

The original flow, due to temperature difference alone, was 199,286 cfm with all openings in use. The effect of the wind is to increase this to 201,920 cfm even though half of the outlets are closed.

A factor of judgment is necessary in the location of the openings in a building, especially those in the roof, where heat, smoke and fumes are to be removed. Usually windward monitor openings should be closed, but if the wind is low enough for the temperature head to overcome it, all windows may be opened.

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights, (2) roof ventilators, (3) stacks connecting to registers, and (4) specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in various ways; they may open by sliding either vertically or horizontally,

by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side. Regardless of their design, the air flow per square foot of opening may be considered to be the same under the same conditions. The type of pivoting should receive consideration from the standpoint of weather protection, and certain types may be advantageous in controlling the distribution of incoming air. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weatherproof air outlet. These are actuated by the same forces of wind and temperature head which create flow through other types of openings. The capacity of a ventilator depends upon four things: (1) its location on the roof, (2) the resistance it and the duct work offer to air flow, (3) the height of draft, and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof where it will receive the full wind without interference. If ventilators are installed within the suction region created by the wind passing over the building, or in a light court, or on a low building between two high buildings, their performance will be seriously influenced. Their normal ejector action, if any, may be completely lost.

The base of the ventilator should be of a taper-cone design to produce the effect of a bell-mouth nozzle whose coefficient of flow is considerably higher than that of a square-entrance orifice. If a grille is provided at the base, or if the base or structural members present obstructions, additional resistance is introduced, and the base opening should be increased in size accordingly.

Air inlet openings located at lower levels in the building should be at least equal to, and preferably larger than, the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, and, in general, its performance will be the same as any monitor opening located in the same place but, due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Roof ventilators may be classified as stationary, pivoting or oscillating, and rotating. Generally, these have a round throat, but the continuous-ridge ventilator would fall in the stationary classification. When selecting roof ventilators, some attention should be given to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibility of noise, original cost, and maintenance.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to stop the power-driven units. Units are not subject to code tests for ratings. Generally they must be selected from manufacturers' tables. It is, therefore, very important to consider the reliability of the ratings used.

Controls

Gravity ventilators may have dampers controlled by hand, thermostat, or wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor

being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys which function through the effects of the wind and temperature difference. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction. With little or no wind, the chimney effect depends entirely on temperature difference to produce a removal of air from the rooms where the inlet openings are located.

GENERAL RULES

A few of the important considerations in addition to those already outlined are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone to be ventilated.

2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc., outside nor by partitions inside.

3. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

4. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. Where the wind's direction is quite variable, the openings should be arranged in sidewalls and monitors so that, as far as possible, there will be approximately equal areas on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force and others to a suction force, and effective movement through the building will be assured.

5. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

6. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

7. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

8. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

9. In an industrial building where furnaces that give off heat and fumes are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas-laden air.

10. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that

the wind, in splashing therefrom, will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. In single story industrial buildings, particularly those covering large areas, natural ventilation must be accomplished by taking air in and out of the roof openings. Openings in the pressure zones can be used for inflow and openings in the suction zone, or openings in zones of less pressure, can be used for outflow. The ventilation is accomplished by the manipulation of openings to get air flow through the zones to be ventilated.

DAIRY BARN VENTILATION[3]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hr. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hr. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume and construction permit adequate heating by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hr, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cfh of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per (hr) (cu ft of barn space), and 0.197 to 0.305 Btu per (hr) (sq ft of barn exposure).

GARAGE VENTILATION

Because of hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be over-emphasized. During the warm months of the year, garages are usually ventilated adequately

because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means, particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilating Garages, adopted in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impracticable to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage[4].

Research

Cooperative research on garage ventilation, undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., and tests conducted at the A.S.H.V.E. Research Laboratory, have resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than that obtained with mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

REFERENCES

[1] Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 605).

[2] Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. Transactions, Vol. 32, 1926, p. 59).

[3] Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. Transactions, Vol. 34, 1928, p. 181). Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 149). For additional information on this subject refer to *Technical Bulletin, U. S. Department of Agriculture* (1930), by M. A. R. Kelley. Also

see Air Conditioning of Farm Buildings, by F. L. Fairbanks (*Agricultural Engineering*, November, 1937, p. 485).

[4] Code of Minimum Requirements for Heating and Ventilating Garages (A.S.H.-V.E. Transactions, Vol. 41, 1935, p. 30).

Airation Study of Garages by W. C. Randall and L. W. Leonhard (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 233).

A.S.H.V.E. Research Report No. 874—Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 511).

A.S.H.V.E. Research Report No. 935—Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 439).

A.S.H.V.E. Research Report No. 934—Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 424).

A.S.H.V.E. Research Report No. 967—Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 263).

CHAPTER 10

AIR CONTAMINANTS

Classification of Air Contaminants; Sizes of Airborne Particles; Air Pollution by Smoke, Ash and Cinders; Smoke Abatement; Odor Nuisance; Maximum Allowable Concentrations of Industrial Air Contaminants; Flammable Gases and Vapors; Combustible Dusts; Atmospheric Pollen; Airborne Bacteria

THE normal constituents of the earth's atmosphere are oxygen, nitrogen, carbon dioxide, water vapor, argon, small or negligible amounts of other inert gases, hydrogen, variable traces of ozone, and small quantities of microscopic and submicroscopic solid matter, sometimes called *permanent atmospheric impurities.* From the viewpoint of the air conditioning engineer, all other airborne substances may be termed *contaminants.* This term is applied preferably, however, to undesirable or chance impurities, since the occasion may arise for adding to the air controlled amounts of solid or gaseous diluents for the prevention of explosions; germicidal vapors or mists (aerosols) for bacteria control; masking substances for odor control; or a substitute for one of the normal gases, as, for example, when helium is used to replace nitrogen in atmospheres for compressed air workers or divers.

Control of the chemical quality of air is one of the functions of complete air conditioning, and some knowledge of the composition, concentration and properties of air contaminants under various circumstances is therefore essential.

Air contaminants arise from the normal processes of wear, erosion, windstorm, sea-spray evaporation, thermal disintegration, earthquake, volcanic eruption, combustion, manufacturing, transportation, agriculture, and the biochemical or biological processes of life. They are classified at various times as organic and inorganic, visible or invisible, microscopic or macroscopic, particulate or gaseous, toxic or harmless, beneficial or destructive. The following classification is based chiefly upon the origin or method of formation of air contaminants.

CLASSIFICATION OF AIR CONTAMINANTS

Dusts, Fumes, and Smokes are solid particulate air contaminants.

Dusts are solid particles projected into the air by natural forces, such as wind, volcanic eruption or earthquake, and by mechanical processes, such as crushing, grinding, milling, drilling, demolition, shovelling, conveying, screening, bagging and sweeping. Some of these forces *produce* dust from larger masses, while others simply *disperse* materials that are already pulverized. Generally, particles are not called *dust* unless they are smaller than about 100 microns. Dusts may be of *mineral* type, such as rock, ore, metal, sand; *vegetable,* such as grain, flour, wood, cotton, pollen; or *animal,* such as wool, hair, silk, feathers, leather.

Fumes are solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Metallic fumes generally occur as the oxides in air because of the highly reactive nature of finely divided matter. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction, whenever such processes create airborne particles predominately below the 1 micron size. Fumes permitted to *age* tend to flocculate into clumps or aggregates of larger size, thereby facilitating removal from air.

Smokes are the extremely small solid particles produced by incomplete combustion

of organic substances such as tobacco, wood, coal, oil, tar and other carbonaceous materials. The term *smoke* is commonly applied to the mixture of solid, liquid and gaseous products of combustion, although the technical literature prefers to distinguish between such components as soot or carbon particles, fly-ash, cinders, tarry matter, unburned gases, and gaseous combustion products. The finest particulate constituents are much less than 1 micron in size, often in the range of 0.1 to 0.3 micron.

Mists and Fogs are liquid particulate air contaminants.

Mists are very small airborne droplets of materials that are ordinarily liquid at normal temperatures and pressures. They may be formed by atomizing, spraying, splashing, mixing, violent chemical reaction, electrolytic evolution of gas from a liquid, or escape of a dissolved gas upon release of pressure. Very small droplets expelled or atomized into the air by sneezing constitute mists containing microorganisms that become air contaminants.

Fogs are limited by some classifications to airborne droplets formed by condensation from the vapor state. This arbitrary distinction between mist and fog is of minor importance, as both terms are used to indicate the particulate state of airborne liquids (occasionally termed *aerosols*). *Fog nozzles* are so named because of their ability to produce extra fine droplets as compared to the mist from ordinary spray devices. The highly volatile nature of some liquids quickly reduces their airborne droplets from the mist to the fog range, and eventually to the vapor phase until the air becomes saturated with that liquid. Many droplets in fogs or clouds are microscopic and submicroscopic in size, and may be conceived as the transition state between the larger mists and the vapors.

Vapors and Gases are non-particulate air contaminants.

Vapors are the gaseous phase of substances that are either liquid or solid in their commonly known state, examples being gasoline, kerosene, benzene, carbon tetrachloride, mercury, iodine, camphor. Vapors may be changed to the solid or liquid form by increasing the pressure, decreasing the temperature or applying both processes simultaneously. They are removed from the air by condensation with less difficulty than are the gases.

Gases are normally formless fluids which tend to occupy a space or enclosure completely and uniformly at ordinary temperatures and pressures. The following substances qualify as gases: oxygen, nitrogen, carbon dioxide, carbon monoxide, hydrogen, ammonia, sulfur dioxide. Gases, likewise, may be solidified or liquefied by the proper control of temperature and pressure.

The preceding classification is not suitable for the airborne living organisms, which range in size from the submicroscopic viruses to the largest pollen grains, not considering the smallest insect life. Bacteria range from about 0.2 to 5 microns in size, fungus spores from 1 to 10 microns, and pollen from 5 to 150 microns.

SIZES OF AIRBORNE PARTICLES

Fig. 1 is a graphic tabulation of the properties of airborne solids and liquids arranged according to size on the micron scale. There are 25,400 microns in 1 inch.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength, but settle out by gravity at speeds dependent upon the shape, size and specific gravity of the particle, wind velocity, orientation of the collecting surface, and topography. These larger particles are of major interest to the engineer in the solution of nuisance problems, but it is usually the smaller particles, or those below 10 microns, that remain in the air long enough to be of hygienic as well as economic significance.

Industrial dust particles are predominantly of the order of 1 micron in size. Tremendous numbers are also present in the sub-microscopic range below 0.5 micron, but those below 0.1 micron are not believed at

present to be of practical importance, possibly due to their exceedingly small mass in comparison with the balance of airborne matter. In fact, particles this small may become the *permanent* atmospheric impurities that have little if any opportunity of settling because of the continual motion imparted to them by air currents and the molecular activity of gases (Brownian Movement).

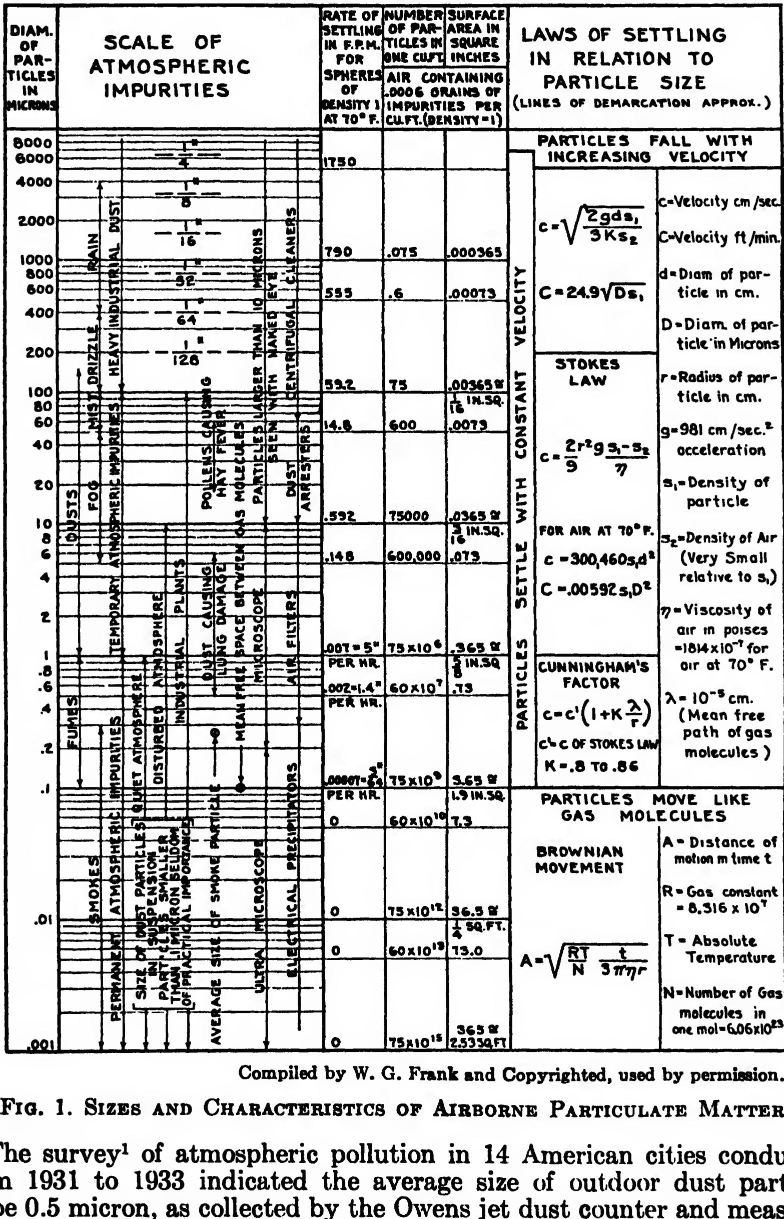


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FIG. 1. SIZES AND CHARACTERISTICS OF AIRBORNE PARTICULATE MATTER

The survey[1] of atmospheric pollution in 14 American cities conducted from 1931 to 1933 indicated the average size of outdoor dust particles to be 0.5 micron, as collected by the Owens jet dust counter and measured under the microscope. Inability of the *light field* microscope to reveal particles in the 0.1 micron vicinity may have influenced the determination of average particle size.

Table 1. Relation of Screen Mesh to Particle Size

U. S. Standard Sieve Mesh	400	325	200	140	100	60	35	18
Nominal Sieve Opening in Microns	37	44	74	105	149	250	500	1000

The lower limit of particle size visible to the naked eye cannot be stated definitely. It depends not only upon the individual eye, but also upon the shape and color of the particle, intensity and quality of the light, and nature of the background or opportunity for contrast. Under ideal conditions a particle of 10-micron size might be recognized, while under less favorable conditions it may be impossible to distinguish a particle smaller than 50 microns. The lower limit of visibility probably ranges from 10 to 50 microns.

Dusts, powders and granular materials are frequently classified by reference to the size of screens used for separation. Particles above 40 microns are said to be the *screen sizes* and those below, the *sub-screen* or microscopic sizes. Approximate or theoretical sizes of particles corresponding to the mesh scale of the U. S. Standard Sieve Series are given in Table 1.

Microscopic examination of screened dust indicates that the average diameter of a sample of irregular particles may be substantially larger than the openings of the screen through which it has passed, if the particle shapes deviate considerably from the spherical form[2]. The smallest dimension of many such particles will correspond with the maximum permissible distance between the wires of commercial screens made to *ASTM* Standard specifications. Screening does not give sharp separation into size groups, and accordingly such a classification is statistical rather than absolute.

AIR POLLUTION BY SMOKE, ASH AND CINDERS

Total airborne solids settling in urban areas are usually reported as *soot fall* in tons per (square mile) (month). Such data published for the cities in this country range from 20 to 200 tons per (square mile) (month). To the air conditioning engineer this information may indicate the effectiveness of smoke abatement or fuel combustion control methods in his locality, but it does not provide a suitable index of the suspended dust that air cleaners in a ventilating system are expected to capture[3, 4, 5]. Gravimetric or weight data of the type given in Table 2 are preferable. In some cases airborne particle counts may be necessary, as for pollen, bacteria, spores, and insoluble dusts causing illness or lung disease.

Dust concentrations by weight cannot be converted readily to concentrations by particle count because of the variability of particle size, shape

Table 2. Dust Concentration Ranges

Location	Grains per 1000 Cu Ft[a]	Milligrams per Cubic Meter
Rural and suburban districts	0.02–0.2	0.05– 0.5
Metropolitan district	0.04–0.4	0.1 – 1.0
Industrial districts	0.1 –2.0	0.2 – 5.0
Ordinary factories or workrooms	0.2 –4.0	0.5 –10
Excessively dusty factories or mines	4–400	10–1000
Minimum explosive concentrations	4000–200,000	10,000–500,000

[a] 1 grain per 1000 cu ft = 2.3 milligrams per cubic meter.
1 oz per cubic foot = 1 gram per liter = 1000 grams per cubic meter.

and specific gravity, and the inherent characteristics of dust counting and weighing procedures. One milligram of dust per cubic meter of air may represent dust counts from 1 million to 100 million particles per cubic foot of air (lightfield microscope technic) according to the size distribution of the airborne dust sample. Information of this type for a specified application is best obtained by simultaneous sampling for both counting and weighing and noting carefully at the time all factors that might affect the reproducibility of the count-weight ratio.

Smoke Abatement

Successful abatement of atmospheric pollution caused by smoke requires the combined efforts of the combustion engineer, industrial executive, public health officer, city planning commission and the community at large. Electrification of industry and railroads, increases in the use of domestic oil and gas furnaces, and segregation of industrial districts is gradually providing effective aid in the solution of this problem. In the large cities where nuisance from smoke, fly-ash and cinders is more serious, limited areas obtain some relief by the use of district heating. (See Chapters 16, 17 and 18 for further discussion on fuel burning technic.)

Legislative measures at the present time are largely concerned with reduction of the visible smoke discharged from chimneys. Practically all ordinances limit the number of minutes in any one hour that smoke of a specified density may be discharged, as measured by comparison with a Ringelmann Chart (Chapter 11, Instruments and Measurements). Ordinances generally do not make specific provision for control of the corrosive and irritant gases, such as oxides of sulphur and nitrogen discharged with the gases of combustion. Where high sulfur coals are burned, sulfur gases present a serious hazard to property, vegetation, and, in some cases, to health and comfort.

In foggy weather the accumulation of these gases in the lower strata of the atmosphere may cause irritation of the eyes, nose and respiratory passages, and possibly even more dangerous consequences. The Meuse Valley fog disaster (Belgium 1930) is a classic example in the history of gaseous air pollution. It is believed that sulfur dioxide, sulfur trioxide and other toxic gases released in a rare combination of atmospheric calm and dense fog were responsible for the 63 human deaths, the illness of several hundred persons, and also the death of many domestic and wild animals.

Absorption of Solar Radiation

Absorption of solar ultraviolet light by smoke and soot is recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[6] by actinic methods demonstrated that ultraviolet light intensity in the country was 50 per cent greater than in the city. In New York City[7] a loss as great as 50 per cent in visible light was found by photoelectric measurements.

ODOR NUISANCE

A problem companionate with smoke abatement is the control of odor nuisance in the neighborhood of industrial plants discharging noxious or offensive air contaminants. Community planning and zoning will avoid much of the difficulty in the future, but meanwhile many industrial cities must resort to corrective measures by requiring installation of air cleaning devices, the alteration of manufacturing processes, or termination of the offensive operation in residential or commercial districts.

3. Physiological Response to Gases and Vapors[a]

Concentrations in Parts of Substance per Million Parts of Air by Volume (ppm)

Substance	Rapidly Fatal	Dangerous to Life in ½ to 1 Hr	Maximum Allowable Concentration for Daily Exposures
Acrolein	2,000	100	0.5
Ammonia	5,000	2,500	100
Amyl acetate	....	....	200
Aniline	....	200	5
Arsine	250	10	0.05
Benzene (benzol)	20,000	5,000	100[b]
Bromine	500	40	1
Butyl acetate	....	10,000	200
Carbon dioxide	100,000	50,000	5,000
Carbon disulfide	2,000	1,000	20[b]
Carbon monoxide	4,000	1,000	100[b]
Carbon tetrachloride	50,000	....	50
Chlorine	1,000	10	1
Dichlorobenzene	....	....	50
Dichloroethyl ether	....	500	15
Ether (diethyl)	....	35,000	400
Ethyl acetate	40,000	10,000	400
Ethyl alcohol	....	....	1,000
Ethylene dichloride	....	4,000	100
Formaldehyde	....	....	10[b]
Gasoline	....	....	500
Hydrogen chloride	1,000	....	10
Hydrogen cyanide	200	100	20
Hydrogen fluoride	....	50	3
Hydrogen sulfide	600	200	20[b]
Methyl acetate	....	....	200
Methyl alcohol (Methanol)	....	....	200[b]
Methyl bromide	20,000	2,000	20
Methyl chloride	150,000	20,000	100
Methylene chloride	....	....	500
Monochlorobenzene	....	....	75
Nitrobenzene	....	....	1
Nitrogen oxides	300	100	25[b]
Phosgene	50	5	1
Phosphine	1,000	400	0.05
Styrene	....	....	400[b]
Sulfur dioxide	400	150	10
Tetrachloroethane	7,000	....	5
Tetrachloroethylene	....	....	100
Toluene (toluol)	20,000	5,000	200
Trichloroethylene	....	....	200[b]
Turpentine	....	....	100
Xylene (xylol)	20,000	5,000	200[b]

[a] Adapted from: Manual of Industrial Hygiene, by W. M. Gafafer et al, U. S. Public Health Service (W. B. Saunders Co., 1943); Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by M. B. Jacobs (Interscience Publishers, 1941); Noxious Gases, by Henderson and Haggard (Reinhold Publishing Co., N. Y., 1943). Report of the Committee on Threshold Limits of the American Conference of Governmental Industrial Hygienists; and other authoritative sources.

[b] Adopted by the *American Standards Association* (American Standard Z-37)]

Control of outdoor odor nuisance is especially troublesome because of the extremely minute quantities of contaminant that are capable of offending through a wide area. New industrial chemicals with strange or unfamiliar odors tend to receive more attention from the neighborhood than the customary odors generated by well-known processes and raw materials. Methods of odor control currently in use include charcoal adsorption, scrubbing towers and air washers, chlorination, condensation, masking, passage of the odorous air through combustion chambers, and best of all, substitution of less offensive materials whenever possible[8, 9, 10, 11].

Control of air quality within buildings ventilated for human occupancy is discussed in Chapter 12. Tobacco smoke odors, cooking odors and body odors are air contaminants of the nuisance type which now command a decisive position in the standards of air quality for indoor comfort. However, the engineer will find, at times, that odors originating *outside* buildings in industrial or business districts may determine the kind and capacity of equipment he must provide for a high quality air supply installation.

TABLE 4. MAXIMUM ALLOWABLE CONCENTRATIONS OF DUSTS, FUMES AND MISTS[a]

SUBSTANCE	MILLIGRAMS PER CUBIC METER, DAILY EXPOSURES[b]
Antimony	0.1
Arsenic; arsenic trioxide	0.15[c]
Cadmium, and compounds	0.1[c]
Chlorodiphenyls	1.0
Chromic acid mist and Chromates (as CrO_3)	0.1
Fluorides	2.5
Lead; lead carbonate; lead chloride; lead nitrate; lead oxides; lead sulfate	0.15[c]
Manganese, and compounds	6.0[c]
Mercury, and compounds	0.1
Pentachloronaphthalene	0.5
Selenium	0.1
Tellurium	0.1
Trichloronaphthalene	5.0
Trinitrotoluene	1.5
Zinc oxide fume	15.0

[a] Recommendations of state and local industrial hygiene agencies and other authoritative sources.
[b] 1 milligram per cubic meter = 0.44 grain per 1000 cu ft.
[c] Adopted by the *American Standards Association* (American Standard Z-37).

INDUSTRIAL AIR CONTAMINANTS

Many industrial processes are sources of contaminants. Their control is an important function of the ventilating or air conditioning engineer, because the atmosphere within buildings is the medium whereby such finely divided matter is dispersed and transported from the source to remote locations where it may cause property damage, nuisance, fire, explosion, disease and even death.

Tables 3, 4 and 5 give the maximum allowable concentrations for industrial air contaminants as currently accepted in most sections of the country. They apply to *exposures of 8 hours per day*, and refer to the quantities of contaminant permissible in the *workers' breathing zone.* Some of these figures may be altered as the result of continuous research, and some may differ from those in force in a few cities or states. The

prudent engineer will design equipment using these values as the *upper* limits of air contamination, and will incorporate a reasonable margin of safety in his estimates of ventilation capacity.

Information on the properties and effects, with respect to health, of specific industrial air contaminants is available in publications listed at the end of this chapter.

TABLE 5. MAXIMUM ALLOWABLE CONCENTRATIONS OF DUSTS[a]

Substance	Million Particles per Cubic Foot of Air, Daily Exposures[b]
Aluminum oxide abrasive	15–100
Asbestos	5
Carborundum (silicon carbide)	15–100
Cement (Portland)	50–100
Coal (less than 5 per cent quartz)	50–100
Dusts containing less than 10 per cent *free* silica	10–100
Granite	10– 25
Gypsum (hydrated calcium sulfate)	50–100
Limestone (calcium carbonate)	50–100
Marble (calcium carbonate)	50–100
Mica	10–100
Nuisance dusts (non-toxic, non-silica)	50–100
Quartz (silicon dioxide)	5
Sand (silica, silicon dioxide)	5
Sandstone	5
Silica (*free* or uncombined silicon dioxide)	5
Silicates (combined silicon dioxide)	15–100
Slate	15–100
Talc	10– 50
Total (maximum concentration for mixed dusts)	50–100

[a] Recommendations of state and local industrial hygiene agencies compiled by the American Conference of Governmental Industrial Hygienists; and other authoritative sources.
[b] Includes only particles from 1 to 10 microns approximately, as determined by the light field microscope counting technic, using the 10X objective. Dark field counts (and the corresponding allowable concentrations) are anywhere from 2 to 100 times the light field counts for the same sample, according to the proportion of dust smaller than 1 micron (See Industrial Dust, Chapter VII, by Drinker and Hatch, McGraw Hill Book Co.).

FLAMMABLE GASES AND VAPORS

Adequate ventilation is a primary requirement for minimizing the hazard of fire or explosion due to gases and vapors. The need for good ventilation is not removed by the use of other precautions, such as the elimination of known ignition sources, segregation of hazardous operations, adoption of safe building construction, and installation of automatic alarms. Some safety engineers regard overventilation of an operation employing flammable liquids as a legitimate operating charge for the privilege or necessity of using a dangerous process. However, it is not possible to apply a reasonable safety factor to the ventilation estimate without consideration of the concentrations of gases or vapors that approach the danger point. Safety engineers prefer to limit the concentration to ¼ or ⅕ of the lower explosive limit, and this fact should be given full weight in determining the capacity and design of ventilating equipment. Rarely should consideration be given to *operation above the upper explosive limit* in the open areas of buildings or rooms—even though unoccupied—because the

danger of temporary drop of gas concentration to a point within the explosive range is too great.

Ability of a flammable liquid to form explosive mixtures is determined largely by its vapor pressure, volatility, or rate of evaporation. ***Flash point*** is a convenient method of expressing this property in terms of the temperature scale. It may be defined as the temperature to which a combustible liquid must be heated to produce a flash when a small flame is passed across the surface of the liquid. The higher the flash point, the more safely can the liquid be handled. Liquids with flash points under 70 F should be regarded as highly flammable.

Upper and lower limits of flammability of gases and vapors, and the flash points of the corresponding liquids are given in Table 6.

Methods for estimating the flammable limits of *mixtures of* gases or vapors must be applied with caution; the reader is referred to other publications for this information[12, 13].

Design of equipment for the control of combustible anesthetics is outlined in Chapter 13. Construction of equipment for handling air containing flammable substances, or operating in atmospheres so contaminated, is discussed in Chapter 46.

It is customary to report the concentrations of flammable gases or vapors in per cent by volume, or *volume per cent.* Comparison with concentrations on the *part per million* scale used in chemical, medical or industrial hygiene literature is readily made by the conversion: 1 per cent = 10,000 ppm (parts of contaminant per million parts of air, by volume, or in other words, cubic feet of contaminant per million cubic feet of air). It will be noted in Table 6 that nearly all of the substances listed have lower explosive limits above 1.0 per cent, while the *maximum allowable concentrations* for gases and vapors in Table 3 are below 1000 ppm or 0.1 per cent in most cases. Therefore, control of toxic or injurious vapors to levels below their maximum allowable concentrations for health usually requires much more effective ventilation than for the prevention of a fire hazard.

COMBUSTIBLE DUSTS

A dust explosion is essentially a sudden pressure rise caused by the very rapid burning of airborne dust. The primary explosion often originates from a small amount of dust in suspension exposed to a source of ignition, and the pressure and vibration it creates may be sufficient to dislodge large accumulations of dust on horizontal ledges or surfaces of the building and equipment, thereby creating a secondary explosion of great force. Thus the air conditioning engineer is involved for two reasons: (1) to obtain a movement of dust-laden air into exhaust hoods or openings and through ventilating or pneumatic conveying ducts in a manner that will prevent accumulation of highly flammable dust at points where it could ignite inside the equipment; and (2) to so design *process ventilation* as to prevent the escape of dust which might settle on horizontal surfaces and become a potential source of disaster at some distance from the dusty operation. (See Chapter 46).

Intensity of a dust explosion depends upon: chemical and thermal properties of the dust; particle size and shape; concentration in air; proportion of inert dust in the air; moisture content and composition of the air; size and temperature of the ignition source; and degree of dispersion of the dust cloud. Investigations on the explosibility of dusts require determination of the maximum pressure developed during explosion of a known air concentration, as well as determination of the rate of pressure rise. In-

TABLE 6. APPROXIMATE LIMITS OF FLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

Gas or Vapor	Lower Limit Per cent by Volume	Upper Limit Per cent by Volume	*Closed Cup* Flash Point F Deg[c]
Acetaldehyde	4.0	57	−17
Acetone	2.1	13.0	0
Acetylene[b]	2.5	80	....
Allyl alcohol	2.4	....	70
Ammonia	16.0	27.0	....
Amyl alcohol	1.2	....	91
Amyl chloride	1.4	....	....
Amylene	1.6	....	....
Benzene (benzol)	1.4	8.0	12
Benzyl chloride	1.1	....	140
Butane	1.6	8.5	−76
Butyl acetate	1.7	15.0	72
Butyl alcohol	1.7	....	84
Butylene	1.7	9.0	....
Carbon disulfide	1.0	50	−22
Carbon monoxide	12.5	74	....
Crotonaldehyde	2.1	15.5	55
Cyclohexane	1.3	8.4	1
Cyclopropane	2.4	10.3	....
Decane	0.67	2.6	115
Dichloroethylene (1, 2)	9.7	12.8	43
Diethyl selenide	2.5	....	....
Dioxan	2.0	22.2	65
Ethane	3.1	15.0	....
Ether (diethyl)	1.7	48.0	−20
Ethyl acetate	2.2	11.5	24
Ethyl alcohol	3.3	19.0	55
Ethyl bromide	6.7	11.3	....
Ethyl cellosolve	2.6	15.7	104
Ethyl chloride	3.6	14.8	−58
Ethylene	3.0	34.0	....
Ethylene dichloride	6.2	15.9	56
Ethyl formate	2.7	16.5	−4
Ethyl nitrite	3.0	....	−31
Ethylene oxide	3.0	80	....
Furfural (125 C)	2.1	....	140
Gasoline	1.3	6.5	−50
Heptane	1.0	6.0	25
Hexane	1.2	6.9	−7
Hydrogen cyanide	5.6	40.0	0
Hydrogen	4.1	74	....
Hydrogen sulfide	4.3	45.5	....
Illuminating gas	5.3	31.0	....
Iso-butyl alcohol	1.7	....	82
Iso-pentane	1.3	....	....
Iso-propyl acetate	1.8	7.8	43
Iso-propyl alcohol	2.5	....	53
Methane[b]	5.0	15.0	....

[a] Adapted from: Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones (U. S. Bureau of Mines, Bulletin No. 279, 1939); Properties of Flammable Liquids, Gases and Solids (Associated Factory Mutual Fire Ins. Cos., January, 1940); and National Fire Codes for Flammable Liquids, Gases, Chemicals and Explosives—1945 (*National Fire Protection Association*).
[b] Turbulent mixture.
[c] *Closed cup* refers to the equipment used in flash point determinations.

Table 6. Approximate Limits of Flammability of Single Gases and Vapors In Air at Ordinary Temperatures and Pressures[a] (Continued)

Gas or Vapor	Lower Limit Per cent by Volume	Upper Limit Per cent by Volume	*Closed Cup* Flash Point F Deg
Methyl acetate	3.1	15.5	15
Methyl alcohol	6.0	36.5	54
Methyl bromide	13.5	14.5	....
Methyl butyl ketone	1.2	8.0	....
Methyl chloride	8.0	19.7	....
Methyl cyclohexane	1.1	....	25
Methyl ethyl ether	2.0	10.1	−35
Methyl ethyl ketone	1.8	11.5	30
Methyl formate	5.0	22.7	−2
Methyl propyl ketone	1.5	8.2	....
Natural gas	4.8	13.5	....
Naphtha (benzine)	1.1	6.0	20–45
Naphthalene	0.9	....	174
Nonane	0.74	2.9	88
Octane	0.84	3.2	56
Paraldehyde	1.3	....	....
Pentane	1.4	8.0	....
Propane	2.4	9.5	....
Propyl acetate	1.8	8.0	58
Propyl alcohol	2.5	....	59
Propylene	2.0	11.1	....
Propylene dichloride	3.4	14.5	59
Propylene oxide	2.1	21.5	....
Pyridine (70 C)	1.8	12.4	68
Toluene (toluol)	1.3	7.0	40
Turpentine	0.8	....	95
Vinyl ether	1.7	27.0	....
Vinyl chloride	4.0	22.0	....
Water gas (variable)	6.0	70	....
Xylene (xylol)	1.0	6.0	63

vestigators frequently experience difficulty in obtaining dust suspensions of uniform dispersion, and this should be kept in mind when comparing results from several sources[14].

Minimum explosive concentrations of airborne dusts already tested range from 0.01 to 0.5 oz per cubic foot, or 10 to 500 grams per cubic meter of air. Maximum pressures generated have been reported as high as 500 psi, although they are more likely to be of the order of 50 psi. Investigations on the flammable characteristics of dusts are currently made at 0.1 and 0.5 oz per cubic foot[15–21].

ATMOSPHERIC POLLEN

Properties of pollen grains discharged by weeds, grasses and trees and responsible for hay fever are of special interest to designers of air cleaning equipment (see Allergic Disorders in Chapter 13, and Air Cleaning Devices, Chapter 33). Whole grains and fragments transported by the air range chiefly between 10 and 50 microns in size, but some have been measured as small as 5 microns and others over 100 microns in diameter. Ragweed pollen grains are fairly uniform in size within the range of 15 to 25 microns. Pollen grains can be removed from the air more readily than the particles

of dust prevalent in outdoor air or produced by dusty processes, since the latter predominate in the range of 0.1 to 10 microns in size.

Most grains are quite hygroscopic and therefore vary in weight with the humidity. Illustrations and data on individual pollen grains are available in the botanical literature[22, 23, 24]. Geographical distribution of plants known to produce hay fever is also recorded[25, 26].

The quantity of pollen grains in the air is generally estimated by exposing an adhesive-coated glass plate outdoors for 24 hr and then counting calibrated areas under the microscope. Methods are available for determining the number of grains in a measured volume of air[25, 27, 28] but their greater accuracy has not caused them to replace the more simple *gravity slide method* used for most pollen counts. Counting technics vary somewhat, but the daily pollen counts reported in local newspapers during the hay fever season usually represent the number of grains found on 1.8 sq cm of a 24-hr *gravity slide.*

Hay fever sufferers may notice the first symptoms when the pollen count is 10 to 25, and in some localities the maximum figures for the seasonal peak may approach 1000 for a 24-hr period, depending upon the sampling and reporting methods of the laboratory. Translation of gravity counts by special formulas to a volumetric basis, or the number of grains per cubic yard or per cubic foot of air, is unreliable because of the complexity of the modifying factors. When such information is important, it is best obtained directly by a volumetric instrument. The number of pollen grains per cubic yard of air evidently varies from 2 to 20 times the number found on 1 sq cm of a 24-hr gravity slide, depending on grain diameter, shape, specific gravity, wind velocity, humidity and physical placement of the collecting plate[29, 30, 31].

AIRBORNE BACTERIA

Study of the occurrence and significance of micro-organisms in the atmospheres of the indoor world is absorbing the energies of a substantial number of physicians, bacteriologists, aerobiologists, physicists, public health workers, engineers and hospital personnel. Some data are avialable on the types and quantities of bacteria found in a variety of spaces, but it is not possible at present to use this information as a conclusive index of the potential health hazard of a given environment. The reported number of airborne organisms may vary from 1 to 1000 per cubic foot of air, influenced somewhat by the method of testing[32]. Many are attached to the dust particles present in the air.

Where it seems advisable or desirable to control the bacterial content of rooms, public conveyances or buildings, highly effective methods are available (see Chapter 13), and their extended use may do much to assist the workers in this field in accumulating the necessary mass of evidence that will decide the practical value of air sterilization for the control of communicable disease. It is now well established that ultraviolet radiation is feasible for the protection or preservation of pharmaceuticals, cosmetics, and food products.

REFERENCES

[1] Atmospheric Pollution of American Cities for the Years 1931 to 1933, J. E. Ives et al (*U. S. Public Health Service Bulletin No.* 224, March 1936).

[2] Micromeritics, The Technology of Fine Particles, by J. M. DallaValle (Pitman Publishing Corporation, 1943).

[3] Atmospheric Pollution Due to Smoke, by A. C. Stern (*Heating and Ventilating*, May, 1945).

Atmospheric Pollution Due to Dust and Cinders, by A. C. Stern (*Heating and Ventilating*, July, 1945).

[4] Sootfall Studies for New York City, by J. Siegel and B. Feiner (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Septemper, 1945, p. 495).

[5] The Use of Fuel Consumption and Equipment Data in the Abatement of Atmospheric Pollution, by A. C. Stern (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1945, p. 447–454).

[6] Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz and F. A. Korff (*American Journal of Public Health*, Vol. 19, 1929, p. 7).

[7] Studies in Illumination—III: A Study of the Loss of Light Due to Smoke on Manhattan Island, by J. E. Ives (*U. S. Public Health Service Bulletin No.* 197, June, 1930).

[8] Study and Control of Industrial Atmospheric Pollution Nuisances, by F. M. Stead (*American Journal of Public Health*, Vol. 35, May, 1945, p. 491–498).

[9] Evaluation of Odor Nuisance in the Manufacture of Kraft Paper, by J. M. DallaValle and H. C. Dudley (*U. S. Public Health Service* Reprint No. 2022, Public Health Reports, Vol. 54, January 13, 1939, p. 35–43).

[10] Disposal of Refinery Wastes, Section II: Waste Gases, Vapors, Sludges and Dusts (*American Petroleum Institute*, New York City, 1938).

[11] Offensive Trades, by David Ronald (William Hodge and Co., London, 1935).

[12] Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones (*U. S. Bureau of Mines*, Bulletin No. 279, 1939).

[13] Inflammation Limits and Their Practical Application in Hazardous Industrial Operations, by G. W. Jones (*Chemical Reviews*, Vol. 22, February, 1938).

[14] Private Communication, by Hylton R. Brown (*Bureau of Mines*, College Park, Maryland).

[15] Explosibility of Agricultural and other Dusts as Indicated by Maximum Pressure and Rates of Pressure Rise, by P. W. Edwards and L. R. Leinbach (*U. S. Department of Agriculture Technical Bulletin No.* 490, October, 1935).

[16] Dust Explosion Hazards in Plants Producing or Handling Aluminum, Magnesium, or Zinc Powder, by H. R. Brown (*U. S. Bureau of Mines Information Circular No.* 7148, March, 1941).

[17] Inflammability and Explosibility of Metal Powders, by I. Hartman, J. Nagy, and H. R. Brown (*U. S. Bureau of Mines Report of Investigation No.* 3722, October, 1943).

[18] Inflammability and Explosibility of Powders Used in the Plastics Industry, by I. Hartman and J. Nagy (*U. S. Bureau of Mines Report of Investigation No.* 3751, May, 1944).

[19] Industrial Dust Explosions, by H. R. Brown (*U. S. Bureau of Mines*, Information Circular No. 7309, January, 1945).

[20] Proposed Code for the Prevention of Dust Explosions in the Plastics Industry (*National Fire Protection Association*, Boston, May, 1945).

[21] National Fire Codes for the Prevention of Dust Explosions (*National Fire Protection Association*, Boston, 1944). Contains codes for aluminum, magnesium, coal, pulverized fuel, flour, spice, starch, sugar, cocoa, sulfur and wood.

[22] An Introduction to Pollen Analysis, by G. Erdtman (Chronica Botanica Co., Waltham, Mass., 1943).

[23] Pollen Grains, by R. P. Wodehouse (McGraw Hill Book Co., New York, 1935).

[24] Atmospheric Pollen, by R. P. Wodehouse (*Aerobiology*, p. 8–31, Publication No. 17, *American Association for the Advancement of Science*, Washington, D. C., 1942).

[25] Hayfever Plants, by R. P. Wodehouse (Chronica Botanica Co., Waltham, Mass., 1945).

[26] Hay Fever: A Geographical and Botanical Survey, by E. R. Squibb and Sons, New York, 1937.

[27] Techniques for Appraising Air-Borne Populations of Microorganisms, Pollen and Insects (*Phytopathology*, Vol. 31, March, 1941, p. 201–225).

[28] Apparatus for Determining the Pollen Concentration of the Atmosphere, by B. J. Cody, W. F. Kinney and N. A. Kerstein (Research Department, the Detroit Edison Company, Detroit).

[29] The Volumetric Incidence of Atmospheric Allergens, by O. C. Durham (*Journal of Allergy*, Vol. 14, September, 1943, p. 455–461).

[30] The Volumetric Incidence of Atmospheric Allergens, II: Simultaneous Measurements by Volumetric and Gravity Slide Methods, by O. C. Durham (*Journal of Allergy*, Vol. 15, May, 1944, p. 226–235).

[31] Air-Borne Fungus Spores as Allergens, O. C. Durham (*Aerobiology*, p. 32–47, Publication No. 17, *American Association for the Advancement of Science*, Washington, D. C., 1942).

[32] Sampling Devices, by H. G. DuBuy and A. Hollaender (*American Journal of Medical Science*, Vol. 209, February, 1945, p. 172–177).

BIBLIOGRAPHY

Abstracts and Bulletins (monthly, annual and special) Industrial Hygiene Foundation, Inc., Pittsburgh, Pa.

Aerobiology, Publication No. 17 (*American Association for the Advancement of Science*, Washington, D. C., 1942).

Air Sanitation and Industrial Ventilation, by W. N. Witheridge (Detroit, Mich., 1945).

Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by M. B. Jacobs (Interscience Publishers, New York, 1941).

Bibliography of Industrial Hygiene, 1900–1943 (*U. S. Public Health Service Bulletin* No. 289, 1945).

Clouds and Smokes, by W. E. Gibbs (P. Blakiston's Son & Co., Philadelphia, Pa., 1924).

Determination and Control of Industrial Dust, by Bloomfield and DallaValle (*U. S. Public Health Service Bulletin* No. 217, 1935).

Dust, by S. C. Blacktin (The Sherwood Press, Cleveland, 1934).

(The) Environment and Its Effect upon Man (Harvard School of Public Health, Boston, 1937).

Industrial Dust, by Drinker and Hatch (McGraw Hill Book Co., New York, 1936).

Journal of Industrial Hygiene and Toxicology (monthly) (Harvard School of Public Health, Boston, Mass.). See cumulative abstract and subject indexes.

Manual of Industrial Hygiene, by W. M. Gafafer et al (*U. S. Public Health Service*, W. B. Saunders Co., Philadelphia, 1943).

Noxious Gases and the Principles of Respiration Influencing their Action, by Henderson and Haggard (*American Chemical Society Monograph Series No.* 35, Reinhold, New York, 1943).

Occupation and Health, two volumes (*International Labor Office*, Washington, D. C.).

Toxicology and Hygiene of Industrial Solvents, by Lehmann and Flury, translated by Eleanor King and H. F. Smyth, Jr. (Williams and Wilkins, Baltimore, 1943).

CHAPTER 11

INSTRUMENTS AND MEASUREMENTS

Temperature Measurement, Pressure Measurement, Measurement of Air Movement, Air Change Measurements, Measurement of Relative Humidity, Dust Determination, Heat Transfer Through Building Materials, Measurement of Heat Exchange for Comfort Conditions, Combustion Analysis, Smoke Density Measurements, Sound and Vibration Measurements

HEATING and air conditioning engineers and technicians require instruments for both laboratory and field use and somewhat more precision is attainable and essential in the laboratory, where research and development are undertaken, than in the field, where acceptance and adjustment tests are conducted. Some instruments have attained an adequate state of development while others fall far short of the desirable. For instance temperatures can now be measured readily with ample accuracy for most purposes while a convenient and precise method for determining or comparing the dustiness of atmospheres is lacking and improvement in existing hygrometers and humidity controllers is essential.

Codes and standards covering different types of heating and air conditioning devices and apparatus have been promulgated by various authoritative organizations and instruments essential for performance or compliance testing are enumerated in the relevant publications.[1, 2, 3, 4] The present purpose is to discuss the use and characteristics of the more important instruments.

TEMPERATURE MEASUREMENT[5, 6]

Thermometers

Any device capable of indicating temperature is a thermometer but in common usage the term *thermometer* without qualification has come to signify the ordinary mercury-in-glass temperature indicating device. This type has a useful range from −40 F, the freezing point of mercury, to about 1000 F, at or near which the glass usually softens. Lower temperatures can be measured with alcohol-filled thermometers for which the range is about −94 F to +248 F. The better thermometers have their scales, either Fahrenheit or Centigrade, etched with acid into the glass which forms their stems. The probable error for etched stem thermometers is plus or minus one scale division, and calibration is desirable for much test work. Thermometers are calibrated during manufacture at not less than two temperatures—the freezing and boiling points of water—and calibration is often accomplished with the instrument completely immersed in a bath at the known temperature. The intervening scale divisions are then applied to the stem. When such a thermometer is used with the stem incompletely immersed, a correction known as the stem correction is necessary for accurate measurements, and its magnitude is usually computed by means of the following formula:

$$K = 0.00009\, D\, (t_1 - t_2) \qquad (1)$$

where

K = correction to be added, Fahrenheit degrees.

D = number of degrees on the thermometer scale which are not immersed.
t_1 = temperature indicated on the thermometer, Fahrenheit degrees.
t_2 = temperature of the non-immersed mercury column, Fahrenheit degrees.
0.00009 = difference in the coefficient of expansion of the mercury and glass.

When a thermometer is used in a liquid or in air or gas near room temperature the effects of radiation can often be ignored, but when the temperature of hot air or gas is desired, means are usually provided for minimizing the effect of radiation.[7] These include bright metallic shields around the thermometer bulb and the use of aspirated thermometers in which a stream of the air or gas is drawn at considerable speed across the bulb. This increases the influence of the gas temperature on the thermometer indication. In any case, to prevent errors in temperature measurements, there should be ample circulation so that the thermometer will indicate a true temperature of the medium under observation, and ample time should be allowed for the thermometer to reach the same temperature as the medium. In reading a thermometer the eye should be at the same level as the top of the liquid to avoid parallax.

Industrial-type thermometers are available for permanent installation in pipes or ducts. These instruments are fitted with metal guards to prevent breakage and are useful for many purposes. However, the considerable heat capacity and conductance of the guards or shields do not permit such thermometers to follow the fluctuations in a medium of varying temperature as well as those of the etched stem type.

Thermocouples

When two wires made of dissimilar metals are joined by soldering, welding or merely by twisting, a thermocouple or thermo-junction is formed and an electromotive force, which depends upon the temperature of the junction, is found to exist between the wires. When the wires are joined at two points a thermocouple circuit is formed, and if one junction is kept at a temperature different from the other, an electric current flows through the circuit due to the difference in emf developed by the two junctions. This phenomenon is employed for temperature measurements in thermocouple systems, one junction being ordinarily kept at a constant temperature, as in an ice bath, while the other junction is placed at a point at which it is desired to observe the temperature. In practice it is desirable to utilize emf to indicate temperature because, at small or zero current flow, the resistance of the circuit is unimportant. A high resistance millivolt meter is useful in some cases but the potentiometer yields better results. In the potentiometer the electromotive force generated by the thermocouples is balanced against an electromotive force from the battery so that observations are made with no flow of current through the thermocouple circuit. A conventional arrangement is illustrated in Fig. 1. The thermocouple leads A-B are so connected that their polarity opposes that of battery C. If the position of E on the graduated slide wire rheostat DF is adjusted until galvanometer G shows no current flowing, resistance DE will indicate directly the voltage generated by the thermocouple. In order to calibrate the instrument, switch S is thrown over to the standard cell circuit while rheostat R is adjusted so that the galvanometer shows zero current. Battery C then exerts the known voltage of the standard cell at DH.

The choice of materials for thermocouple wires is determined by the range of temperature to be measured. Up to about 600 F base metals such as iron-constantan or, preferably from the corrosion standpoint, copper-constantan are satisfactory and develop a relatively large emf of 40 to 60

microvolts per degree. Chromel-alumel couples are useful in the flue gas temperature range while platinum- (platinum-rhodium) couples are used for higher temperatures. Impurities make large differences in the performance of thermocouple wires and for this reason calibration of samples from each spool of wire is essential for precise work. Data on wire can usually be obtained from the manufacturer.

The act of adjusting rheostat D-F (Fig. 1) for zero current flow is known as balancing the potentiometer. Automatic self balancing instruments are on the market of both the indicating and recording types. They usually contain an automatically compensating cold junction to avoid the use of an ice bath; and special thermocouple wire is furnished with them from the factory.

With a suitable potentiometer, small wires serve as well for thermocouples as large ones and the fineness of the wires is limited only by consid-

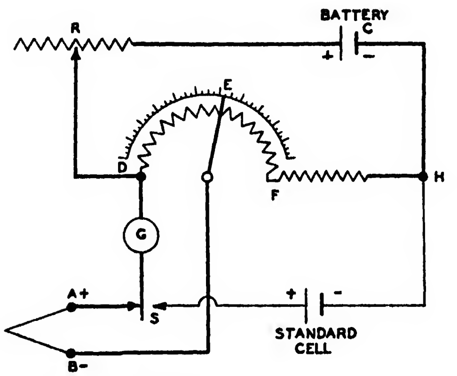

FIG. 1. BASIC CIRCUIT AND CONNECTIONS FOR THERMOCOUPLE AND POTENTIOMETER

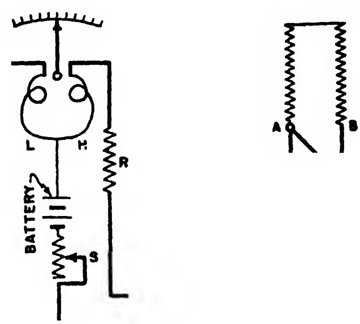

FIG. 2. TYPICAL RESISTANCE THERMOMETER CIRCUIT AND CONNECTIONS

eration of mechanical strength and convenience in handling. Small couples respond more promptly to changes in temperature and are less affected by radiation than large ones. For use in heated air or gases, thermocouples are often shielded as are thermometers and aspirated thermocouples are sometimes used. An arrangement has been described, also for avoiding error due to radiation, which involves several thermocouples of different sizes the true temperature being estimated by extrapolation to zero diameter[8].

By the use of thermocouples, temperatures at remote points may be indicated or recorded on conveniently located instruments; average temperature may be readily obtained by connecting several couples in parallel or in series; and temperatures may be obtained within thin materials, narrow spaces, or otherwise inaccessible locations.

Thermocouples in series with every alternate junction maintained at a common temperature will give an emf which, divided by the number of couples to give the average emf[9] per couple, may be used to find the average temperature.

Thermocouples in parallel having the similar metals of a number of couples connected together and run to a common cold junction will cause an indication on a potentiometer which is the true emf only if the electrical resistances of the parallel junctions are the same[9, 10].

Temperatures of surfaces below red heat are difficult to determine by any other means than thermocouples. For this purpose, a thermocouple of fine wires is preferable to minimize the possibility of error due to the conduction of heat along the wires. It may be attached to a metal surface in any of several ways. For permanent installations, soldering, brazing or peening may be desirable. A small hole is drilled for the peening operation; the thermocouple is inserted and the metal is peened to retain it. The fact that the thermocouple is in electric contact with the surface is unimportant in usual circuits. For temporary arrangements, couples may be attached by means of surgical or cellophane tape. For many boiler or furnace surfaces, furnace cement serves very well. The thermocouple may be attached by means of the cement when the surface is cold and must be treated gently and usually supported until the cement dries, due to heat, and hardens—after which it has ample strength. It is good practice to use as little cement as possible, and also to plaster the wires to the surface for an inch or so from their junction to avoid errors due to heat conduction along the wire. Electric insulation between the wires should be perfect except at the junction since, otherwise, the indicated emf will be between those existing at the junction and at the short circuit.

Resistance Thermometers

Resistance thermometers depend for their operation upon the change of electric resistance of metal with change in temperature. The resistance generally increases with rising temperature. Their use largely parallels that of thermocouples, although readings tend to be unstable above 950 F. Two-lead temperature elements are not recommended, since they do not permit correction for lead resistance. Three leads to each resistor are necessary to obtain consistent readings.

A typical circuit used by several manufacturers is shown in Fig. 2. In this design a differential galvanometer is used, in which coils L and H exert opposing forces on the indicating needle. Coil L is in series with the thermometer resistance AB, and coil H is in series with the constant resistance R. As the temperature falls, the resistance of AB decreases allowing more current to flow through coil L than through coil H. This causes an increase in the force exerted by coil L, pulling the needle down to a lower reading. Likewise, as the temperature rises the resistance of AB increases, causing less current to flow through coil L than through coil H. This forces the indicating needle to a higher reading. Rheostat S must be adjusted occasionally to maintain a constant flow of current.

As compared to the thermocouple the resistance thermometer does not require a cold junction, and it can be simply scaled for more accurate measurements; but, generally because of its construction it is more costly and is apt to have considerable lag. It gives best results when used to measure steady or slowly changing temperature. For accurate results the entire thermometer coil must be exposed to the temperature to be measured.

Pyrometers

The pyrometer is the usual instrument for measuring high temperatures such as those of incandescent bodies or furnace interiors. There are two types. In the radiation pyrometer the radiant energy from an observed surface falls on a thermopile and the emf generated by the pile, measured by a galvanometer or potentiometer, is an index of the surface temperature. With the optical pyrometer a narrow spectral band, usually red, emitted by the surface is matched visually with the filament of a special electric

lamp. The emf necessary to cause the filament to match the surface in brightness is the index of the temperature of the surface. Pyrometers are calibrated by means of various metals with known melting or freezing points. Portable as well as laboratory models are manufactured.

Color Indicating Crayons

Crayons are available, the marks of which change color or melt at specified temperatures. Such crayons have been sold in boxes covering the range from about 100 F to about 800 F in 100 deg steps with a precision of some 10 deg. They are a rough but convenient means of determining temperatures and of locating isothermal lines on surfaces below red heat.

PRESSURE MEASUREMENT

Pressure Gages

The Bourdon is the most common type of pressure gage and its appearance probably is familiar to any one having an acquaintance with power plants or laboratories. The essential element of such a gage is the Bourdon tube, a metal tube of oval cross section curved along its length to form almost a complete circle. One end is closed and the other is connected to the vessel in which the pressure is to be measured. With an increase of pressure, the tube tends to straighten, and vice versa, and the resulting motion of the closed end is communicated by suitable linkages to a needle moving over a graduated dial. If the range is above about 20 psi, such gages are usually calibrated by means of a *dead weight tester* whereby known pressures are produced in a fluid by imposing known weights on a piston of known area. Gages are commonly set to read accurately at or near the pressure of probable use. Gages of several different types or qualities are available on the market[11]. Suction gages and pressure gages with ranges below about 20 psi are ordinarily calibrated against mercury manometers.

Manometers

The manometer is a simple and useful means for measuring partial vacuum and low pressure. It is, moreover, a primary instrument; it does not require calibration, and it is often used as a standard for the calibration of other instruments. It is so universally used that both the inch of water and the inch of mercury have become accepted units of pressure measurement. In its simplest form, the manometer consists of a U-shaped glass tube partially filled with a liquid. A difference in height of the two fluid columns denotes a difference in pressure in the two legs which is proportional to the difference in height.

For converting manometer readings into other pressure units, certain proposed standard factors are applicable for precise work. These are based on a standard gravitational acceleration of 32.1740 ft per (sec) (sec) and are as follows:

$$\begin{aligned}\text{1 Standard Atmosphere} &= \text{14.696 lb per sq in.} = \text{29.921 in. mercury at 32 F}\\ &= \text{33.96 ft water column at 68 F}\end{aligned}$$

For most ordinary purposes, the following figures are amply accurate:

$$\begin{aligned}\text{1 Atmosphere} &= \text{14.7 lb per sq in.} = \text{29.9 in. mercury}\\ &= \text{34.0 ft (408 in.) water column}\end{aligned}$$

Manometer tubes should be chemically clean. The bore is not important except insofar as it affects the meniscus through wetting or surface tension. Bores of at least $\frac{3}{16}$ in. for rough and $\frac{1}{2}$ in. for more precise measurements

are recommended. Liquids other than water are sometimes used for low pressure measurement and, when this is done, the readings must be corrected for the density of the fluid used.

For measuring pressure differences of a few inches of water, or less, U-gages are often set at an angle for scale amplification. In many gages of this type, commonly termed draft gages or inclined manometers, only one tube of small bore is used and the other leg is replaced by a reservoir. The scale is calibrated to read in inches of water and it is necessary to use a fluid having the same gravity as that for which the gage was originally calibrated, or to apply a correction if another fluid is used. Such gages may be checked one against another. For more accurate calibration the gage may be checked against a micromanometer or a calibrating device known as a hook gage[12]. The accuracy of a draft gage is dependent on the slope of the tube, and consequently the base of the gage must be leveled carefully. It is not desirable to use a slope of less than 1 in 10. Where pressures are read under extreme conditions of temperature, and calibration is possible only at normal temperature, it is necessary to correct for the change in density of the liquid in the manometer[13]. For measuring low pressure differences to within 0.001 in. of water very sensitive micromanometers are available, such as the Illinois or Wahlen, the Meriam, the Trimount, and the Emswiler[14, 15].

When using a manometer or other pressure gage for measuring air flow, by means of orifices, the type of duct openings used for manometer connections and their location are important. Where velocities are low, as in some plenum chambers, or where the flow is free of large eddies and parallel to the walls of a duct, a drilled hole cleared of burrs and at right angles to the stream is satisfactory[16]. For higher velocities, it is common practice to provide four holes or taps around the periphery of the duct. Diametrically opposite pairs of taps are connected together, and then such pairs are manifolded together.

An alternate method involves the use of the Pitot tube, shown in Fig. 3. This instrument should be pointed up-stream, parallel with the air flow. Where the flow is not axial or parallel to the side walls of the duct a very close approximation of the static pressure and the flow direction can be obtained by a Fechheimer tube[17].

Barometer

The simplest and earliest type of barometer consists of a glass tube somewhat more than 30 in. long filled with mercury and inverted in a cup partially filled with mercury. The height of the mercury column in the tube above the mercury surface in the cup is a measure of the existing atmospheric pressure except for the slight pressure of the mercury vapor in the space above the mercury in the tube. This can ordinarily be ignored.

Elaborate mercury barometers are manufactured, fitted with vernier scales and, for precise work, corrections must be made for the thermal expansion of the mercury and of the scales[18]. The instruments are usually calibrated for 32 F mercury and 62 F scale temperature and the correction C to be subtracted from the observed barometer's height is obtained by means of Equation 2.

$$C = \frac{h(t - 28.630)}{(1.1123\,t - 10978)} \tag{2}$$

where

C = correction to be subtracted, inches of mercury.

h = observed height, inches of mercury.
t = observed temperature of the barometer, Fahrenheit degrees.

Standard atmospheric pressure at sea level is 29.921 in. Hg and since normal atmospheric pressure decreases about 0.01 in. Hg for each 10 ft increase in elevation, it is important to make a correction if the elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby Weather Bureau Station, in which case inquiry should be made as to whether the value is for *station* or *sea level* pressure.

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure deflect the thin surface of a sealed diaphragm capsule. Most commercially available aneroid barometers are not as accurate as the mercurial type, and the best require occasional recalibration. Open-scale aneroid barometers are more expensive than common mercurial barometers. Most of the pressure gages used in engineering work indicate gage pressures, that is, the

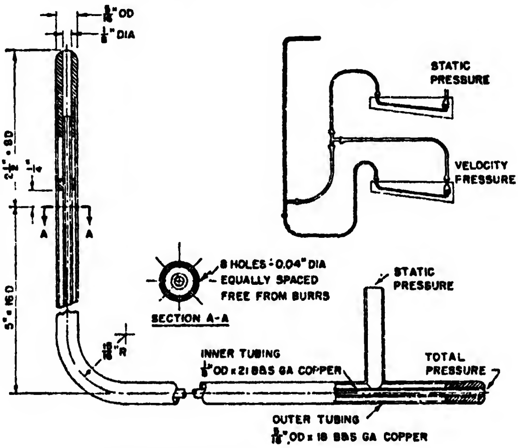

Fig. 3. Standard Pitot Tube

difference between the pressure being measured and the atmospheric pressure. Such pressures are called gage pressures. Absolute pressure may be obtained by adding barometric pressure and gage pressure algebraically.

AIR FLOW MEASUREMENTS

The theory of various means for measuring the flow of fluids is discussed in Chapter 4, Fluid Flow. Heating and air conditioning engineers are called upon to measure the flow of air more often than that of other gases, and usually the air is measured at or near atmospheric pressure. Under this condition the air can be treated substantially as an incompressible fluid which implies that simplified formulas can be used with sufficient accuracy for the solution of many problems[19].

The Pitot Tube

The construction of the Standard Pitot Tube[12] is shown in Fig. 3 and a formula (No. 73 from Chapter 4) used in conjunction with it, is as follows:

$$V_{fm} = 1096.5 \sqrt{\frac{h_{aw}}{\rho}} \qquad (3)$$

where

V_m = velocity, feet per minute.
h_{vw} = velocity pressure (Pitot tube manometer reading), inches of water.
ρ = density of air, pounds per cubic foot.

Since the velocity in a duct is seldom uniform across any section and since a Pitot tube reading indicates a velocity at only one location, a *traverse* of a duct is usual to determine the average velocity so that the flow can be computed. Suggested Pitot tube locations for traversing round and rectangular ducts are shown in Fig. 4. In general the velocity is lowest near the edges or corners, and greatest at or near the center. For this reason a large number of readings should be taken, in the case of round ducts not less than 20, along two diameters at centers of equal annular areas. In rectangular ducts the readings should be taken in the center of equal areas over the cross-section of the duct. The number of spaces should not be less than 16 and need not be more than 64. When less than 64 are taken the number of equal spaces should be such that the centers of the areas are not more than

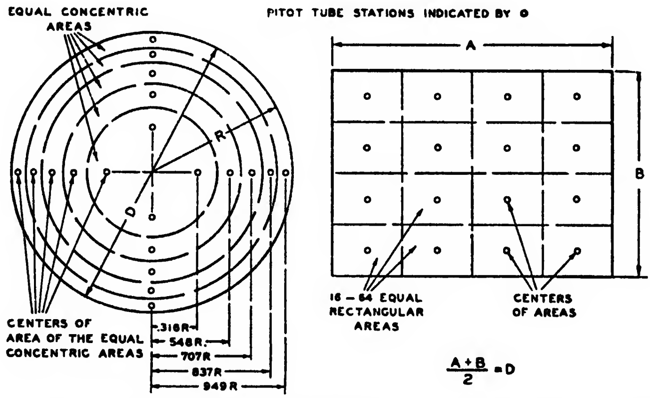

FIG. 4. PITOT TUBE TRAVERSE FOR ROUND AND RECTANGULAR DUCTS

6 in. apart. In determining the average velocity in the duct from the readings given, the calculated individual velocities or the square roots of the velocity heads must be averaged. It is incorrect to use the average velocity head for this purpose. Pulsating or disturbed flow will give erroneous results and therefore, if possible, the Pitot tube should be located at least $7\frac{1}{2}$ diameters down-stream from a disturbance such as that caused by a turn; or a criss-cross type of flow straightener should be installed in the duct $1\frac{1}{2}$ diameters ahead of the Pitot tube[12]. Flow straighteners do not equalize flow velocity across a duct. They merely serve to improve the precision of measurements. Equalization can be effected if desirable for measuring purposes by the use of wire netting, perforated plates or cloth screens across the duct.

Many forms of Pitot tubes other than the one described have been used and calibrated[20]. A double-ended tube[21], one end pointing down-stream, and one up-stream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight $\frac{1}{8}$ in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities in exhaust inlets, such as hoods placed around

grinding wheels. To meet special conditions, different sized Pitot tubes which are geometrically similar to the standard tube can be used.

Plate Orifices

Application of the Pitot tube is often inconvenient when velocities are low because the resultant velocity pressures become so small that extraordinary means are necessary for measuring them. In addition, velocity surveys of the whole cross-sectional area of a duct are inexpedient when numerous test runs are in prospect. Chiefly for these reasons orifices are favored for much test work. There are two types: the plate orifice and the shaped orifice or nozzle. Plate orifices are simple to construct and convenient to use in that a frame can be made to support them in the duct such that one can be removed and another inserted when it is desirable to use an orifice of a different size.

Formulas for Orifices

In the heating and air conditioning fields it is usually convenient to obtain orifice pressure drops in inches of water column, temperatures in Fahrenheit degrees and barometric pressure in inches of mercury. The air flow is usually desired in cubic feet per minute at the existing condition, so that velocities in various ducts can be computed, and in pounds per hour so that computations of heat transferred by the air can be based on weight, temperature change and specific heat. Equation 55, Chapter 4, is applicable and for most purposes; it can be altered for convenience to read as follows:

$$Q_M = 5.2KD^2 \sqrt{\frac{T_t}{B_t} h_w} \qquad (4)$$

where

Q_M = air flow, cubic feet per minute.
K = orifice coefficient.
D = orifice diameter, inches.
T_t = temperature of air at orifice, Fahrenheit degrees, absolute.
B_t = absolute pressure ahead of orifice, inches of mercury.
h_w = pressure drop through orifice, inches of mercury.

As most laboratories are less than 1000 ft above sea level precision is adequate in many cases if standard atmospheric pressure, 29.92 in. Hg, is assumed. Equation 4 then becomes:

$$Q_M = 0.95\, KD^2 \sqrt{T_t h_w} \qquad (5)$$

After the flow in cubic feet per minute is determined, it can be expressed in pounds of air per hour by means of the relation

$$W = 60 \frac{PQ_M}{RT_t} \qquad (6)$$

P = pressure, pounds per square inch, absolute.
R = 53.3, the gas constant for air.
T_t = absolute temperature of the flowing air, Fahrenheit degrees, absolute.

The thin-plate square-edged orifice often has a discharge coefficient K near 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the

edge[22,23]. Other information on orifices and their use is contained in Chapter 4, Fluid Flow.

Shaped orifices or nozzles have the advantage if well made that their discharge coefficients are close to unity so that the probability of large errors is less. Orifices of this type have been adopted for several specific purposes and designs are described in the A.S.H.V.E. Unit Heater[1] and Unit Ventilator Codes[2] and in A.S.R.E. Circular 13[3] entitled "Standard Methods of Rating and Testing Air Conditioning Equipment". In some instances nozzles are used in multiple so that the capacity of the testing equipment can be changed by shutting off the flow through one or more nozzles. An apparatus designed for testing the air flow and capacity of air conditioning equipment is described by Wile[24] in an article in which pertinent information on nozzle discharge coefficients, Reynolds numbers, and the resistance of perforated plates is also presented. Such apparatus in some laboratories is commonly referred to as a *code tester*.

The Venturi meter is like the nozzle except for the addition of a downstream transition section that reduces the pressure drop through the measuring apparatus.

In some cases air velocity through a duct, heater coil, or heating unit may be most conveniently estimated by computation from the heat given up by the coil and the temperature rise (measured by thermocouples) of the air passing through. It is essential to have a uniform flow over the entire inlet and outlet of the heater at the plane of temperature measurement.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wind-driven wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range and subtractive at the upper range with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm. Anemometers of this type are practically standard for wind measurements and may be used in large ducts where the air flow is not seriously altered by the presence of the instrument itself.

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case, against which air exerts a pressure as it passes through the instrument from an up-stream to a down-stream opening. The movement of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. With fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful for studying motion of the air in a room[25] and in locating objectionable drafts. Various attachments are available, such as the double tube arrangement for determining velocities in ducts, and a device for measuring static pressures. Each instrument and the attachments for it must receive individual calibration. For determining average velocity in a duct it is necessary to *traverse* the duct as is done when using the Pitot tube.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often desirable to make volume measurements at the face of the supply openings. It is rare to have access to the interior of duct sections where the flow is sufficiently uniform for measurement. For accuracy the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field.

Tests have shown that the propeller type anemometer can be used successfully on most of the common types of supply grilles[26,27]. The core area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To obtain the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille (core) in square feet.

On exhaust openings, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross core area of the grille in square feet and by a coefficient for average conditions of 0.85[28].

When a propeller type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method and by air volume measurements in a duct approach to the inlet[29].

Anemometers can be used within their range at the face of supply grilles when properly applied. In principle it is a case of finding the velocity at many points and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles[30]. On modern air conditioning grilles the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance the constriction due to the thin bars has disappeared since the small air jets have reunited, and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. With suitable traversing tips and calibration, this instrument may also be used on exhaust grilles if proper grille factors are applied[31]. Those contemplating such measurements should consult the references cited.

Smoke is a qualitative tool which is very useful in studying air movements. Satisfactory smoke can be obtained from titanium tetra-chloride (which, however, is very irritating to nasal membranes) or by mixing potassium chlorate and powdered sugar (a non-irritating smoke) and firing the mixture with a match. This latter process evolves considerable heat and it should be confined in a pan away from flammable materials. The titanium tetra-chloride smoke lends itself to spot determinations, particularly for leakage through casings and ducts, as it can be easily handled in a small pistol-like ejector. The fumes of aqua ammonia and of sulfuric acid, if permitted to mix, form a white precipitate which is useful for some purposes. Two bottles, one containing ammonia water and the other acid, are connected to a common nozzle by means of rubber tubing. Air is forced over the surfaces of the liquids in the bottles by means of a syringe and the two streams, upon mixing at the nozzle, form a white cloud.

The hair hygrometer consists of from one to several strands of hair with a mechanism whereby changes in length of the strands, due to changes in humidity, cause an indicator to move across a dial. In the recording instrument, a pen is moved across and marks a moving paper ribbon, indexed in relative humidity. In a controller, or humidistat, the motion makes or breaks an electric contact governing the air conditioning equipment. Such devices require initial calibration and, for precise work, frequent recalibration or setting especially if they are exposed to extremes of either high or low humidity. For continuous operation with only slight changes in humidity some operators report satisfactory reproducibility of results.

Electrolytic Hygrometers

The dampness and therefore the electrical resistance of a salt film varies with the humidity of the atmosphere to which the film is exposed and at least two types of instruments based on this fact have been developed. The Dunmore hygrometer was originally designed for use in radio-sondes or small balloons and means were devised whereby the device transmits humidity data back to earth in the form of a radio signal. In the more usual form this hygrometer consists of a dual winding of small wire on a non-conducting tube. The whole is coated with an electrolytic film, usually containing a salt such as lithium chloride, which forms an electric connection between the windings. Means are provided for determining the electrical resistance of the film which is an indication of the humidity. In the radio-sonde, variations in the resistance of the film affect the frequency of an oscillating circuit. These hygrometers are usually calibrated by comparison with a wet and dry bulb psychrometer. For calibration for some purposes, particularly for use at sub zero temperatures, means have been provided for producing atmospheres of known humidity[41]. Advantages of instruments of this type for some scientific and industrial purposes are becoming apparent and some forms of it are on the market.

The Weaver type hygrometer is particularly useful for determining the humidity of air or other gas in pipes or closed vessels at various pressures and temperatures. It consists of a threaded plug carrying a central electrode insulated from the plug except for a gelatinous electrolytic film. The film is exposed to the air or gas from the pipe or vessel and provision is made for determining its electrical resistance. The hygrometer, or detector, is mounted in a manifold equipped with valves whereby the film can be alternately exposed to the test gas and to a standard gas having a known absolute humidity. The pressure of the standard gas is varied until contact with it establishes the same resistance in the film as the test gas and the humidity of the test gas is then determined by computation based on the gas laws.

Chemical Hygrometry

The humidity of an atmosphere can be measured directly by extracting and weighing the water vapor from a known sample. For precise laboratory work, powerful desiccants such as sulphuric acid and phosphorus pentoxide are used for the extraction process while for some purposes, calcium chloride, lithium chloride or silica gel are satisfactory. Freezing the water vapor out of a measured stream of air or gas with solid carbon dioxide and weighing the resulting ice is a similar operation. A thermal conductivity method for gas analysis can be used for temperatures above 212 F or for very low humidities[42].

DUST DETERMINATION AND AIR CLEANER TESTING

Two absolute measures of air dustiness are in use; particles per unit volume of air and weight per unit volume of air. A comparative method is also used consisting in drawing known samples of air through a known area of filter cloth or paper and comparing the density of the resulting spots with blackness charts or with spots from other sources.

For counting, particles are captured in a device such as the Smith-Greenburg impinger, the Owens jet dust counter or in an electrostatic or a thermal precipitation device designed for the purpose[43]. Counting is done with a microscope and the method yields important results when the nature or constituents of the dust are of interest.

For a weight determination, a known volume of air is drawn through a porous crucible or thimble and the weight gained by the thimble during the operation is the weight of the dust captured from the air sample[44]. For precise work, the thimble must be dried in a desiccating chamber before weighing each time. The test method specified in the A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilating Work is based on a weight method for evaluating the cleanliness of air after passing through an air cleaner[45].

The Code has not filled all needs for an air cleaner testing method and the subject is now under investigation by the A.S.H.V.E. Research Laboratory. At the *National Bureau of Standards* a test method was developed for interested Government agencies under which measured samples of the uncleaned air and of the air cleaned by a device under test are passed through filter papers[46]. The ratio of the flow rates through the two filter papers are adjusted during successive tests until the resulting dust spots approach equality in density as shown by a photometer. The ratio of the flow rates is then indicative of the effectiveness of the cleaner in arresting dust.

Other instruments for evaluating air dustiness are described by Drinker and Hatch[43].

A statement of an air cleaner's efficiency by any test method, is meaningless unless the test dust is specified.

HEAT TRANSFER THROUGH BUILDING MATERIALS

Thermal Conductivity

Use of the guarded hot plate apparatus for determining the thermal conductivity (k value) of homogeneous materials was adopted by A.S.H.V.E. in 1942, and has become practically universal and the apparatus is described in an A.S.T.M. publication[47,48]. It consists essentially of an electrically heated plate and two water cooled plates. Two identical specimens or slabs of a material are required for a test and one is mounted on each side of the hot plate. A cold plate is then pressed against the outside of each specimen by a clamp screw. Hot plate apparatus accommodating specimens on the order of one foot square and an inch or more thick is common. The apparatus at the *National Bureau of Standards* takes specimens 8 in. square while plates as large as 3 ft square have been used. The heated plate is divided into two portions: the central or measuring section and the outer or guard section. During tests the two sections are maintained as nearly as possible at the same temperature and the purpose of the guard section is to minimize errors due to edge effects. The electric energy required to heat the measuring section is carefully observed and, converted to Btu per hour, is divided by the area and the temperature gradient to obtain the conductivity of a material tested.

Wall Conductances

The thermal conductances (C values) of many walls can be satisfactorily estimated from the conductivities of their components and their dimensions but some walls are complicated, by the inclusion of metal, for instance, and tests for conductance are required. The apparatus is required to accommodate large specimens representing actual construction. The shielded hot box apparatus was developed for this purpose[49]. Specimens for the apparatus at the National Bureau of Standards are 5 ft long and 8 ft high while others require different sizes, some larger, others smaller.

The guarded hot box is described in the A.S.H.V.E. Standard Test Code for Heat Transmission through Walls[49]. The apparatus consists essentially of three boxes; a cold box, cooled by a refrigerating machine, a hot box, heated electrically, and a metering box also heated electrically. Each box has an open side to be placed against the specimen. The cold box is clamped against one side of the specimen and the hot box against the other. The hot box encloses the metering box and is kept at the same temperature to minimize heat exchanges to or from the metering box except through the specimen. The electric energy necessary to heat the metering box is measured, converted to Btu per hour, and divided by the area and the temperature difference through the wall, from surface to surface, to yield the conductance of the wall. The transmittance or U-value of the wall is then computed by means of the surface coefficients from Chapter 6.

The Nicholls heat flow meter is sometimes useful for measuring steady heat flow through a wall or other building member[50]. In essence, this meter consists of a plate or slab of material of known thermal resistance having attached thermocouples on both sides. For use, the device is pressed against or cemented to the wall to be tested. At steady state, the temperature difference through the slab, measured with the thermocouples, with the known thermal resistance of the slab, indicates the heat flow through the slab and hence through the wall covered by it. For best results, such meters are calibrated by means of a guarded hot plate or other suitable apparatus. The chief precaution is to assure that the heat flow is steady at the time of measurement.

EVALUATION OF THE THERMAL ENVIRONMENT

Advocates of radiant heating emphasize the fact that comfort depends on radiant heat exchanges as well as air temperature. For this reason several instruments have been devised to evaluate the comfort or warmth of rooms, taking radiant as well as convective effects into account. Prominent among these are the eupatheoscope, the globe thermometer, the thermal integrator and the heated globe[51,52]. Descriptions are contained in the references and are omitted here because these devices are not widely used in America for several reasons, among which is the fact that radiant heating with high temperature sources is not a chief method of comfort heating in this country.

EQUIPMENT TESTING

There are two approaches to the problem of measuring the capacities of fuel burning devices such as boilers and furnaces. The direct or calorimetric test consists in measuring the change in enthalpy or heat content of the fluid, air or water, heated by the device and multiplying by the flow rate in pounds per hour to arrive at the capacity in Btu per hour[53]. The indirect test consists in determining the heat lost in the flue gases and deducting it from the heat evolved by combustion of the fuel[54]. A heat

TABLE 1. RINGELMANN SMOKE CHART SPACINGS

NUMBER OF CARD	THICKNESS OF LINES, MM	DISTANCE IN CLEAR BETWEEN LINES, MM
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

balance consists in the simultaneous application of both tests to the same device. The indirect test almost invariably indicates the greater capacity and the difference is credited to radiation from the boiler or furnace casing and *unaccounted for loss.*

In the case of some small equipment, the expense of the direct test is not considered justifiable and the indirect test is relied upon with an arbitrary *radiation and unaccounted for* factor[54].

Flue Gas Analysis

The Orsat apparatus is commonly used for analyzing flue gases. In its ordinary form it consists of three pipettes and a means for isolating a sample of flue gas in a graduate. After measuring, the sample is expelled from the graduate into the first pipette where the carbon dioxide is extracted by potassium hydroxide. The sample is then remeasured and successively passed into the second and third pipettes where the oxygen and the carbon monoxide are respectively extracted by potassium pyrogallate and cuprous chloride.

For field testing and burner adjustment, simpler portable devices are available which analyze only for carbon dioxide and curves are in use for several common fuels, based on typical hydrogen contents, by means of which efficiencies can be estimated. More elaborate laboratory equipment is sometimes provided for precise determination of carbon monoxide content by burning the carbon monoxide to carbon dioxide by means of a catalyst[55,56]. In large plants, carbon dioxide recorders are used to obtain a continuous indication of the plant's efficiency[57].

SMOKE DENSITY MEASUREMENTS

Ringelmann charts are widely used for evaluating the density of smoke discharged from chimneys or stacks, and smoke ordinances are based on them in some cities. Each chart is composed of a series of crossed black lines on white paper which, at a distance of about 50 ft, is visually compared with the smoke under observation. Four charts are used with different degrees of blackness as shown in Table 1. The smoke density is specified by Ringelmann numbers, from 1 to 4.

The photo-electric cell is used in some apparatus developed for smoke density recording in large plants. The same device is included in the testing equipment for domestic oil burners described in *National Bureau of Standards*, Commercial Standard CS75-42[58]. Under *Laboratory Tests* this publication contains the following section: "Smoke Determination.—After combustion has reached equilibrium, the amount of smoke in the flue gases, when viewed lengthwise through 4 feet of the smoke pipe in accordance with the *Underwriters' Laboratories, Inc.*, Standard for Domestic Oil Burners (Subject 296), March 1934 and subsequent revisions, shall not reduce the output of a standard photoelectric cell from 9 microamperes, with a clear

smoke pipe, to less than 8 microamperes." The commercial standard also requires that during a test after installation, the burner shall operate without visible smoke at the chimney top.

A method of evaluating smoke produced by pot type oil burners was developed for the *Institute of Cooking and Heating Appliance Manufacturers* by R. N. St. John. A glass rod is interposed between a light source and a photo-sensitive cell both before and after being exposed to the flue gases from a heating device. The diminution of the light transmitted by the rod due to the deposit of soot on its surface causes a reduction in the cell emf which is taken as an index of the concentration of smoke in the flue gases. A description of the method is contained in National Bureau of Standards Commercial Standard CS104-46[54].

Sound and Vibration Measurements

Approximate measurements of sound intensity can be made by aural methods. The ear is used to compare the measured noise with sounds of known strength.

Electrical devices in which the ear plays no part furnish the most satisfactory means of measuring noise intensities. The sound meter consists essentially of a microphone coupled to an amplifier designed with an ear-like response. The output of the microphone is read on a sensitive direct current milliammeter graduated to read directly in decibels. Instruments of this type if connected with suitable band pass filters can be used to study the intensity of the sound over its entire range of frequencies.

Electrical instruments are available for measuring the frequency, amplitude and acceleration of a vibrating mass. They are usually more convenient and accurate than the vibrating reed tachometer and the seismic type displacement meters and accelerometers which can also be used for this purpose. Sound level meters are discussed in several text books[59,60] and standards[61].

REFERENCES

[1] Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165), adapted January, 1930, by A.S.H.V.E.

[2] A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25), adopted June, 1932.

[3] A.S.R.E. Standard Methods of Rating and Testing Air Conditioning Equipment, A.S.R.E. Circular No. 13.

[4] A Survey of Testing Methods and Rating Limits for Domestic Heating Devices, by R. S. Dill (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 185).

[5] For a comprehensive treatment of temperature measurement the reader is referred to Temperature, Its Measurement and Control in Science and Industry, a symposium sponsored by the *American Institute of Physics* and published by Reinhold Publishing Corp.

[6] Errors in the Measurement of the Temperature of Flue Gases, by P. Nicholls and W. E. Rice (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 473).

[7] Temperature Measurement (*A.S.M.E.* Power Test Code, Part 3).

[8] Investigation of Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz and V. S. Day (*Illinois Engineering Experiment Station*, *Bulletin* No. 120).

[9] Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 94).

[10] Parallel-Connected Thermocouples for the Testing of Gas Appliances, by Walter B. Kerk and George J. Pacanovsky (*Gas*, September, 1939, p. 51).

[11] American Standard for Indicating Pressure and Vacuum Gages, Round Dial Type, with Elastic Pressure Chamber, B. 40.1-1939 (*American Standards Association*).

[12] Standard Test Code for Centrifugal and Axial Fans, Edition of 1938. See also Standard Code for the Testing of Centrifugal and Disc Fans (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407; Vol. 37, 1931, p. 363).

[13] Fan Engineering, Buffalo Forge Company, 5th Edition, p. 165.

[14] Illinois Micromanometer (*University of Illinois, Engineering Experiment Station Bulletin* No. 120, p. 91).

[15] The Weathertightness of Rolled Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 527).

[16] Pressure Measurement (*A.S.M.E.* Power Test Code 1936, Part 2, Chapter 2).

[17] The Measurement of Static Pressure, by C. J. Fechheimer, *Mechanical Engineering*, August, 1927.

[18] Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point (*U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.).

[19] For technical data refer to Fluid Meter Reports, Parts 1—1937, 2—1931, and 3—1933 (*American Society of Mechanical Engineers*).

[20] Technical Notes No. 546 (*National Advisory Committee for Aeronautics*, November, 1935).

[21] The Characteristics of Double Pitot Tubes, by F. R. Ingram, E. Diez-Canseco and L. Silverman (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1942, p. 708).

[22] Discharge Coefficients of Square Edged Orifices for Measuring the Flow of Air, by H. S. Bean, E. Buckingham and P. S. Murphy (*Bureau of Standards Journal of Research*, Vol. 2, 1929, p. 561).

[23] Flow Measurement by Nozzles and Orifice Plates (*A.S.M.E.* Power Test Codes, Chapter 4 of Part 5, 1940).

[24] Air Flow Measurement in the Laboratory, by D. D. Wile, *Refrigerating Engineering*, June, 1947, p. 515.

[25] A.S.H.V.E. RESEARCH REPORT No. 1204—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 241).

[26] A.S.H.V.E. RESEARCH REPORTS Nos. 857, 911 and 966—Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 201, Vol. 37, 1931, p. 619, and Vol. 39, 1933, p. 373).

[27] A.S.H.V.E. RESEARCH REPORT No. 1162—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 313).

[28] A.S.H.V.E. RESEARCH REPORT No. 1092—The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr. and M. H. Dean (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 387).

[29] A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463).

[30] A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution From Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

[31] Measuring Air Flow, by G. L. Tuve (*Heating, Piping and Air Conditioning*, December, 1941).

[32] A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645).

[33] Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 103).

[34] A.S.H.V.E. RESEARCH REPORT No. 1165—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 351).

[35] Development of Testing Apparatus for Thermostats, by D. D. Wile (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 349).

[36] The Heated Thermometer Anemometer, by C. P. Yaglou (*Journal Industrial Hygiene and Toxicology*, Vol. 20, October, 1938, No. 8).

[37] A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[38] The Temperature of Evaporation, by Willis H. Carrier (A.S.H.V.E. Transactions, Vol. 24, 1918, p. 25).

[39] Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point (*U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.).

[40] A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems, by W. H. Carrier and C. O. Mackey (*A.S.M.E. Transactions*, January, 1937, p. 33; Discussion *A.S.M.E. Transactions*, August, 1937, p. 528).

[41] Divided Flow Low Temperature Humidity Test Apparatus, by Arnold Wexler (*National Bureau of Standards*, Research Paper No. 1894).

[42] Gas Analysis by Measurement of Thermal Conductivity, by H. A. Daynes (*Cambridge Press*, 1933).

[43] Industrial Dust, by Phillip Drinker and Theodore Hatch (McGraw Hill Book Company, New York).

[44] Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by S. R. Lewis (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 277).

[45] A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, adopted January, 1934 (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 225).

[46] A Test Method for Air Filters, by Richard S. Dill (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 379).

[47] Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate, adopted July, 1942, by A.S.H.V.E. (*A.S.T.M. Designation* C177-42T).

[48] Guarded Hot Plate Apparatus Complying With the Requirements of Section 4 of *A.S.T.M.* Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate, *A.S.T.M.*

[49] A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls (A.S.H.V.E. Transactions, Vol. 34, 1928, p. 253), adopted 1928.

[50] A.S.H.V.E. Research Report No. 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. Transactions, Vol. 30, 1924, p. 65).

[51] Instruments and Methods for Recording Thermal Factors Affecting Human Comfort, by C. P. Yaglou, A. P. Kratz and C.-E. A. Winslow (Year Book, *American Journal Public Health*, 36–37).

[52] The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 149).

[53] I = B = R Testing and Rating Codes for Low Pressure Heating Boilers, 1947 (*Institute of Boiler and Radiator Manufacturers*).

[54] Commercial Standard for Warm Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners, C.S. 104-46, *National Bureau of Standards*.

[55] Rapid Determination of Small Amounts of Carbon Monoxide, by Martin Shepherd. *Ind. Eng. Chem. Anal.* Ed. *19*, 77: 1947.

[56] Determination of Small Amounts of Carbon Monoxide in Air by Various Reference Methods, by Martin Shepherd, *National Bureau of Standards* Journal of Research *38*, 351–8; 1947 R. P. 1777.

[57] A Carbon Monoxide Recorder, by S. H. Katz, D. A. Reynolds, H. W. Frevert and J. J. Bloomfield (*U. S. Bureau of Mines, Technical Paper* No. 355, 1926).

[58] Commercial Standard for Mechanical Draft Oil Burners Designed for Domestic Installations, C.S. 75-42, *National Bureau of Standards*.

[59] Electrical Engineer's Handbook, by Harold Pender and Knox McIlvaine (John Wiley and Sons, New York).

[60] Elements of Acoustical Engineering, by Harry F. Olson (D. Van Nostrand Co., New York).

[61] American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, Z 24.3-1944 (American Standards Association).

CHAPTER 12
PHYSIOLOGICAL PRINCIPLES

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Interchanges Between the Body and Its Environment, High Temperature Hazards, Acclimatization, Upper Limits of Heat for Men at Work, Application of Physiologic Principles to Air Conditioning Problems, Effective Temperature Index and Comfort Zones

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VENTILATION is defined in part as *the process of supplying air to, or removing air from, any space by natural or mechanical means.* The word in itself implies quantity, but air must be of the proper quality also. The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. The A.S.H.V.E. Code of Minimum Requirements for Comfort Air Conditioning[1] defines it "as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed."

CHEMICAL VITIATION OF AIR

People living indoors bring about certain physical and chemical changes in the air about them. The oxygen content of the air diminishes and the carbon dioxide increases, but these changes are too slight to be significant except in air tight spaces as in submarines. Organic matter which is usually perceived as odors comes from the body or clothes. Moisture and heat are given off by the body. There is no evidence of any toxic volatile material given off by man to the ambient air. Stale air may be offensive because of odors and may induce loss of appetite and loss of energy. Objectionable body odors have the same effects. These reasons, whether esthetic or physiological, usually make it desirable in the design of air conditioning systems to provide for the elimination or control of odors arising from occupancy, cooking, or other sources. This may be accomplished by introducing odor-free air in sufficient quantities to reduce odor concentrations by dilution to a level which is not objectionable. Odor-free air may be outdoor air or air which has been cleared of odors by sorption, washing, or other appropriate means.

In the case of vitiation by a few hazardous gases such as carbon monoxide from heating, cooking, and certain industrial processes, no satisfactory chemical treatment for the elimination of the impurity has been found. The only satisfactory solution is elimination at the source by local exhaust ventilation; or, if this is impossible, reduction to a safe concentration by dilution. (See Chapter 10.) In the case of contamination by other matter, including volatile vapors and gases, chemical treatment for the removal or reduction of the impurities has been made available through air cleaning methods, which are discussed in Chapter 33.

When the only source of contamination is the human occupant, the minimum quantity of outdoor air needed appears to be that required to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors,

including the dietary and hygienic habits of the occupants (frequently reflecting their socio-economic status), the outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, and temperature and relative humidity. Perception of odor has been found to vary as the logarithmic function of the odor intensity, or inversely with the logarithmic function of the amount of outdoor air supplied and the air space per person.

The relation between air supply and occupancy has been reported by the Harvard School of Public Health[2] (Table 1) and the A.S.H.V.E. Research Laboratory[3]. Outdoor air requirements for removal of objectionable tobacco smoke odors are not accurately known but available information and current practice indicate the need of 15 cfm per person or more[4].

The total quantity of outside air to be circulated through an enclosure is often governed chiefly by physical considerations for controlling temperature, air distribution, and air velocity. Other factors which must be taken into consideration, include the type and usage of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and the operation of the system distributing the air supply. Frequently, some of these factors, particularly the need for air movement and good distribution, may be satisfied by recirculation of inside air rather than outside air.

It will be noted that, with adequate air space, the rate of air change indicated in Table 1 is from 10 to 30 cfm per person. In rooms occupied by only a few persons such an air change will be automatically attained in cold weather by normal leakage around doors and windows, and can easily be secured in warm weather by the opening of windows. With a space allotment of 400 cu ft per person, only $1\frac{1}{2}$ air changes per hour are necessary to provide a ventilation rate of 10 cfm per person.

Therefore, in the ordinary dwelling with adequate cubic space allotment, no special provision for controlling chemical purity of the air is necessary (aside from removal of fumes from heating appliances). For such conditions, the control of air temperature is the major consideration.

In more crowded rooms (large offices, large workrooms, auditoriums), the cubic space per person is less and it is usually impossible to admit untempered outside air without creating drafts. Here, mechanical ventilation is essential for removal of the heat and moisture produced by the occupants.

The present data regarding the effect of cubic space on ventilation requirements are not universally accepted. The Code of Minimum Requirements for Comfort Air Conditioning[1] prescribes definite minimum requirements which should be familiar to the designing engineer. It should be emphasized, however, that the code fixes *minimum*, rather than *adequate* requirements.

Notwithstanding the rapid advance made in air conditioning some persons still believe there is a stimulating quality in outdoor air (particularly country, mountain and seashore air) under ideal weather conditions, which is lacking in artificially conditioned air. It is apparent, however, that modern air conditioning insures control of the phenomena of nature for the service and comfort of man independently of weather conditions. Freedom of movement, action and thought, together with the variability of stimuli experienced by persons under ideal conditions in the country, mountains or seashore, undoubtedly have some stimulating effect. Various experimenters have attempted to duplicate the invigorating qualities of

TABLE 1. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS UNDER LABORATORY CONDITIONS[2]

Type of Occupants	Air Space per Person Cu Ft	Outdoor Air Supply CFM per Person
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
	200	16
	300	12
	500	7
Laborers	200	23
Grade school children of average socio-economic status	100	29
	200	21
	300	17
	500	11
Grade school children of lower socio-economic status	200	38
Children attending private grade schools	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

outdoor air by the use of ozone, ionization, or ultra-violet light, but results to date have been inconclusive or negative[5].

Ozone in amounts of 0.01 to 0.05 ppm of air is allowable in comfort air conditioning. Above this limit there is a pungent, unpleasant odor and perhaps respiratory distress, depression, and stupor[6].

PHYSICAL IMPURITIES IN AIR

Dust particles of almost any type can produce irritation of the mucous membranes of the nose and throat if present in high concentrations. Certain dusts may be very harmful, but coal dust is tolerated well. The effects of various industrial dusts, pollens, etc. are discussed in Chapter 10.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic organisms from infected persons. (See Chapter 13.)

While in some instances it may be possible to reduce the physical impurities of the air by dilution from a non-contaminated source, such a source is rarely available. Frequently outside air contains a higher concentration of physical impurities than indoor air. Therefore, it is usually desirable to reduce the concentration of physical impurities by air cleaning methods. (See Chapter 33.)

THERMAL INTERCHANGES WITH ENVIRONMENT

Body temperature depends upon the balance between heat production and heat loss. Heat resulting from oxidation in the body (metabolism)

maintains the body temperature well above that of the surrounding air in a cool or cold environment. At the same time, heat is constantly lost from the body by radiation, convection and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. During work the body temperature may rise; in fact, afternoon temperatures of normal persons average 1 deg above the resting value of the morning whether working or not.

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M = \pm S + E \pm R \pm C \tag{1}$$

where

M = rate of metabolism, heat produced within the body.
S = rate of storage, change in intrinsic body heat.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

The rate of metabolism, M, is always positive. The storage, S, may be either positive or negative, depending upon whether heat is being stored or depleted owing to a rise or fall in body temperature. Under ordinary circumstances (when the dew-point of the air is below the body surface temperature) the evaporation loss, E, is always positive; that is, heat from metabolism supplies this loss. R and C are positive when body surface temperature is above that of walls and air, and negative when it is below.

DuBois[7], after careful calorimeter studies on fasting, nude men, plotted the partition of body heat loss and heat production as a function of temperature. Fig. 1 shows some disparity between heat production and heat loss. This disparity is S in Equation 1. In the central range of the experiments, S was quite low and no increase in heat loss by vaporization was apparent. Within the range 81 to 86 F a zone of easy regulation of body heat exists for nude persons which may be called a *zone of thermal neutrality* in contrast with the chilling effect noted in the cooler part of the curve, the *zone of body cooling*, and in contrast with the *zone of evaporative regulation* at higher temperatures. In the narrow range of the neutral zone most men feel comfortable under similar conditions (nude and at rest). In this neutral zone the regulation of body temperature is accomplished by automatic variation in the blood flow to the skin, especially of the hands, feet, and head, to vary the radiation and convection losses as required to balance heat loss and heat production.

In the *zone of body cooling* circulation to the skin is greatly reduced; the feet and hands cool sharply; *gooseflesh* appears early; the rectal temperature falls progressively until shivering or work increases heat production. Adolph and Molnar[8] showed that shivering or work can increase heat production four- or five-fold and so constitute protection from exposure to cold. They showed that the cutaneous vaso-constriction during chilling produces a temperature gradient averaging 4.4 cm in thickness below the skin in chilled nude men. Their subjects complained of pain, especially in the feet, which progressed to numbness. Discomfort from cold was followed by mental confusion, dullness and stupor. Fatigue limited their shivering response and shivering stopped with sleep. Hypo-

thermia (abnormally low body temperature) was then inevitable. These authors observed no acclimatization to cold. There is no good evidence that man can adapt himself to cold by increasing heat production except that due to exercise or shivering. In animals and even in certain mammals, such *chemical regulation* is known to exist.

In the zone of evaporative regulation the burden of heat dissipation is assumed by evaporation, radiation and convection losses already being maximal from the warm skin, flooded by maximal cutaneous blood flow[9]. In such hot conditions the pulse rate rises. When sweat first appears it covers only part of the body, especially the head and neck, and gradually extends to drench the entire body surface. All factors which affect the evaporation of water from the skin affect heat regulation in the zone of

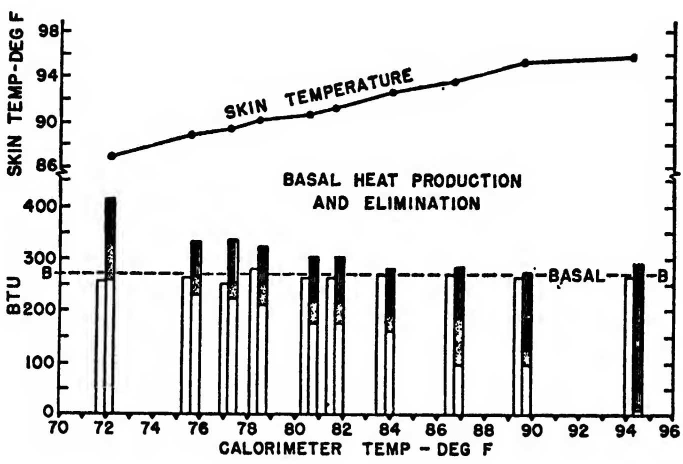


FIG. 1. HEAT LOSS FROM HUMAN BEINGS BY EVAPORATION, RADIATION, AND CONVECTION[a]

[a] Normal control, naked, in calorimeter at temperatures from 72.8 to 94.1 F. First column in each experiment represents heat production as determined by indirect calorimetry, the second column, heat elimination. The portion marked with vertical lines represents vaporization, the dotted area convection, the unmarked area radiation. The skin temperature represents the average reading of 18 spots on the surface.

evaporative regulation. Relative humidity and air motion are most important. With dry-bulb temperature above body temperature, air motion facilitates evaporative heat loss by removing hot humid air from contact with the skin and replacing it with relatively drier air.

Heat regulation in man requires an intact set of sensory nerves, a normal sympathetic nerve supply to sweat glands and blood vessels, a great many sweat glands, and a circulatory system capable of carrying heat from muscles and viscera to the skin by circulation of the blood.

Some of the phenomena of body temperature control are shown graphically in Fig. 2. The dotted curves, from a study at the John B. Pierce Laboratory of Hygiene[10], are for subjects lightly clothed in a semi-reclining position and give the relation between the dry-bulb temperature of the environment (with about 45 per cent relative humidity) and the metabolic rate (heat production), the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation from the skin and the respiratory tract. The smooth line curves from the work of the A.S.H.V.E. Research Laboratory[11] give the same relationships for

healthy, male subjects (18 to 24 years of age), seated at rest and dressed in customary winter indoor clothing. The Pierce Laboratory data for the semi-reclining subjects also include the rate of heat storage (either positive or negative) due to a rise or fall in body temperature. For the normally clothed subjects, a curve gives the total heat loss (that is, the sum of the radiation, convection and evaporative losses). Here, storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic rates for the two groups of subjects may be accounted for by difference in activity. Heat exchange between the body and the environment by radiation and convection is greater for the lightly clothed subject, both for cool conditions where there is excessive heat loss, and for very warm conditions where there is transfer of heat from the atmosphere to the body. The two curves for

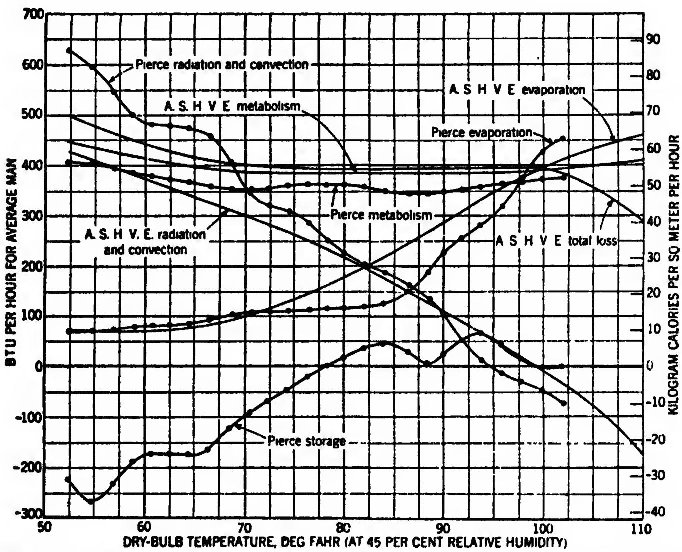


FIG. 2. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND TEMPERATURE FOR THE CLOTHED SUBJECT

evaporative loss serve to show how physiological control uses evaporation of perspiration to maintain equilibrium at high temperatures. Below 75 F for the normally clothed subject, and below 85 F for the lightly clothed subject, evaporation loss is minimal and constant. Burch[12] has shown that this insensible perspiration reflects the permeability of the skin to the moisture of the body. Above these temperatures control is obtained by the availability of perspiration for evaporation. The difference in the curves above 75 F is probably largely determined by the difference in clothing and activity.

In the zone of evaporative regulation air movement facilitates evaporation until the dry-bulb is above body temperature and convection and radiation heat gains interfere with heat regulation. Nelson et al[13] report much help from wind in enduring wet-bulb temperature of 88 to 91 F. The initial manifestation of failure of heat regulation is a rise in body temperature. Work is rarely possible with rectal temperatures over 102 F. Survival time is limited when the accelerated metabolic rate raises body temperature to 105 F.

HIGH TEMPERATURE HAZARDS

Studies at the A.S.H.V.E. Research Laboratory[14] and elsewhere during the past two decades have made available much information dealing with the physiological effects of hot atmospheres on workers and means of alleviating the distress and hazards associated therewith. Table 2 gives some of the physiological responses of men, at rest and at work, to hot environments. Frequent and continued exposure of workers to hot environments results in physiological derangement affecting the leucocyte count of the blood, and other factors dealing with man's mechanism of defense against infection.

Wherever S (Equation 1) becomes strongly positive and body temperature rises progressively men will continue to work until body temperature reaches 101 to 103 F. When these body temperatures are exceeded men work with declining efficiency and are liable to heat exhaustion, heat cramps, or heat stroke.

TABLE 2. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Fahr Deg)	Men at Rest			Men at Work 90,000 Ft-Lb of Work per Hour			
		Rise in Rectal Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	......	......	......	....	225,000	0.0	6	0.5
70	......	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0[b]	103[b]	2.7[b]
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a] Data by A.S.H.V.E. Research Laboratory.
[b] Computed value from exposures lasting less than one hour.

Heat exhaustion is a circulatory failure in which the venous return to the heart is reduced so that fainting results[15]. Early symptoms of heat exhaustion may include fatigue, headache, dizziness when erect, loss of appetite, nausea, abdominal distress, vomiting, shortness of breath, flush ing of face and neck, pulse rate above 150, fever well above 102 F, glazed eyes, and mental disturbances as apathy, poor judgment, and irritability which usually precede fainting (syncope). Recovery is usually prompt when the man is removed to a cool place and kept lying down for a time, unless he has some other illness such as heart disease.

Heat cramps are painful muscle spasms in extremities, back and abdomen due, at least in part, to excessive loss of salt in sweating. Formerly common in hot industries, this manifestation of illness due to heat is now greatly reduced by drinking water containing 0.1 per cent salt, or by proper use of salt tablets. Heat cramps are readily alleviated by administration of salt solution intravenously.

Heat stroke or sun stroke is a serious effect of exposure to great heat. The body temperature climbs rapidly to excessive levels often above 105 F

when for unknown reasons free sweating suddenly stops. At such high temperatures coma appears and death may be imminent. Emergency measures are required to reduce the excessive body temperature by cooling quickly to avoid irreparable damage to the brain[16].

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises.

ACCLIMATIZATION

When men move to deserts or to jungles some adaptation to the climate takes place. If work is gradually increased day by day and if the men can get plenty of water and salt, and can sleep each night, acclimatization may be complete in 7 to 10 days. The acclimatized man works with a lower heart rate, lower skin and rectal temperature, and more stable blood pressure than when unacclimatized. The process of acclimatization requires work in the heat[17]. During the recent war white troops lived and did hard physical work for long periods in tropical conditions when disease hazards were controlled.

In recent tests made at the A.S.H.V.E. Research Laboratory[18], subjects were required to perform light work under very hot conditions for a 4-hr period each day. It was found that the ability of a new subject to endure these conditions showed daily improvement for a period of at least 2 weeks. However, after acclimatization was completed, a recess of several days had no effect on the endurance of the subject. Individuals differ widely in their capacity to acclimatize. Acclimatized men lose these capacities in a few weeks of temperate climate, even though they are vigorously active. Acclimatization to dry heat increases the capacity to sweat and to conserve salt by secreting a dilute sweat[19].

Deterioration of performance and other ill effects of heat may arise during prolonged exposure to heat when men cannot get sufficient rest and sleep each day. Acclimatization to extreme conditions involves a strain upon the heat regulating system and interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature exceeding 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. Some persons regard the unnessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of present knowledge of air conditioning these views are not justified.

The adaptive level changes somewhat with the season. There are also marked differences between the sexes. In the cold zone the thickness of thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating is higher for women. The thickness and insulating value of the clothing worn are also important factors in the determination of the comfort level.

UPPER LIMITS OF HEAT FOR MEN AT WORK

In very hot conditions humidity is the limiting factor and the wet-bulb temperature assumes great importance. In 1905 Haldane recognized that

TABLE 3. UPPER LIMITS OF ENVIRONMENTAL CONDITIONS FOR ACCLIMATIZED, HEALTHY, YOUNG MEN IN MILITARY SERVICE

ENVIRONMENT	REACTIONS AT THE END OF 4 HR	
	Rectal Temp F	Pulse rate
Relatively Easy	Below 101	Below 130
Difficult	101 to 102	130 to 145
Impossible	Above 102	Over 145

88 F wet-bulb was the limit of endurance for coal miners and later observers have concurred.

A study was made at the Armored Medical Research Laboratory[20] to determine the upper limits of environmental conditions under which a man can perform certain work. Thirteen enlisted men, thoroughly acclimatized to the hot conditions, served as subjects. During each test, the subjects were required to march for 4 hr at the rate of 3 mph, carrying 20 lb packs under a wide range of environmental conditions which were rated as relatively easy, difficult, and impossible, on the basis of the physiological reactions of the subjects at the end of the 4-hr period as shown in Table 3 and Fig. 3.

Recognition of the need of air conditioning for workers in hot industries is growing rapidly. The choice of the type of system to be used in any given instance must be determined by the air conditioning engineer after a study of conditions. In some hot industries where few workers are engaged in large spaces the worker himself, rather than the atmosphere, can be cooled by placing him in a small booth, and blowing cooled air over him, or by circulating cooled air through a loose-fitting suit[21].

The A.S.H.V.E. Laboratory has studied the effects of walls of higher temperature than the air[18]. The findings are in part shown in Fig. 4. Mean radiant temperatures up to 40 deg above the dry-bulb did not influence physiologic processes much. For example, at 84 deg ET and 40 deg elevation in MRT, 1 deg ET change was equivalent to a 4 deg rise in MRT. Similarly, at a constant ET of 90 deg and with MRT elevations of 0 and 40 deg, 1 deg increase in ET was equivalent to 7.5 deg and 11 deg rise in MRT respectively. Under ordinary *still air* conditions the effects of air temperature and MRT appear to be interdependent. Various au-

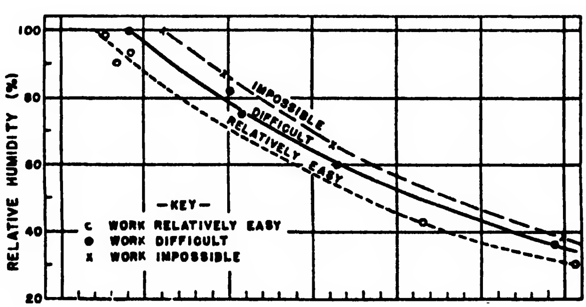


FIG. 3. HEAT ENDURANCE OF ACCLIMATIZED SUBJECTS WORKING AT A SPECIFIC RATE[20]

thorities give 0.3 to 1 deg increase of room temperature to compensate for 1 deg depression of the MRT.

APPLICATION OF PHYSIOLOGIC PRINCIPLES TO AIR CONDITIONING PROBLEMS

In order to estimate cooling loads in occupied spaces it is necessary to know the metabolic rate (heat production) of man. This has been studied extensively and found to remain relatively constant per unit of body surface area in a subject fasting and resting quietly after a good night's sleep. The rate is high in children, and diminishes gradually with age; it increases in certain diseases and in the presence of fever. The metabolic rate is somewhat lower in women. Heat production goes up sharply with work and

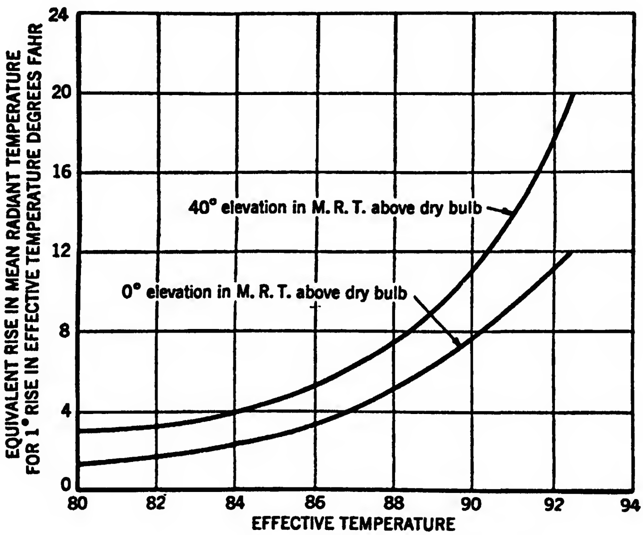


Fig. 4. Evaluation of Effect of MRT Elevation in Terms of Effective Temperature

varies widely in different persons doing the same work. Figs. 5, 6, and 7 and Table 24 of Chapter 15 give sufficient basic data for estimating heat production and heat loss under various conditions.

EFFECTIVE TEMPERATURE INDEX AND COMFORT ZONES

There is no precise physiologic observation by which comfort can be evaluated. Mean skin temperature offers some promise. The zone of thermal neutrality differs with clothing, season, activity, and all the other factors controlling heat production (Table 4). The comfort zone is very similar to the zone of thermal neutrality.

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement, and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation to cold or from warm surfaces is another important factor under certain conditions affecting the comfort reaction of the individual.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of studies[22] at the A.S.H.V.E. Research Laboratory established the equivalent conditions for practical use. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also to a considerable degree determines the physiological *effects* on the

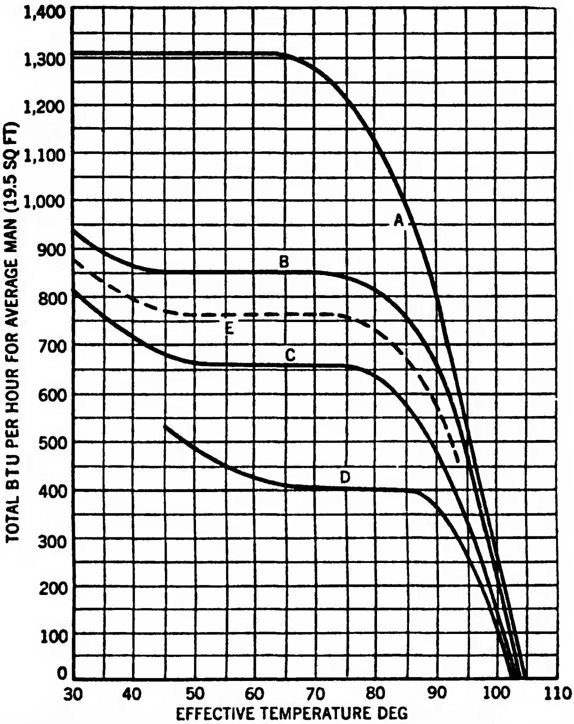


FIG. 5. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a] [14]

[a] Curve *A*—Persons working, metabolic rate 1310 Btu per hour. Curve *B*—Persons working, metabolic rate 850 Btu per hour. Curve *C*—Persons working, metabolic rate 660 Btu per hour. Curve *D*—Persons seated at rest, metabolic rate of 400 Btu per hour. Curves *B* and *D* based on test data covering a wide temperature range. Curves *A* and *C* based on test data at an Effective Temperature of 70 deg and extrapolation of Curves *B* and *D*. All curves are averages of values for high and low relative humidities; variation due to humidity is small.

body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index, and it denotes sensory heat level.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the index for any given air conditions is fixed by the temperature of slowly moving (15 to 25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg, when it induces a sensa-

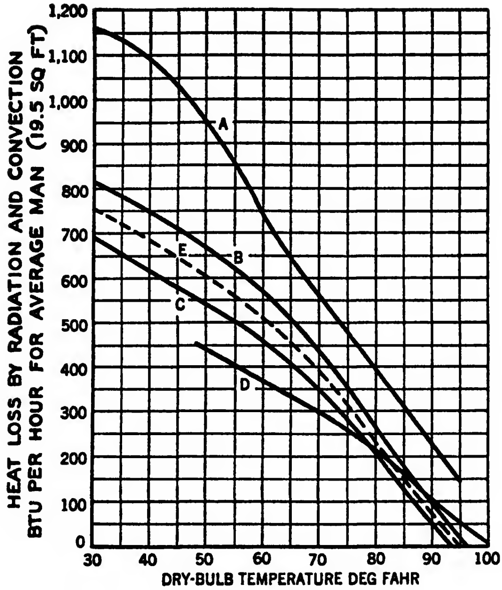


FIG. 6. RELATION BETWEEN RADIATION AND CONVECTION LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a] [14]

[a] Loc. Cit. See footnote a, Fig. 5.

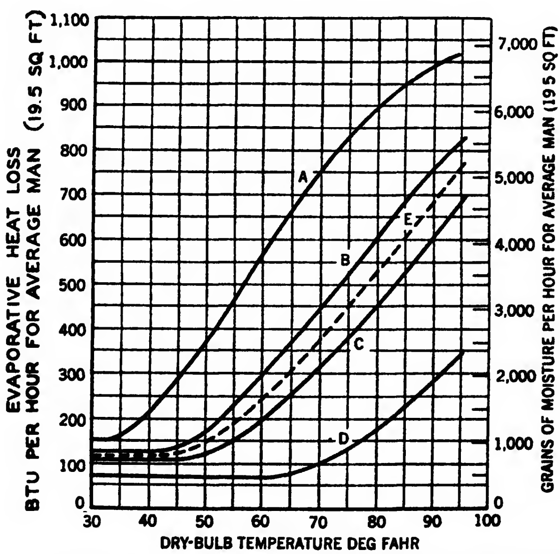


FIG. 7. EVAPORATIVE HEAT AND MOISTURE LOSS FROM THE HUMAN BODY IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a] [14]

[a] Loc. Cit. See footnote a, Fig. 5.

TABLE 4. COMPARISON OF COMFORT RANGES WITH ZONE OF THERMAL NEUTRALITY

Investigators	Effective Temperature		Operative Temp	Remarks
	Optimum Line	Range	Range	
Comfort Zone				
Houghten and Yaglou	66	63–71		Winter non-basal; at rest, normally clothed. Men and women.
Yaglou and Drinker	71	66–75		Summer non-basal; at rest and normally clothed. Men.
Yaglou	72.5	66–82		Entire year; non-basal; at rest and stripped to waist. Men.
Keeton et al.	75	74–76		Entire year; basal, nude. Steady state (9 hr exposure). Men and women.
Zone of Thermal Neutrality				
DuBois and Hardy	75	73.2–76.9		Basal; nude; men.
	71.8	64.8–76.0		Basal; clothed; men.
Winslow, Herrington and Gagge			84.0–87.8	Non-basal; at rest; nude; men.
			74 –84	Non-basal; at rest; clothed; men.

tion of warmth like that experienced in slowly moving air at 60 F saturated with moisture. The *effective temperature index* cannot be measured directly but is determined from dry- and wet-bulb temperatures and air motion observations by reference to an Effective Temperature Chart (see Figs. 8, 9, and 10) or tables.

Fig. 8 gives the effective temperature for any combination of dry- and wet-bulb temperatures for *still air* (15 to 25 fpm) conditions. Charts similar to Fig. 8 for air velocities of 300 and 500 fpm have been presented in some of the earlier editions of the GUIDE. Fig. 9 is another form of effective temperature chart embodying all three variables; dry-bulb and wet-bulb temperatures, and air velocity.

As stated previously, effective temperature is an index of the degree of warmth experienced by the body. An effective temperature line is, therefore, a line defining the various combinations of conditions which will induce like sensations of warmth. It does not necessarily follow that like sensations of comfort will also be experienced along the entire length of an effective temperature line. Some degree of discomfort is likely to be experienced at very high or very low relative humidities, regardless of the effective temperature. It has also been found that the optimum effective temperature varies with the season, and is lower in winter than in summer.

Tests[14] made at the A.S.H.V.E. Research Laboratory in very hot conditions with subjects doing light work were in very close agreement with the effective temperature chart. Other work[20] under similar environmental conditions, but with subjects walking 3 mph and carrying 20 lb packs indicated that the effective temperature lines should be more nearly horizontal. It therefore appears that the slope of the ET lines may vary depending upon the rate of work being performed.

Fig. 10, shows the A.S.H.V.E. Comfort Chart[22]. The summer and winter *comfort zones* indicate conditions under which 50 per cent or more of the people are comfortable. The summer comfort zone extends from 66 ET to 75 ET; a maximum of 98 per cent of the subjects were comfortable at 71 ET. The winter comfort zone extends from 63 ET to 71 ET with a mazimum of 97 per cent feeling comfortable at 66 ET. The 71 ET and 66 ET lines are referred to, respectively, as the summer and winter comfort lines.

The comfort zones and lines as shown in Fig. 10 are based on research

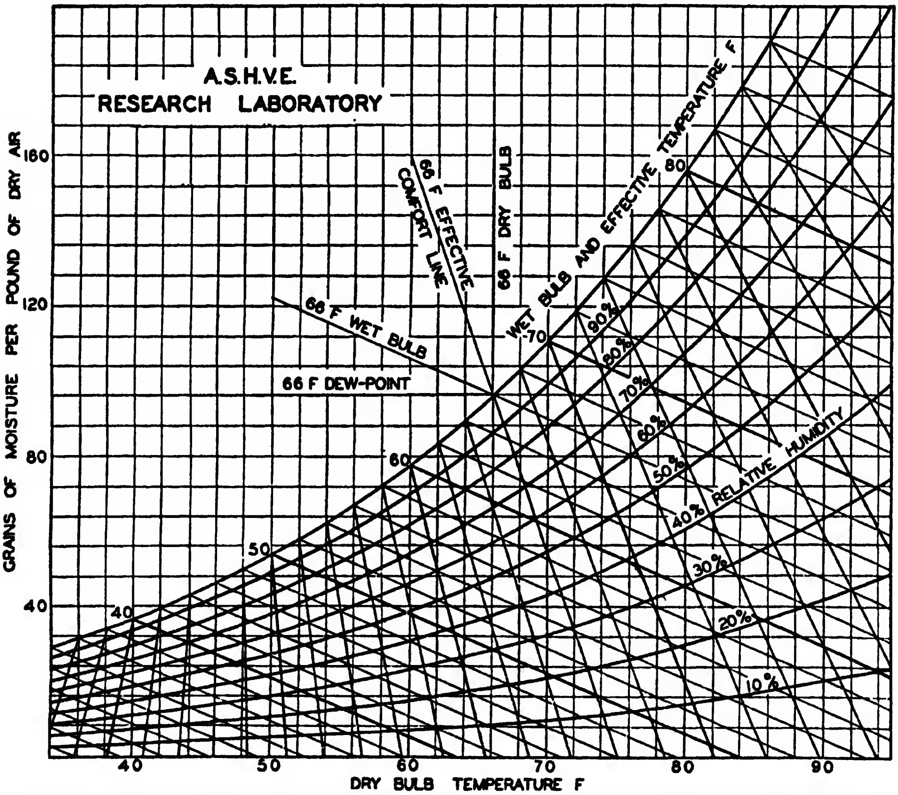


Fig. 8. Psychrometric Chart, Persons at Rest, Normally Clothed, in Still Air

prior to 1932. Later studies[23] by the A.S.H.V.E. Research Laboratory indicate a desirable winter effective temperature of 67 ET, and this finding is confirmed by current practice. The shape of the curve showing the per cent of subjects comfortable in winter also justifies this conclusion, since a drop of only one degree from the 66 ET line seriously reduced the percentage of subjects comfortable. Systems should be designed to assure comfort for the maximum possible number[24].

The comfort zones in Fig. 10 are located between the 30 per cent and 70 per cent relative humidity lines. There is some evidence that the zones could be extended somewhat beyond these limitations.

Radiation from occupants to room surfaces and between the occupants has an important bearing on the feeling of warmth and may alter to some measurable degree the optimum conditions for comfort previously in-

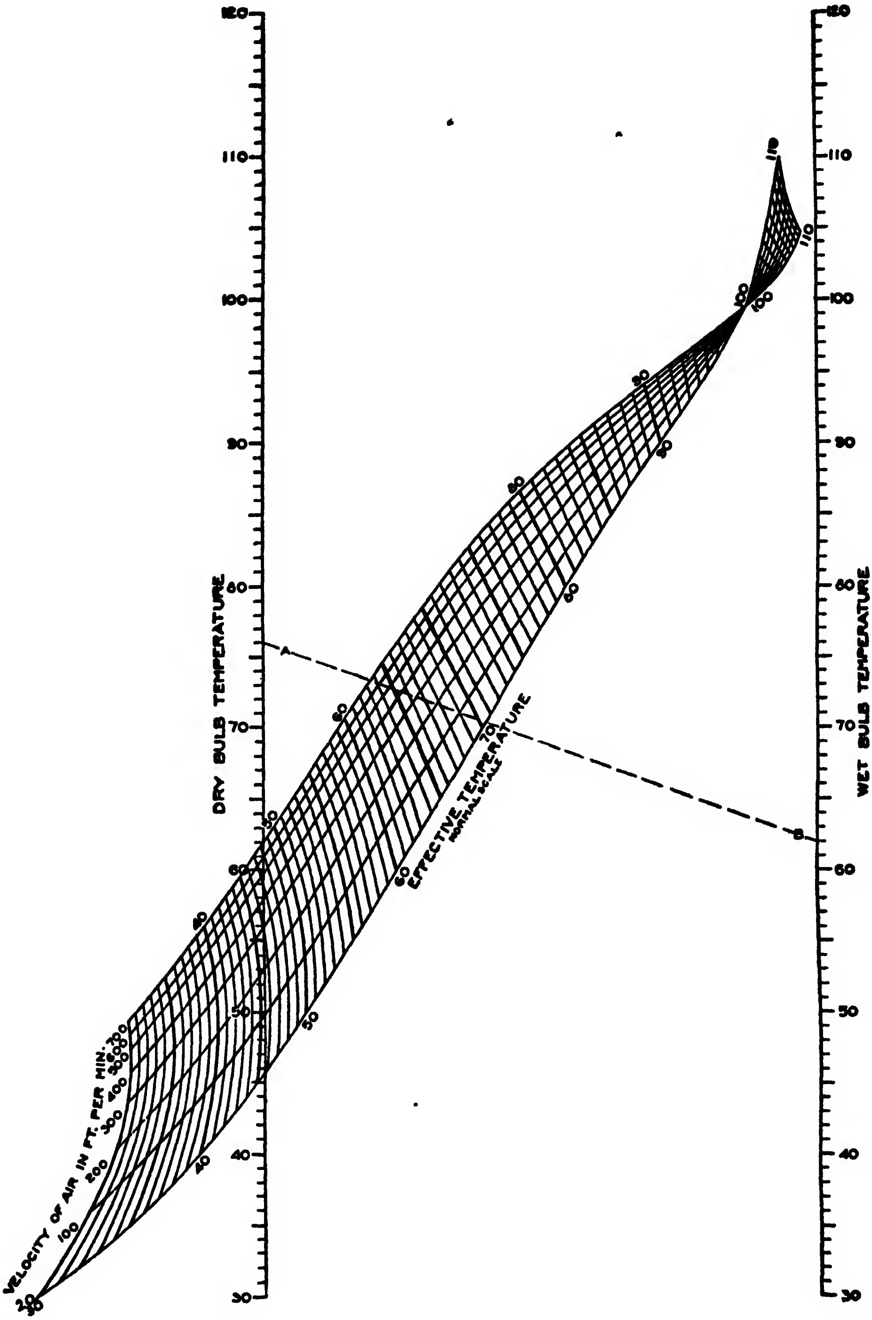


FIG. 9. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE, APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.*, warm air, direct steam or hot water radiators, plenum systems.

dicated. Since the mean radiant temperature of a space is affected by cold walls and windows, as well as by the warm surfaces of heating units placed within the room or imbedded in the walls, these factors must be compensated. Likewise, in densely occupied spaces, such as classrooms,

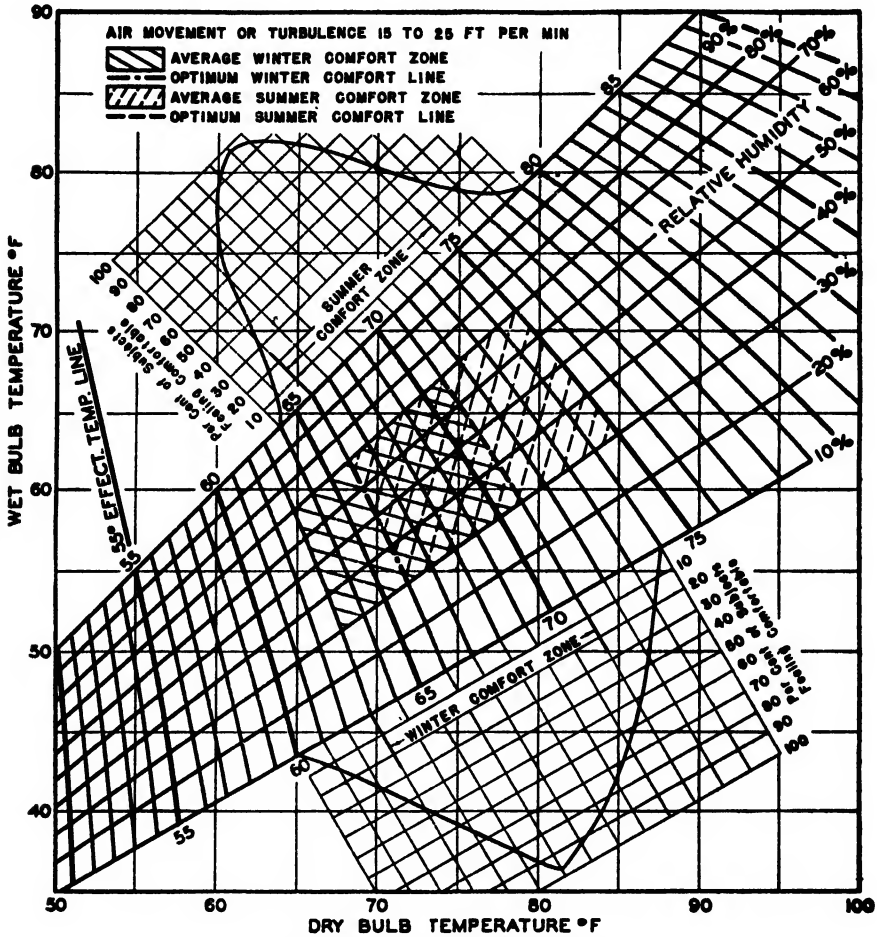


FIG. 10. A.S.H.V.E. COMFORT CHART FOR STILL AIR

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The optimum summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level. An increase of one deg ET should be made approximately per 5 deg reduction in north latitude.

theaters and auditoriums, temperatures somewhat lower than those indicated by the comfort line may be desirable because of counter-radiation between the bodies of occupants in close proximity to each other. Such radiation will also elevate the mean radiant temperature of the room.

Many field studies[23] have been made to determine the optimum indoor effective temperature for both winter and summer in several metropolitan districts of the United States and Canada in cooperation with the manage-

ments of offices employing large numbers of workers (Fig. 11). On the whole women of all age groups studied prefer an effective temperature for comfort 1.0 deg higher than men. All men and women over 40 years of age prefer a temperature 1 deg ET higher than that desired by persons below this age. The persons serving in all of these studies were representative of office workers dressed for air conditioned spaces in the summer season and engaged in the customary office activity.

On the basis of present knowledge, for different geographical regions and age groups, the most popular temperature varies from a low of 66 deg ET in winter to a high of 73 deg ET in summer. The spread for summer comfort is 69 to 73 deg ET.

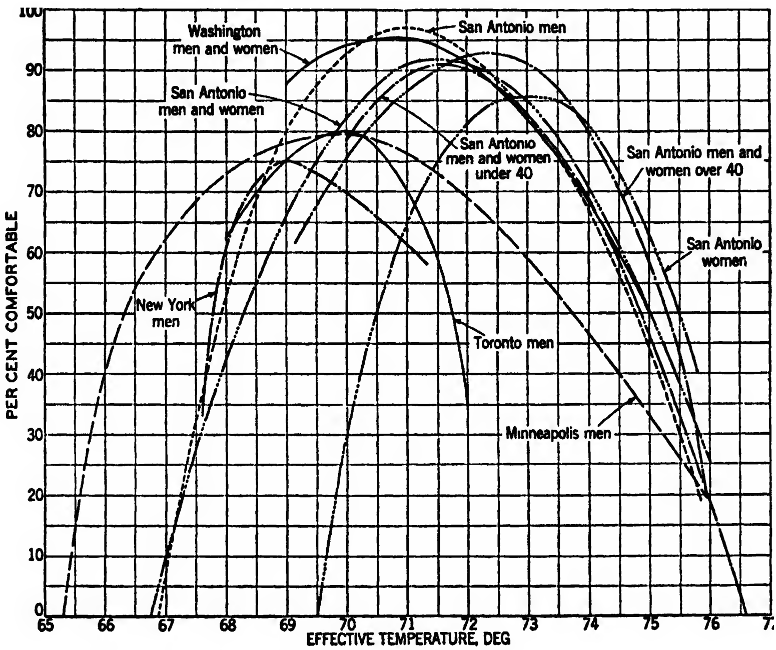


FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OBSERVATIONS INDICATING COMFORT

The A.S.H.V.E. Technical Advisory Committee on Sensations of Comfort, ascribes a spread of 3 deg in the optimum effective temperature for summer cooling to geographical location. However, it should be recognized that variations in sensation of comfort among individuals may be greater for any given location, as shown in Fig. 11, than variations due to a difference in geographical location. The available information indicates that changes in weather conditions over a period of a few days do not alter the optimum indoor temperature.

Sudden chilling (shock) of persons entering a cooled and air conditioned space during the summer months, may at times be important. It is due to the rapid evaporation of perspiration which accumulated on the skin and in the clothing during previous subjection to hot and humid outside conditions. While studies[25] have shown that for healthy individuals this *shock* is not harmful under some conditions it may be unpleasant or

even harmful. People entering and remaining in cooled spaces for short periods, 15 min or less, may be satisfied with less cooling. For long occupancy very little deviation from the optimum effective temperature is indicated.

An exit shock upon re-entering a warm atmosphere is equally plausible. Experiments at the A.S.H.V.E. Research Laboratory[26] indicated no demonstrable harm to a healthy individual. Adaptation occurred as soon as normal perspiration was established. Mild exercise shortened the adaptation time.

A great number of persons seem to be fairly content in summer with a higher plane of indoor temperature. Studies by the University of Illinois[27] in cooperation with the A.S.H.V.E. Committee on Research indicate that effective temperatures as high as 74.5 deg are acceptable in the living quarters of a residence, and while this condition is not representative of optimum comfort it provides sufficient relief in hot weather to be acceptable to the majority of users, in the interest of economy. Individual differences of comfort among the minority should be counteracted by suitable clothing.

Satisfactory comfort conditions for persons at work[28] vary depending upon the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for comfort. Clothing has been evaluated for its overall insulation effects by a physical unit, the *clo* which equals 0.116 C deg per (kilogram calorie) (square meter) (hour)[29]. Yaglou[30] criticizes the concept of overall insulation, and points out that different parts of the body require different amounts of insulation. The literature on effects of clothing is difficult to coordinate at the present time as much of it is still in military service reports. When these reports are published and amplified a considerable amount of useful material will be available.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[31]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68 F with natural indoor humidities. For children (having high metabolism) at school, in winter clothes, 70 F has been considered correct, while in a gymnasium 55 F has been recommended.

REFERENCE

[1] Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27).

[2] A.S.H.V.E. RESEARCH REPORT No. 1031—Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. J. Coggins (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936 p. 133).

[3] A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[4] Tobacco Smoke Control—A Preliminary Study, by Charles S. Leopold (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 255).

[5] A.S.H.V.E. RESEARCH REPORT No. 921—Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 191) A.S.H.V.E. RESEARCH REPORT No. 965—Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S.H.V.E. TRANSAC-

TIONS, Vol. 39, 1933, p. 357). A.S.H.V.E. RESEARCH REPORT No. 985—Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 271). The Nature of Ions in Air and Their Possible Phsyiological Effects, by L. B. Loeb (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 101). The Influence of Ionized Air Upon Normal Subjects, by L. P. Herrington (*Journal Clinical Investigation*, 14, January, 1935). The Effect of High Concentration of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L. P. Herrington and Karl L. Smith (*Journal Industrial Hygiene*, 17, November, 1935). Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, By C.-E. A. Winslow and L. P. Herrington (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 119).

[6] *The British Medical Journal*, Editorial, June 25, 1932, p. 1182.

[7] The Mechanism of Heat Loss and Temperature Regulation, by Eugene F. DuBois (Lane Medical Lectures, Stanford University Publications, *Medical Science*, Vol. 3, 1937, No. 4, p. 348; also *Transactions of the Association of American Physicians*, Vol. 51, 1936, p. 252.

[8] Exchanges of Heat and Tolerance to Cold in Men Exposed to Outdoor Weather, by E. J. Adolph and G. W. Molnar (*American Journal of Physiology*, Vol. 146, 1946, p. 507.)

[9] A.S.H.V.E. RESEARCH REPORT No. 1108—Cardiac Output, Peripheral Blood Flow and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures, by F. K. Hick, R. W. Keeton, (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 123).

[10] A.S.H.V.E. RESEARCH REPORT No. 1107—Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice, by C.-E. A. Winslow, T. Bedford, E. F. DuBois, R. W. Keeton, A. Missenard, R. R. Sayers and C. Tasker. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 111).

[11] A.S.H.V.E. RESEARCH REPORT No. 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245).

[12] Rate of Insensible Perspiration (Diffusion of Water) Locally Through Living and Through Dead Human Skin, by G. E. Burch, and T. Winsor (*Archives of Internal Medicine*, Vol. 74, 1944, p. 437).

[13] The Influence of Clothing, Work and Air Movement on the Thermal Exchanges of Acclimatized Men in Various Hot Environments, by N. A. Nelson, W. B. Shelly, S. M. Horvath, L. W. Eichna, and F. F. Hatch (*Journal of Clinical Investigations*, Vol. 27, 1948, p. 209).

[14] A.S.H.V.E. RESEARCH REPORT No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 129). A.S.H.V.E. RESEARCH REPORT No. 672—Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 353). A.S.H.V.E. RESEARCH REPORT No. 690—Air Motion, High Temperatures and Various Humidities—Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 167). A.S.H.V.E. RESEARCH REPORT No. 718—Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 101). A.S.H.V.E. RESEARCH REPORT No. 719—Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W. J. McConnell, C. P. Yaglou and W. B. Fulton (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 123). A.S.H.V.E. RESEARCH REPORT No. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541). A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry—Physiological Reactions of Individual Workers to High Effective Temperatures, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59). A.S.H.V.E. RESEARCH REPORT No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F C Houghten, A. A. Rosenberg and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 185).

[15] A.S.H.V.E. RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 157).

[16] Heat Disease: Clinical and Laboratory Studies, by M. W. Heilman and E. S. Montgomery (*Journal of Industrial Disease and Toxicology*, 18:651, 1936).

[17] Performance in Relation to Environmental Temperature, by L. W. Eichna, W. B. Bean, W. F. Ashe and N. Nelson, (*Bulletin of Johns Hopkins Hospital*, Vol. 76, 1945, p. 25–58).

[18] Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures, by C. M. Humphreys, Oscar Imalis and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March, 1946, p. 101).

[19] Life, Heat and Altitude, by David B. Dill (Harvard University Press, Cambridge, 1938).

[20] The Upper Limits of Environmental Heat and Humidity Tolerated by Acclimatized Men, Working in Hot Environments, by Ludwig W. Eichna, William F. Ashe, William B. Bean and Walter B. Shelley (*The Journal of Industrial Hygiene and Toxicology*, Vol. 27, March, 1945, p. 59).

[21] A.S.H.V.E. RESEARCH REPORT No. 1188—Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 403).

[22] A.S.H.V.E. RESEARCH REPORT No. 673—Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361). A.S.H.V.E. RESEARCH REPORT No. 691—Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193). A.S.H.V.E. RESEARCH REPORT No. 717—Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89). A.S.H.V.E. RESEARCH REPORT No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 315). How to Use the Effective Temperature Index and Comfort Charts, by C. P. Yaglou, W. H. Carrier, Dr. E. V. Hill, F. C. Houghten and J. H. Walker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 410).

[23] A.S.H.V.E. RESEARCH REPORT No. 1196—Comfort with Summer Air Conditioning, by Thomas Chester, N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber and C. Tasker. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 107).

[24] Conditions for Comfort, by C. S. Leopold (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1947, p. 117).

[25] A.S.H.V.E. RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571). Physiologic Adjustments of Human Beings to Sudden Change in Environment, by N. Glickman, T. Inouye, S. E. Telser, R. W. Keeton, F. K. Hick and M. K. Fahnestock, A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, July, 1947, p. 101.

[26] A.S.H.V.E. RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145).

[27] A.S.H.V.E. RESEARCH REPORT No. 1012—Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207).

[28] A.S.H.V.E. RESEARCH REPORT No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 315).

[29] A Practical System of Units for the Description of the Heat Exchange of Man with His Environment, by A. P. Gagge, A. C. Burton, and H. C. Bazett (Science, Vol. 94, 1941, p. 428).

[30] Thermal Insulation of Clothing, by C. P. Yaglou (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, Sept. 1948, p. 107).

[31] Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 383).

CHAPTER 13

AIR CONDITIONING IN THE PREVENTION AND TREATMENT OF DISEASE

Control of Airborne Infection, Value of Air Cooling Under Tropical Conditions, Treatment of Disease, Operating Rooms, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

THE late war caused an increase of interest in the preventive aspects of air conditioning. It re-emphasized the importance of the control of airborne infection and demonstrated the value of air cooling under tropical conditions for the prevention of heat rash, for promoting proper rest and sleep, and in the convalescence of patients.

CONTROL OF AIRBORNE INFECTION

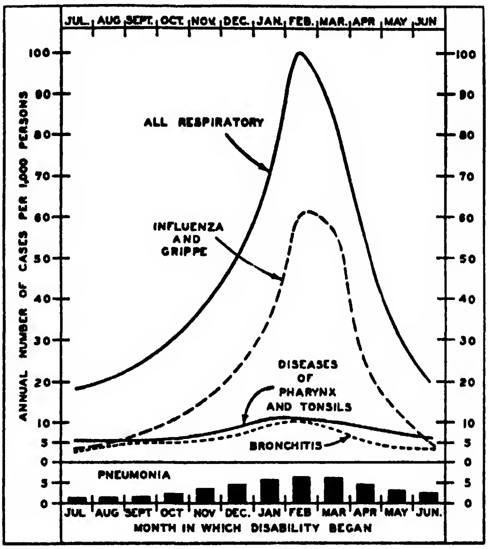
The majority of airborne diseases are spread indoors where people gather. Any program of air sanitation is influenced by a number of factors. In the winter months, the closing of doors, windows and other means of access to the outside air to conserve warmth as well as the crowding of persons indoors provides conditions conducive to a high incidence of contagion. This seasonal phenomenon, illustrated in Fig. 1 which represents a study made by the *U. S. Public Health Service*, will concern the ventilating engineer in so far as air quality, determined by temperature, humidity, air replenishment and type of air movement and by freedom from contamination, is a major intrinsic factor. Apart from the seasonal picture of airborne contagion are such extrinsic factors as rate of turnover of personnel and the marked susceptibility of the recruit in comparison with permanent personnel[1] as shown in Fig. 2 by studies of military personnel housed in barracks. These extraneous variables and the factor of contact infection (direct spray) tend to complicate any evaluation of the effectiveness of air sanitation for elimination of micro-organisms in droplet-nuclei and droplet-dust. Thus, control measures may eliminate consistently 90 per cent of airborne organisms in laboratory tests, but cannot effect a decrease in actual incidence of infection exceeding 30 per cent. Thirty per cent may be the maximal reduction in infection possible by air treatment methods. The distinction should be clearly drawn, therefore, between the effectiveness of a procedure in laboratory tests and its effectiveness and applicability in actually reducing the incidence of airborne disease. On the other hand, recent studies suggest that inhalation of dust borne bacteria is more important than direct inhalation of infectious droplets or droplet nuclei in the spread of respiratory tract infections[2].

The following sequence of events has been postulated as occurring in a large proportion of intra-ward infections: (*a*) ejection of relatively large protected infective particles from patients, (*b*) rapid venting or settling of these particles so that those remaining airborne are in low concentration, (*c*) survival of infective particles to permit the accumulation of high concentrations on surfaces, (*d*) repeated reintroduction of infective particles into the air under the stimulus of ward activities or by air currents of the order of 50 fpm over the floor, and (*e*) extension of infective areas by air turbulence throughout the ward or hospital. The most important link

in this probable infection chain has been demonstrated to be the reintroduction of particles into the air[3].

Intensive studies on air disinfection have indicated two distinct control measures, (*a*) suppression of dust and lint, and (*b*) disinfection of droplet-nuclei.

Well controlled, large scale tests of the various methods of air sterilization conducted in barracks[4, 5] have confirmed the importance of dust control in minimizing the spread of airborne disease, a consideration which has guided the practices of ventilating engineers for a number of years. The importance of the dust factor has been emphasized by many engineers and



Occurrence of diseases causing disability for 8 consecutive days or longer in a group of 100,000 wage earners (10 per cent women) in different industries.
[a] Graph obtained from Dean K. Brundage, U. S. Public Health Service.

Fig. 1. Study of Average Monthly Frequency (1921–1926 inclusive) of Specified Respiratory Diseases[a]

has been convincingly demonstrated by subsequent bacteriologic studies aboard ships.

Treatment of floors and bedclothes with oil emulsions has proved effective in reducing bacterial dispersion by as much as 90 per cent in Army barracks and station hospitals[5]. The incidence of acute respiratory infections was from 10 to 30 per cent lower in barracks with oiled floors and bedclothes than it was in control barracks which received no special treatment. More recent studies, however, have yielded inconsistent results.

An emulsifying mixture, *Fixanol C* containing cetyl pyridinium bromide, when incorporated in the oil-in-water emulsion imparted a bactericidal action to the emulsion. Blankets treated with this substance and oil became bactericidal and retained this property for as long as three months. The possibility of hypersensitivity of an occasional individual to bromide

drugs should be borne in mind when exposing large groups to treated garments or blankets[2].

No simple method for disinfecting droplet-nuclei has yet been devised. Under favorable laboratory conditions, propylene glycol in concentrations of 0.07 to 0.14 milligrams per liter, and triethylene glycol in a concentration of 0.0045 milligrams per liter were highly germicidal for most airborne bacteria in clean air when the relative humidity was between 40 and 60 per cent[6, 7, 8]. Apparatus for the production of glycol vapor and an independent duct system for carrying this vapor and diluting air for large rooms or spaces and unit type vaporizers for small spaces have been described recently[9]. There is also available a device for the automatic regulation of glycol vapor in the air called the glycostat. This instrument has been calibrated to measure the degrees of saturation of the air with glycol vapor by direct reading of the variations in the intensity of light reflected from the glycol condensing surface of the wheel of the instrument[10]. Under

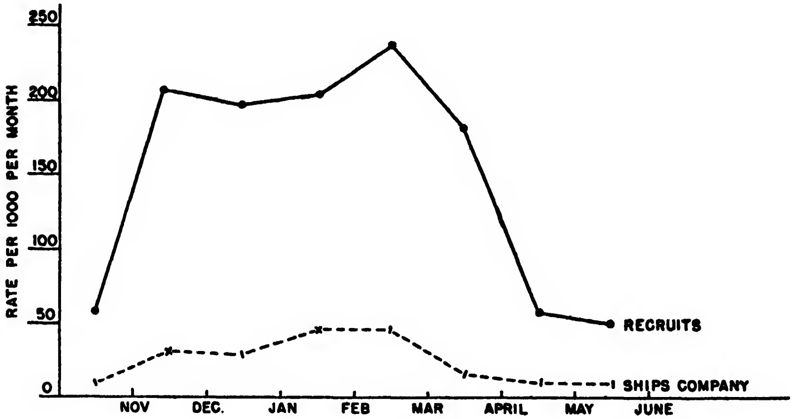


Fig. 2. Monthly Incidence of Acute Respiratory Illness Among Naval Recruits and Ship's Company (Permanent Personnel)[1]

practical conditions, however, particularly in the presence of dust in the air, glycol effectiveness is much reduced. The use of other chemical aerosols that have been tried is limited by their toxicity, odor, or destructiveness to fabrics and metals.

Ultraviolet radiation of floors and upper air has been studied extensively at the Naval Training Center, Sampson, N. Y. In barracks housing naval recruits, hospital admissions for respiratory infections (mostly *catarrhal fever*) were 25 per cent lower in a group of men exposed to ultraviolet radiation—(2537 Angstrom Units, 1 to 7 ergs per (cm^2) (sec) at bed level)—than they were in adjacent control barracks without ultraviolet radiation[11]. Sources of ultra violet radiation should be so situated as to protect the eyes of the occupants of the room from direct or reflected rays. A combination of ultraviolet radiation and dust control measures is believed

to be more effective than when either one of the two is used alone, but the proof for this has yet to come.

The present status (1947) is admirably reviewed by the Committee on Sanitary Engineering of the *National Research Council*[12], and by a subcommittee of the *American Public Health Association*[13]. Both committees feel that the problem of air disinfection is still in the experimental stage.

More experimentation is needed for arriving at a definite conclusion concerning its use in industry and public buildings.

VALUE OF AIR COOLING UNDER TROPICAL CONDITIONS

The commissioning of a class of naval hospital ships with all wards, laboratories and living spaces air cooled is a notable achievement to provide better treatment of patients, especially those suffering from extensive burns, by control of environmental factors. Although statistics are not at hand to indicate the deaths or retarded recoveries of patients due to lack of air cooling in ships operating in tropical waters, it is generally agreed among competent observers that high temperature and humidity are major factors in prolonging disability and increasing mortality of the sick and injured. Physiologic data obtained on healthy men, moreover, show the large loss of body fluids and the stress on the cardiovascular system in terms of increased pulse rate when these men are continuously subjected to high temperatures. Even at rest about 50 cc of fluid per hour are lost as sweat[14] through intact skin. In burn patients the difficulty, encountered in temperate climates, of maintaining fluid and electrolyte balance is tremendously augmented by the additional evaporative fluid loss in hot environments.

Frequently from 50 to 75 per cent of personnel aboard naval vessels operating in tropical waters are afflcted with heat rash to a degree that interferes with rest and sleep. In carefully controlled experiments[14] it was possible to produce a fulminating type of rash in all men living continuously at an effective temperature of 85 (90 F dry-bulb and 83 F wet-bulb). In the control group, 12 out of 24 hr were spent in a relatively cool atmosphere of 75 ET (80 F dry-bulb, and 70 F wet-bulb). These men either remained free from heat rash or occasionally developed a mild form. Thus, intermittent cooling to a degree which prevented sweating in men at rest eliminated a serious handicap to good performance of duty.

In both laboratory tests and aboard hospital ships a relatively cool living environment of 76 to 78 ET provided an atmosphere conducive to rest and sleep without excessive sweating. Berthing spaces tended to have extremely low odor levels. Motivation, initiative and alertness, in contrast to the usual irritability and lack of incentive incident to residence in tropical climate, were maintained[15].

Little has been done, however, to obtain practical methods for application of air conditioning under heavy heat loads and on the enormous scale that would be needed to modify life in the tropics. It is not improbable that cooled houses in a tropical climate, if used consistently for one generation, might modify the whole character of a population. The obvious advantages of part time cooling on personnel to promote rest and sleep in tropical areas would provide a prophylactic measure of great potential importance.

TREATMENT OF DISEASE

In the past few years considerable progress has been made in using air conditioning as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for

premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, the control of allergic disorders, and for the physiological effects in industry.

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger of anesthetic gases; summer cooling with some dehumidification tends to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering aids the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern anesthetic gases and apparatus. Ether administered by the old drop method gives rise to an explosive mixture, but in practice this method is still regarded as comparatively safe. When ether is mixed with pure oxygen, or nitrous oxide in certain concentrations, the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures.

Of the anesthetic gases nitrous oxide alone does not explode but supports combustion. Ether, vinyl ether, ethylene, and cyclopropane are as potentially dangerous as gasoline or illuminating gas in the home[16]. Chloroform does not explode violently in contact with flame but decomposes to liberate phosgene. All of the anesthetic gases and vapors except ethylene are heavier than air. Although the incidence of injury or death from explosion is negligible compared with other hazards in the operating room, the dramatic features surrounding an explosion justify continued investigation to eliminate the hazard.

During the course of ethylene anesthesia, the mixture, usually 80 per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may accumulate to any lower concentration, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patient's lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel.

In a study[17] of 230 anesthetic explosions and fires, 70 per cent of the explosions and 60 per cent of the deaths were caused by igniting agents other then static sparks. In 1941 the *National Fire Protection Association*[18] made certain recommendations for safe practice based on available information. Some of these recommendations are:

Windows should be kept closed so that the air conditioning system can prevent pooling of explosive anesthetic gases. Twelve air changes per hour and a humidity of 55 per cent are advised. If a higher humidity were compatible with the well being of the patient and personnel, it should be maintained. All electrical installations should comply with the standards set by the *National Electrical Code* for use in explosive situations. Cautery

equipment should not be used in hazardous locations. To prevent static sparks, all bodies in an operating room should be conductive or coupled. It is essential that adequate grounding be provided for the floor and every object in the operating room. Conductive rubber should be used on shoes, leg tips, operating table coverings and all rubber parts of the anesthesia equipment. All furniture in contact with the floor should be metal. In the absence of complete grounding facilities, the simple method of intercoupling patient, operating table, anesthetist and gas machine at ground potential may be used.

Experience has shown that neither high humidity nor intercoupling devices have eliminated the danger from static electric discharge. The removal of gas concentrations from the operating table area by means of specially devised exhaust ventilation should be thoroughly tested. Portable duct systems as installed aboard ship should be acceptable. Serious explosions can occur in a closed system but proper precautions will reduce this hazard to a minimum.

It should be realized that when a room and the occupants have been completely grounded there is always the possibility that the patient or the operator might receive a dangerous shock if a short circuit developed in any of the electrical equipment.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards by the *University of Pittsburgh*, the A.S.H.V.E. Research Laboratory, and the *U. S. Bureau of Mines* has led to a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by displacement and, because of its flame quenching properties, it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges and this quality also increases the safety factor of anesthetic administration.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss, although there may be little more than 0.8 F variation in the rectal temperature during the course of the operation[19]. The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months signify the need for proper cooling. A comparison of surgeons' statements who operate in both air conditioned and non-air conditioned rooms strongly indicates that the recuperative power of the patient is greater when operated upon in air conditioned rooms[19].

Although the comfortable air conditions for the operators are not identical with those for the patient, it is usually not difficult to compromise within a range of 55 to 60 per cent relative humidity and 72 to 80 F temperature. The work just cited reported that 68 to 70 F effective temperature not only furnished comfort for the operating room workers, but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient

locally or by suitable covering according to body temperature in individual cases.

In the control of airborne infection in the operating room the prevention of dispersal of infectious materials into the air, control of dust and proper ventilation supersede attempts to remove or kill pathogenic organisms. The bacterial content of conditioned operating rooms is generally lower than that of non-conditioned rooms.

Bacterial counts aboard an air conditioned submarine were found to be exceptionally low and not cumulative with time although all of the air was recirculated for more than 12 hours[20] without replenishment. The removal of bacteria by the process of air cooling and condensation of moisture out of air merits further study[21].

The degree of air contamination can be reduced by proper ventilation if velocity of air over the floor does not exceed 50 fpm. Research is in progress on the use of filtered air flowing through a system of mechanical cleaners which protect the patient against infection from attendants and from bacteria-containing air in the corridor or ward[22].

Operations are frequently postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of air-borne allergens, therefore, is in some cases an important function of the air conditioning system in preparing patients for operation.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 12 air changes per hour of filtered and properly conditioned air without recirculation during the course of anesthesia. A separate exhaust fan system is usually necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather and to minimize drafts. The air flow of 8 to 12 air changes in operating rooms should: (1) reduce the concentration of the anesthetic to well below the pharmacologic threshold in the vicinity of the operating personnel, (2) remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives, and (3) provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by thermal insulation of sterilizing equipment and by thorough exhaust ventilation of sterlizing rooms adjoining the operating rooms. An air conditioned recovery ward in connection with the air conditioned operating room is of great value in stabilizing peripheral circulation and in reducing excessive loss of fluids on hot humid days.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because infants' heat regulating systems are not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain normal body temperatures. The resistance to infection is low and mortality rate high.

Air Conditioning Requirements

The optimum air conditions for growth and development of premature infants were determined by extensive research[23] at the Children's Hospital, Boston, Mass., using four valid criteria, namely, stability of body tempera-

ture, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to 100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 per cent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Children's Hospital were kept at relative humidity between 25 and 50 per cent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low as under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment

Many of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A ventilation rate, between 8 and 12 air changes, is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation should not be used in these wards owing to odors and the possibility of infection. There should be a frequent change in spray water.

Control of Airborne Infection

The protection of the premature and older infant against infection is of the utmost importance. It was found in one installation equipped with air conditioning, germicidal lights and mechanical barriers that air conditioning alone did not prevent the spread of respiratory cross-infections. Bactericidal ultraviolet light barriers and air conditioning or mechanical barriers and air conditioning were efficient[24].

FEVER THERAPY

Artificial production of high fever in man can be considered an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body.

Although the action may be direct and specific by destruction of the invading organisms within the safe human limits, fever therapy exerts much of its benefit through the improvement of the mechanism of bodily defense. A serious challenge to the theory on which fever therapy is based comes from the demonstration that high fever causes a reduction in the concentration of circulating antibodies in experimental animals.

Patients for fever therapy should be carefully selected. The most serious complications which may arise are heat stroke, heat exhaustion and circulatory collapse. The chief minor complications are heat cramps, fever blisters and mild dehydration.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis and chorea.

The most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life. However, the use of fever therapy has decreased since penicillin has been found so effective in the treatment of gonorrhea and syphilis. Mild fever, up to 101 F for one hour has recently been used in the treatment of rheumatoid arthritis. This degree of fever is not bactericidal but is believed to stimulate the body defense mechanism.

Equipment for Production of Fever

Artificial fever can be induced by injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms, or by physical methods using hot baths, radiant heat cabinets, hot humidified air cabinets, or by short wave diathermy in combination with a cabinet.

The relative advantages of various methods have been evaluated clinically[25]. Among the devices for the production of fever by physical means, the one most widely used is the hot humid air or *air conditioned* cabinet. This apparatus was developed at the *Kettering Institute for Medical Research* at Miami Valley Hospital in Dayton, Ohio.

In the earlier studies of the Society, temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus[26] using

these same principles has proved efficient as a means of inducing and maintaining fever in a body with small likelihood of burns because of the comparatively low dry-bulb temperatures.

When heat is necessary in treating legs or arms, such media as short or long wave diathermy, micro-waves, infrared, water baths, etc. have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks. Local heating has been somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease.

This procedure, however, is not without danger. Elevation of tissue temperature increases cell metabolism and the need for oxygen. The inadequate blood supply and oxygen deficiency may lead to tissue death or gangrene. Application of heat to the trunk or abdomen with consequent reflex dilatation of the vessels of the extremities eliminates this danger of local heat application.

Short wave diathermy within the cabinet during the induction phase has been used. When the desired body temperature has been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. The two underlying principles in the production of fever by the hot, humid air cabinet are: (1) the transfer of heat by conduction from the circulating hot air to the body and (2) prevention of heat loss. The latter is more important. In an atmosphere of high humidity, the heat loss by evaporation is markedly decreased.

COLD THERAPY

Cold as an anesthetic agent was advocated by Allen several years ago[27]. Freezing of the tissue must be avoided. For certain patients, in whom amputation of an extremity is indicated, the application of a tourniquet with cooling of the affected extremity down to near freezing (5C or 40F) is of value. The patient, following this procedure can be prepared for surgery without the handicap of absorption of septic products and severe pain. This procedure has proven especially valuable in the neglected diabetic patient with an infected gangrenous extremity. Time for treatment of coma and hydration of the patient is gained. However, if amputation of an extremity is not indicated, the application of a tourniquet and packing in ice are dangerous procedures since loss of the limb usually results. An extremity with inadequate blood supply can be readily cooled without the use of a tourniquet but such an extremity is also usually eventually lost. Theoretically, cooling is said to reduce the metabolism of the tissue with suspension of the vital processes. It also reduces the blood flow to practically zero and few extremities with inadequate blood supply remain viable or recover.

Packing in ice or use of low temperatures is contra-indicated in the treatment of patients with frostbite, immersion foot or trench foot. The affected extremities should be exposed to the air in a cool room and not rubbed with snow or packed in ice. The lowering of temperature by packing the body in ice for treatment of cancer has not proven successful.

The methods used for refrigeration depending upon available facilities are as follows[27]:

(1) Cracked or shaved ice which is simple and has the advantage of not freezing tissues. However, it is cumbersome and sloppy to handle and is unsuited to prolonged treatments.

(2) Use of ice in a pail for immersion of local parts.

(3) Special boxes for holding ice with padded or curtained openings for the limb.

(4) Bare ice bags and cloth bags for iced wet dressings for prolonged treatments and convenience.

(5) A double chambered cabinet using dry ice has been constructed.

(6) Electrical refrigerating apparatus, consisting of a compact noiseless unit that pumps fluid to various types of applications, is available. The applicators may be in the form of blankets containing rubber tubes suitable for covering the entire body or all or part of a limb. Special applicators are available for insertion into various body cavities and for inducing dental anesthesia.

(7) An air chamber at regulated temperature for treatments of frostbite and immersion foot, and amputation stumps.

The electrical apparatus is costly but has the advantages of thermostatic regulation, light weight, freedom of movement, and permits prolonged treatments with heat as well as cold over the range of temperatures therapeutically desirable.

ALLERGIC DISORDERS

Hay fever, asthma, eczema and contact dermatitis are classified as allergic disorders. The allergic individual responds to contact with a variety of substances, which are innocuous to a non-allergic person, with severe manifestations of hypersensitivity.

These substances are known as allergens and consist of airborne irritants such as dusts, molds, feathers, pollens, animal dander and others; of food protein such as milk, wheat, eggs, etc., or of simple chemicals brought in contact with the skin. They may enter the body by various routes of which inhalation is the most common type. Ingestion of offending food substances is not infrequent.

The offending substance reacts with the sensitized cells of the mucous membranes or skin. During this reaction, histamine or a histamine-like substance is released which causes (a) increased capillary permeability, (b) secretion of mucus and (c) muscular contraction. In the eyes and nose this produces itching, redness and lacrimation or rhinorrhea, in short, the symptoms of hay fever. In the lungs it causes, in addition to the secretory response, a contraction of the smooth muscles of the bronchi resulting in bronchial asthma.

It is commonly known that non-specific environmental factors such as dust irritating gases, change of temperature and humidity may precipitate asthmatic attacks in allergic subjects even in the absence of exposure to specific allergens. It is assumed that the presence of frequent allergic bronchial constriction renders the smooth muscles of the bronchi so sensitive to various non-specific stimuli that the threshold of their response to such irritation is considerably lower than that of a non-allergic individual.

Air Conditioning Apparatus

Of all the measures to relieve a specifically sensitive individual, elimination of exposure to the responsible allergen is the most efficient though not always a practical form of treatment. In recent years considerable effort has been made to accomplish such elimination by removal of respiratory allergens from the air of enclosures by filtration or other air conditioning processes.

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory in many cases, but since dust and smoke frequently

cause asthmatic attacks, it is desirable that an air filter, to be of full value in the treatment of asthma, should remove all possible dusts and pollens regardless of size or amount. Electrostatic air cleaners are more efficient than most commonly used types for capturing very fine dust.

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F and a relative humidity well below 50 per cent proved satisfactory[28]. Direct drafts, overcooling or overheating are apt to initiate or aggravate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. For all practical purposes filtration gives only temporary relief. In mild cases sleeping in an air conditioned space may make it possible for the individual to pass more comfortable nights. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond to treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief[29]. Pollen cases are usually relieved of most of their symptoms within 1 to 3 hr after exposure to properly filtered air. A pollen-free atmosphere is especially valuable when desensitization has given little or no relief, and when desensitization is not advisable owing to intercurrent illness.

OXYGEN THERAPY

Oxygen therapy is used to prevent or relieve anoxia. Some of the more important clinical conditions in which oxygen treatment is beneficial include pneumonia, severe anemia, cardiac decompensation, pulmonary atelectasis, asphyxia and asthma. The effectiveness of oxygen therapy is dependent on the concentration of the oxygen in the inspired air or the partial pressure of oxygen in the pulmonary alveoli.

Oxygen is usually administered by nasal catheter, face mask or tent[30]. The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. Air conditioning is applied to the oxygen tent or chamber through reconditioning of the atmosphere in a closed circuit. Excessive heat, moisture and carbon dioxide are removed.

Oxygen Tents

In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the

patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hour gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are confining to the patient. They may terrify the restless and delirious patient. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. However, with attention to details, the patient can be made quite comfortable.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber. The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents.

The temperature and humidity requirement in oxygen therapy depends primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias[31] prescribed conditions should be a temperature of 60 to 75 F, humidity 20–50 per cent, moderate air movement, oxygen concentration of 50 per cent, and carbon dioxide of less than one per cent.

Oxygen in Aviation

An important application of the principle of oxygen therapy is in aviation. At the present time all high altitude military airplanes in this country are provided with gaseous oxygen equipment and military personnel are required to utilize oxygen at all times while in flight above 15,000 ft, or between 12,000 to 15,000 ft for longer than two hours, or between 10,000 to 12,000 ft for longer than six hours. The use of oxygen in commercial aviation will depend on the height and duration of the flights as well as the state of health of the passengers. The necessity for portable, comfortable equipment, the possible fire hazards due to smoking, and the use of oxygen on sleeper planes are some of the difficulties facing civil airline operators. The pressure cabin airplane is a solution to the problem.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expenses which may not be justified. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of conventional heating equipment in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may

preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

F. L. Grocott of the Anglo-Iranian Oil Co. states that in Iran, the medical staff after 10 years experience with air conditioning, demand a uniform environment of 75 F and 50 per cent relative humidity (70 ET) under all summer outside conditions for general wards and treatment rooms and 70 F with 30 to 50 per cent relative humidity (65–66 ET) for winter conditions. In the operating rooms, 70 F and 50 per cent relative humidity (66 ET) is demanded all the year 'round, although the annual external range is 40 F to 120 F. No ill effects have been noted in the medical personnel though they are exposed to changes from external to internal conditions many times daily. Temperature shock in either direction seems to create discomfort for a short interval but if the individual is in good health, no injury results[32].

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, thyroid crisis, and in a variety of other ailments which often accompany summer heat waves.

Problem of Odors

The evacuation of battle casualties in aircraft and their subsequent hospitalization have stimulated efforts to minimize odors arising from draining wounds, old odorous casts, and gangrenous wounds. For aircraft, chemical sprays and vapors, perfumes, oxidizing gases and ventilation methods are unsatisfactory. An ideal deodorant would purify the air by means of odor adsorption so that subsequently the air can be recirculated. Based upon the effectiveness of activated carbon commercially and industrially to adsorb odors, individual adsorption units have been used successfully. In hospital wards the question of superiority of adsorption methods for elimination of odors over other methods remains to be answered. The present status of the problem is that the commercial aspect is highly controversial.

REFERENCES

[1] Factors in the Control of the Spread of Acute Respiratory Infections with Reference to Streptococcal Illness and Acute Rheumatic Fever, by S. M. Wheeler and T. D. Jones (*American Journal of the Medical Sciences*, 209:58, 1945).

[2] Dust Control, Report on Suppressive Measures for the Control of Dust, to the Standard Methods Committee for the Examination of Germicides and Antibacterial Agents, by Clayton G. Loosli (*American Journal of Public Health*, Vol. 38, p. 409, March, 1948).

[3] Laboratory and Field Studies of Glycols and Floor-Oiling in the Control of Air-Borne Bacteria, by A. P. Krueger *et al* (*U. S. Naval Medical Bulletin*, 42:1288, 1944).

[4] Bacterial Content of Air in Army Barracks, by H. M. Lemon, H. Wise and M. Hamburger (*War Medicine*, 6:92, 1944).

[5] A Study of the Nature and Control of Air-Borne Infection in Army Camps, by O. H. Robertson, M. Hamburger, C. G. Loosli, T. T. Puck, and H. M. Lemon (*Journal of the American Medical Association*, 126:993, 1944).

[6] Lethal Effects of Triethylene Glycol Vapor on Air-Borne Bacteria and Influenza Virus, by O. H. Robertson *et al* (*Science* 97:142, 1943).

[7] The Present Status of Glycol Vapors in Air Sterilization, by M. Hamburger, Jr., O. H. Robertson, and T. T. Puck (*American Journal of the Medical Sciences*, 209:162, 1945).

[8] Summary of a 3-Year Study of the Clinical Applications of the Disinfection of Air by Glycol Vapors, by T. N. Harris and J. Stokes, Jr. (*American Journal of the Medical Sciences*, 209:152, 1945).

[9] Triethylene Glycol Vapor Distribution for Air Sterilization, by Edward Bigg, B. H. Jennings, and F. C. W. Olson (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1947, p. 103.

[10] Glycol Vapors for Disinfecting Purposes, Editorial, (*Journal of the American Medical Association*, 133:696, March 8, 1947).

[11] Ultra-Violet Light Control of Air-Borne Infections in a Naval Training Center, by S. M. Wheeler, H. S. Ingraham, A. Hollaender, N. D. Lill, J. Gershon-Cohen, and E. W. Brown (*American Journal of Public Health*, Vol. 35, p. 457, 1945).

[12] Recent Studies on Disinfection of Air in Military Establishments. (*American Journal of Public Health*, Vol. 37, p. 189, Feb. 1947.)

[13] The Present Status of the Control of Airborne Infections. (*American Journal of Public Health*, Vol. 37, p. 13, Jan. 1947).

[14] A Comparative Study of the Effect on Men of Continuous Versus Intermittent Exposure to a Tropical Environment, by N. Pace, M. B. Fisher, J. E. Birren, G. C. Pitts, W. A. White, Jr., W. V. Consolazio, and L. J. Pecora (Research Project X-205, Report No. 2, *Naval Medical Research Institute*, Bethesda, Md., May, 1945).

[15] Environmental and Physiologic Studies Aboard an Air-Cooled Hospital Ship En Route from Norfolk, Virginia to Canal Zone (U. S. S. TRANQUILITY (AH-14), 6–13 June 1945), by A. R. Behnke (Research Project X-205, Report No. 4, *Naval Medical Research Institute*, Bethesda, Md., September, 1945).

[16] Fundamentals of Anesthesia (American Medical Association Press, Chicago, Ill., 2nd Edition, 1944, p. 204).

[17] The Hazard of Fire and Explosion in Anesthesia, by B. A. Green (*Anesthesiology* 2:144, 1941).

[18] Control of Physical Hazards of Anesthesia, by R. M. Tobell and A. W. Friend (*Canadian Medical Association Journal*, 46:560, 1942).

[19] A.S.H.V.E. RESEARCH REPORT No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 161).

[20] Unpublished Naval Studies by A. R. Behnke and O. Schneider (1940).

[21] Disinfection of Air by Air Conditioning Processes, by C. P. Yaglou and Ursula Wilson (*American Association for the Advancement of Science*, Publication No. 17, p. 129).

[22] The Control of Cross-Contamination by the Use of Mechanical Barriers, by J. A. Reyniers (*Aerobiology, American Association for the Advancement of Science, Symposium*, 17:254, 1942).

[23] The Premature Infant: A Study of the Effect of Atmospheric Conditions on Growth and on Development, by K. D. Blackfan, C. P. Yaglou and K. McKenzie (*American Journal Diseases of Children*, 46:1175, 1933).

[24] Observations on the Control of Respiratory Contagion in the Cradle, by I. Rosenstern (*Aerobiology, American Association for the Advancement of Science, Symposium*, 17:242, 1942).

[25] Physical Medicine, by F. H. Krusen (W. B. Saunders Co., Philadelphia and London, 1941).

[26] A.S.H.V.E. RESEARCH REPORT No. 1054—Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber, and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 131). A.S.H.V.E. RESEARCH REPORT No. 1161—Fever Therapy Locally Induced by Conditioned Air, by M. B. Ferderber, F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 307).

[27] Refrigeration for Anesthesia and Therapy, by L. W. Crossman and S. K. Safford (*The Modern Hospital*, 64:86, 1945).

[28] The Effect of Low Relative Humidity and Constant Temperature on Pollen Asthma, by B. Z. Rappaport, T. Nelson, and W. H. Welker (*Journal of Allergy*, 6:111, 1935).

[29] Hospital Air Conditioning, by C. P. Yaglou (The Environment and Its Effect Upon Man, *Harvard School of Public Health*, p. 244, 1939).

[30] Principles and Practices of Inhalational Therapy, by A. L. Barach (J. B. Lippincott Co., Philadelphia, London, Montreal, 1944).

[31] The Management of Pneumonia, by J. G. M. Bullowa, (*Oxford University Press*, p. 260, 1937).

[32] What Are the Right Conditions for Comfort Cooling, by Cyril Tasker, (*Heating, Piping, and Air Conditioning*, Aug., 1948, p. 84).

CHAPTER 14

HEATING LOAD

General Procedure, Design Outdoor Weather Conditions, Inside Temperatures, Attic Temperatures, Temperatures in Unheated Spaces, Ground Temperatures, Basement Temperatures and Heat Loss, Transmission Heat Loss, Infiltration Loss, Selection of Wind Velocities, Auxiliary Heat Sources, Intermittently Heated Buildings, Residence Heat Loss Problems

IN the design of a heating system, an estimate must be made of the maximum probable heat loss of each room or space to be heated, based on maintaining a specified inside air temperature during periods of design outdoor weather conditions. The heat losses may be divided into two groups, namely (1) the transmission losses or heat losses through the confining walls, floor, ceiling, glass or other surfaces and (2) the infiltration losses or heat losses due to air leakage through cracks and crevices, around doors and windows, opening of doors and windows, or heat required by outside air used for ventilation.

GENERAL PROCEDURE

The general procedure for caculating heat losses of a structure is:

1. Select the design outdoor weather conditions: temperature, wind direction and wind velocity. The data on climatic conditions given in Table 1 and the isotherms of average design temperature in Fig. 1 will be useful but should be applied with judgment as suggested in the section Design Outdoor Weather Conditions.

2. Select the inside air temperature, at the 60-in. breathing line or the 30-in. line which is to be maintained in each room during the coldest weather. (See Table 2).

3. Estimate temperatures in adjacent unheated spaces and the attic. The attic temperature need not be estimated if the combined roof and ceiling coefficient is used.

4. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 6).

5. Measure amount of net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building, using inside dimensions.

6. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet, and the temperature difference between the inside and outside air. (See Items 1, 2, and 3).

7. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack, wind velocity, and the temperature difference between the inside and outside air; the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 8).

8. When positive ventilation using outdoor air is provided by an air heating or an air conditioning unit, the heat required to raise the outside air to room temperature must be provided by the unit; if mechanical exhaust from the room is provided, in amount equal to the fresh air drawn in by the unit, the natural infiltration losses must also be provided for by the unit. If no mechanical exhaust is used, and the outdoor air quantity equals or exceeds the amount of natural infiltration which would occur without ventilation, the natural infiltration may be neglected.

9. The sum of the heat losses by transmission (Item 6) through the outside walls and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent

(Item 7) of the cold air entering by infiltration, or required to replace mechanical exhaust, represents the total heat loss equivalent for any building.

DESIGN OUTDOOR WEATHER CONDITIONS

There are no hard and fast rules for selecting the design outdoor weather conditions to be used for a given locality or type of building or heating system, and the problem is to some extent a matter of judgment and experience. The outside design temperature is seldom taken as the lowest temperature, or even the lowest daily mean temperature ever recorded in a given locality. Such temperatures are rarely repeated in successive years. Likewise the wind direction and velocity prevailing at the time of design outside conditions frequently are entirely different from those prevailing during the winter season.

The A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions has recommended the adoption for heating load calculations of an *outside design temperature which is equalled or exceeded during 97½ per cent of the hours in December, January, February and March.*

Complete data of this nature are not available but Column 7, Table 1, lists this recommended design temperature based on airport station readings for the period indicated, generally a five year period 1935–1939 inclusive. It is pointed out that in most cases these stations are outside of the city and that these data would apply primarily to rural areas. A comparison of the lowest recorded temperatures, with due regard to the period of record, makes it possible to determine an equivalent design temperature for city stations. In general the use of the airport data for buildings within an adjacent city will not make any appreciable difference in design load.

The calculation of these design temperatures is being carried on but due to the extensive amount of work involved will not be completed for some time. It should also be noted that the period of record for the stations listed occurred in what is known as a warm cycle and when longer periods are used it is expected that many of these design temperatures will be somewhat lower.

Because of the limited data available, design temperatures in common use are listed in Column 9. Many of these values were furnished by A.S.H.V.E. members—the balance were taken from an ACRMA Bulletin[1], manufacturers' publications and other sources[2], and a few were estimated. The map, Fig. 1, shows isotherms approximated for these design temperatures. They may be used as a guide for localities not listed in the table. Interpolation between these lines is suggested and due consideration must be given to elevations and other local conditions.

Column 8 of the table gives the maximum wind velocity which occurred with temperatures the same as, and lower than, those shown in Column 7. Winter average velocities for all temperatures are given in Column 10.

Column 6 lists the average annual minimum temperature which is the average of readings of the one lowest temperature occurring for each year the station has been in existence. It is of interest as a guide to the lowest temperature to be encountered except for an occasional extreme of short duration.

Finally, large differences in climate occur within relatively small distances of Weather Bureau stations in hilly and mountainous regions. Experience and judgment are necessary to deal properly with this factor.

Table 1. Winter Climatic Conditions[a]

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Design Dry-Bulb Temp. on TAC 97½% Basis[f] °F	Col. 8 Wind Vel. at Design Temp.[g] Mph	Col. 9 Design Dry-Bulb Temp. in Common Use[h] °F	Col. 10 Avg. Wind Vel.-Dec. Jan.-Feb.[i] Mph
Ala.	Anniston CO	733	1893-1947	−10				5	
	Birmingham CO	711	1893-1945	−10	12			10	8.0
	Birmingham AP	615	1939-1947	−10		21	7.4		
	Mobile CO	143	1872-1947	−1	22			15	9.9
	Mobile AP	219	1940-1947	13		30	8.5		
	Montgomery CO	293	1872-1947	−5	18			10	7.5
	Montgomery AP	226	1938-1944	9	19				
Ariz.	Flagstaff CO	6957	1899-1947	−25	−15			−10	7.7
	Kingman AP	3473	1935-1939[d]	8		22	8.6		
	Phoenix CO	1122	1895-1947	16	26			25	5.4
	Phoenix AP	1112	1937-1947	21		31	5.6		
	Tucson CO		Up to 1946	6				25	5.2
	Tucson AP	2561	1935-1939[d]	19		30	6.3		
	Winslow CO	4853	Up to 1946	−19	−1			−10	
	Winslow AP	4899	1932-1947	−18		6	7.5		
	Yuma CO	146	1876-1946[d]	22	27			30	6.7
Ark.	Fort Smith CO	545	1882-1945	−15	6			10	8.3
	Fort Smith AP	463	1945-1947	7					
	Little Rock CO	451	1879-1942	−12	10			5	8.3
	Little Rock AP	282	1942-1947	7		21	6.6		
Cal.	Bakersfield AP	499	1937-1946[d]	19	25	33	5.6	25	
	Burbank AP	740	1931-1947	21		35	4.9		
	Daggett AP	1925	1935-1939[d]			27	8.5		
	Eureka CO	115	1886-1947	20	29			30	7.3
	Fresno CO	387	1887-1939	17	26			25	5.4
	Fresno AP	281	1939-1947	23		32	4.5		
	Los Angeles CO	534	1877-1947	28	37			35	6.4
	Oakland AP	21	1929-1947	23		36	7.5	30	
	Red Bluff CO	341[e]	1877-1934	17					6.1
	Red Bluff AP	346	1944-1947	25					
	Redding AP	579	1935-1939[d]	17		32	7.1		
	Sacramento CO	116	1877-1947	17	28			30	7.2
	Sacramento AP	22	1938-1947	22					
	San Diego CO	90	1871-1940	25	37			35	6.3
	San Diego AP	34	1939-1947	34		43	5.8		
	San Francisco CO	164	1875-1947	27	37			35	7.5
	San Jose CO	100[e]	Up to 1946	22				25	
	Williams AP	124	1935-1939[d]			29	8.3		
Col.	Denver CO	5398	1871-1947	−29	−11			−10	7.5
	Denver AP	5379	1934-1947	−30		0	8.8		
	Grand Junction CO	4587[e]	1889-1945[d]	−21	−2			−15	4.4
	Pueblo CO	4770	1889-1938[d]	−27				−20	7.9
	Pueblo AP	4810	1939-1947	−26		2	6.4		
Conn.	Hartford CO	229	1905-1940	−18				0	8.7
	Hartford AP	20	1940-1947	−24		4	8.3		
	New Haven CO	180	1872-1946[d]	−15	−1			0	9.4
	New Haven AP	17	1943-1947	−4		11	8.1		
D of C.	Washington CO	128	1871-1947	−15	−1			0	7.8
	Washington AP	20	1935-1939[d]			14	7.4		
Fla.	Apalachicola CO	23	1922-1947	18				25	8.4
	Jacksonville CO	104	1871-1947	10	29			25	9.0
	Jacksonville AP	29	1938-1947	16		31	7.0		
	Key West CO	23	1871-1947	41				45	10.6
	Key West AP	48	1939-1947	42					
	Miami CO	253	1896-1947	27	38			35	10.1
	Miami AP	13	1940-1947	28	35	45	7.3		
	Pensacola CO	67	1879-1947	7	23			20	10.9
	Pensacola AP	113	1943-1947	20					
	Tampa CO	111	1890-1940[d]	19	32			30	8.6
	Tampa AP	12	1941-1946	31					
	Titusville AP	52	1935-1939[d]			38	6.5		
Ga.	Atlanta AP	1020	1878-1945[d]	−8	12	22	10.2	10	11.7
	Augusta CO	195	1871-1946[d]	3				10	6.5
	Augusta AP	424	1939-1947	10					
	Macon CO	408	1899-1947	7				15	6.7
	Macon AP	432	1939-1947	8					
	Savannah CO	115	1871-1945	8	22			20	9.5
	Savannah AP	56	1939-1947	16		29	7.7		
Idaho	Boise CO	2818	1864-1939[d]	−28				−10	9.1
	Boise AP	2849	1939-1947	−13	−1	5	4.5		
	Burley AP	4150	1935-1939[d]	−35		2	8.8		
	Idaho Falls AP	4744	1935-1939[d]	−18		−7	7.6		
	Lewiston CO	763	1900-1944	−23	1	[illegible]		5	4.1
	Pocatello CO	4522	1899-1947	−28	−12			−5	8.9
	Pocatello AP	4467	1938-1947	−23		6	7.2		
Ill.	Cairo CO	319	1872-1947	−16				0	9.8
	Chicago CO	601[e]	Up to 1946	−23	−8			−10	12.0
	Chicago AP	615	1935-1939[d]			−3	11.7		
	Moline AP	594	1935-1939[d]			−6	10.8	−10	
	Peoria AP	660	1935-1939[d]			6	10.7	−10	8.3

Blank spaces indicate data not yet completed.

TABLE 1. WINTER CLIMATIC CONDITIONS[a]—(CONTINUED)

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Design Dry-Bulb Temp. on TAC 97½% Basis[f] °F	Col. 8 Wind Vel. at Design Temp.[g] Mph	Col. 9 Design Dry-Bulb Temp. in Common Use[h] °F	Col. 10 Avg. Wind Vel.-Dec. Jan.-Feb.[i] Mph
Ill.	Springfield CO	603	1879-1947	−24	−7			−10	11.9
	Springfield AP	608	1930-1947	−19		−2	11.6		
Ind.	Evansville CO	464	1897-1940	−16	1				9.6
	Fort Wayne CO	885	1911-1941	−24				−10	10.4
	Helmer AP	970	1935-1939[d]			−1	11.5		
	Indianapolis CO	816	1871-1946[d]	−25	−6			−10	11.3
	Indianapolis AP	800	1932-1946[d]	−18		2	11.4		
	Terre Haute CO	1146	1893-1946[d]	−18	−5				10.2
	Terre Haute AP	589		−11					
Iowa	Charles City CO	1023	1891-1945[d]	−34	−22				7.9
	Davenport CO	648	1872-1947	−27	−13			−15	10.5
	Des Moines CO	805	1878-1945[d]	−30				−15	10.1
	Des Moines AP	979	1935-1939[d]			−8	14.5		
	Dubuque CO	740	1874-1947	−32	−17			−20	7.1
	Keokuk CO	637	1871-1945[d]	−27	−12				8.3
	Sioux City CO	1093[e]	1889-1944	−35	−20			−20	11.5
	Sioux City AP	1098	1940-1946[d]	−24					
Kansas	Concordia CO	1425	1885-1947	−25	−13			−10	7.7
	Dodge City CO	2515	1942-1947	−26	−10			−10	10.6
	Dodge City AP	2599	1874-1942	−26					
	Topeka CO	991	1887-1947	−25				−10	9.2
	Topeka AP	883	1946-1947	−21					
	Wichita CO	1497	1888-1939	−22	−4			−10	12.4
	Wichita AP	1423	1939-1947	−10		6	14.7		
Ky.	Louisville CO	563	1871-1947	−20	−5			0	9.8
	Louisville AP	544	1937-1947	−15		9	8.8		
La.	New Orleans CO	85	1874-1947	7	26			20	8.6
	New Orleans AP	8	1937-1947	19		36	12.8		
	Shreveport AP	179	1935-1939[d]		16	27	8.9	20	8 8
Maine	Eastport CO	100	1873-1947	−23	−15			−10	12.6
	Portland CO	70	1885-1940	−21	−6			−5	10.4
	Portland AP	66	1940-1947	−39					
Md.	Baltimore CO	114	1871-1947	−7	8			0	8.2
	Baltimore AP	43	1935-1939[d]			13	8.9		
Mass.	Boston CO	356	1870-1935[d]	−18	−3			0	12.4
	Boston AP	45	1936-1947	−14		8	12.3		
	Nantucket CO	45	1886-1947	−6				0	14.8
	Nantucket AP	48	1946-1947	12					
Mich.	Alpena CO	615	Up to 1946	−28	−12			−10	11.0
	Detroit CO	1000	1873-1933	−24	−11			−10	12.0
	Detroit AP	632	1935-1939[d]			4	11.0		
	Escanaba CO	645	1878-1945[d]	−32				−15	9.5
	Grand Rapids CO	706	1891-1946[d]	−24				−10	12.1
	Lansing CO	861	1910-1947	−25				−10	9.8
	Lansing AP	863	1940-1947	−10					
	Marquette CO	721	1874-1947	−27	−13			−10	10.6
	Sault St. Marie CO	724[e]	1888-1942[d]	−37	−22			−20	8.9
Minn.	Duluth CO	1133	1874-1947	−41	−28			−25	13.4
	Duluth AP	1413	1941-1947	−33					
	Minneapolis CO	945	1890-1947	−34	−23			−20	11.3
	Minneapolis AP	873	1938-1947	−31					
	St. Paul CO	951	1871-1933	−41	−25			−20	9.5
	St. Paul AP	708	1937-1947	−26	−18	−15	9.9		
Miss.	Meridian CO	410	1889-1947	−6	15			10	6.3
	Meridian AP	298	1939-1947	−7					
	Vicksburg CO	316	1874-1947	−1	18			10	8.3
	Vicksburg AP	266	1941-1947	10					
Mo.	Columbia CO	739	1889-1947	−26	•			−10	8.9
	Columbia AP	787	1939-1947	−18					
	Kansas City CO	741[e]	1889-1940	−22	−6			−10	10.3
	Kansas City AP	780	1935-1939[d]			2	10.6		
	St. Louis CO	646	1871-1947	−22	−2			0	11.8
	St. Louis AP	597	1930-1947	−19		3	10.8		
	Springfield AP	1270	1935-1939[d]		−5	8	11.0		11.0
Mont.	Billings AP	3584	1935-1939[d]		−30	−17	9.1	−25	12.4
	Butte CO	5700[e]	1894-1931	−34				−20	
	Butte AP	5538	1931-1947	−52		−18	4.8		
	Havre CO	2498	1880-1947	−57	−36			−30	9.4
	Helena CO	4175[e]	1880-1940[d]	−42	−24			−20	7.4
	Kalispell CO	3004	1897-1947	−34	−17			−20	5.2
	Miles City CO	2400	1892-1942[d]	−49	−30			−35	5.6
	Miles City AP	2629[e]	1935-1939[d]			−18	8.4		
Neb.	Lincoln CO	1189	1887-1947	−29	−13			−10	10.6
	Lincoln AP	1185	1933-1947	−26		−2	12.7		
	North Platte CO	2815	1874-1947	−35	−17			−20	7.9
	North Platte AP	2788	1935-1939[d]			−9	10.7		
	Omaha CO	1219	1873-1935	−32	−14			−10	9.7
	Omaha AP	1009	1935-Pres.	−21		−8	11.5		
	Valentine CO	2627	1889-1947	−38	−22			−25	9.2

[e] Approximate value.

TABLE 1. WINTER CLIMATIC CONDITIONS[a]—(CONTINUED)

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Lowest[d] Temp on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Design Dry-Bulb Temp. on TAC 97½% Basis[f] °F	Col. 8 Wind Vel. at Design Temp.[g] Mph	Col. 9 Design Dry-Bulb Temp. in Common Use[h] °F	Col. 10 Avg. Wind Vel.-Dec. Jan.-Feb.[i] Mph
Nev.	Elko.....AP	5079	1935-1939[d]			−4	4.0		
	Las Vegas.....AP	1882	1937-1947	8	16	23	5.3		
	Reno.....CO	4588	1905-1942	−19		0		−5	6.0
	Reno.....AP	4417	1940-1947	−16		7	3.6		
	Winnemucca.....CO	4293	1871-1947	−36	−10			−15	8.1
N. H.	Concord.....CO	343	1871-1941	−35	−15			−15	6.2
	Concord.....AP	359	1941-1947	−37					
N. J.	Atlantic City.....CO	45	1874-1947	−9	6			5	15.8
	Camden.....AP	20	1935-1939[d]			12	9.5		
	Newark.....AP	15	1931-1947	−14		10	11.6	0	
	Sandy Hook.....CO	19	1914-1938[d]	−11				0	16.1
	Trenton.....CO	144	1866-1946	−14	2			0	10.9
N. M.	Albuquerque.....CO	5022	1931-1933	5				0	7.3
	Albuquerque.....AP	5319	1933-1947	−6		16	7.1		
	El Morro.....AP	7120	1940-1947	−25	−19	−6	4.6		
	Rodeo.....AP	4116	1935-1939[d]			25	8.4		
	Roswell.....CO	3643	1905-1947	−29				−10	7.1
	Tucumcari.....AP	4054	1935-1939[d]			13	9.2		
N. Y.	Albany.....CO	114	1874-1947	−24	−11			−10	10.5
	Albany.....AP	280	1938-1947	−22		0	9.6		
	Binghamton.....CO	915	1891-1946	−28	−11			−10	6.8
	Binghamton.....AP	836	1942-1947	−17					
	Buffalo.....CO	693[c]	1873-1945[d]	−20	−4			−5	17.1
	Buffalo.....AP	726	1935-1939[d]			3	14.0		
	Canton.....CO	458	1889-1947	−43	−26			−25	10.5
	Elmira.....AP	948	1935-1939[d]			5	8.0		
	Ithaca.....CO	888	1879-1937	−24	−10			−15	11.3
	New York.....CO	425	1871-1947	−14	−3			0	16.8
	Oswego.....CO	363	1871-1943	−23	−9			−10	12.1
	Rochester.....CO	609	1872-1947	−22	−4			−5	9.6
	Rochester.....AP	560	1935-1939[d]	−16		4	11.9		
	Syracuse.....CO	465	1928-1940	−24				−10	11.2
	Syracuse.....AP	404	1940-1947	−26		−1	8.9		
N. C.	Asheville.....CO	2280	1902-1947	−6	6			0	9.5
	Charlotte.....CO	809	1878-1947	−5	12			10	7.3
	Charlotte.....AP	757	1939-1947	−3		22	7.5		
	Greensboro.....AP	896	1928-1947	−7		17	7.8	10	
	Raleigh.....CO	405	1944-1947	−2	13			10	7.9
	Raleigh.....AP	446	1935-1939[d]			20	8.5		
	Wilmington.....CO	78	1871-1947	5	18			15	9.4
N. D.	Bismarck.....CO	1675	1875-1940	−45	−31			−30	9.1
	Bismarck.....AP	1655	1940-1947	−38		−24	7.1		
	Devils Lake.....CO	1481	1904-1947	−46	−33			−30	10.1
	Dickinson.....AP	2599	1935-1939[d]			−20	12.4		
	Fargo.....AP	900	1935-1939[d]			−25	10.9	−25	
	Pembina.....AP	830	1935-1939[d]			−30	11.9		
	Williston.....CO	1919	1879-1947	−50				−35	8.6
Ohio	Akron.....CO		1887-1931[d]	−20				−5	
	Akron.....AP	104	1935-1939[d]			9	10.6		
	Cincinnati.....CO	772	1870-1947	−17	−2			0	8.5
	Cincinnati.....AP	488	1931-1947	−14		7	8.0		
	Cleveland.....CO	669	1871-1946[d]	−17	−2			0	14.7
	Cleveland.....AP	813	1930-1946[d]	−5		6	13.8		
	Columbus.....CO	812	1878-1946[d]	−20				−10	11.6
	Columbus.....AP	820	1939-1947	−15	−3	4	10.5		
	Dayton.....CO	1086	1883-1943	−28				0	11.1
	Dayton.....AP	1002	1940-1947	−11					
	Sandusky.....CO	608	1878-1946[d]	−16				0	11.0
	Toledo.....CO	668	1871-1947	−16	−5			−10	12.1
	Toledo.....AP	626	1940-1947	−13		4	12.1		
Okla.	Ardmore.....AP	762	1935-1939[d]			18	9.7		
	Oklahoma City.CO	1264	1890-1947	−17	2			0	11.5
	Oklahoma City.AP	1311	1939-1947	−10		14	14.7		
	Tulsa.....AP	686	1932-1947	−5		13	11.3	0	
	Waynoka.....AP	1529	1935-1939[d]			10	11.3		
Ore.	Arlington.....AP	881	1935-1939[d]			7	7.8		
	Baker.....CO	3501	1889-1947	−25	−17			−5	5.6
	Baker.....AP	3374	1939-1947	−19		3	6.4		
	Eugene.....CO	366	1890-1942[d]	−4				15	
	Eugene.....AP	368	1942-1947	9		23	5.3		
	Medford.....CO	1428[c]	1911-1929	−10				5	4.3
	Medford.....AP	1343	1929-1947	−3		23	4.3		
	Portland.....CO	98	1874-1947	−2	18			10	7.3
	Portland.....AP	25	1940-1947	3		22	8.0		
	Roseburg.....CO	523	1877-1947	−6	19			10	3.9
Pa.	Curwensville.....AP	2219	1943-1947	−10		0	13.5		
	Erie.....CO	771	1873-1946	−16	−3			−5	13.6
	Erie.....AP	736	1935-1939[d]			6	12.1		
	Harrisburg.....CO	335[c]	1888-1938[d]	−14	3			0	7.6
	Harrisburg.....AP	339	1935-1939[d]			7	9.0		

Table 1. Winter Climatic Conditions[a]—(Continued)

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Design Dry-Bulb Temp. on TAC 97½% Basis[f] °F	Col. 8 Wind Vel. at Design Temp.[g] Mph	Col. 9 Design Dry-Bulb Temp. in Common Use[h] °F	Col. 10 Avg. Wind Vel.-Dec. Jan.-Feb.[i] Mph
Pa.	Philadelphia CO	200	1871-1947	−11	6			0	11.0
	Philadelphia AP	18	1940-1947	1					
	Pittsburgh CO	929	1875-1947	−20	−2			0	11.6
	Pittsburgh AP	1284	1935-1947	−16		6	12.1		
	Reading CO	311	1913-1947	−14		9k		0	9.0
	Scranton CO	877	1901-1947	−19				−5	7.6
	Sunbury AP	448	1935-1939d			7	7.1		
R. I.	Block Island CO	46	1881-1947	−10				0	20.6
	Providence CO	77	1904-1947	−17	1			0	12.1
S. C.	Charleston CO	59	1871-1947	7	22			15	10.5
	Charleston AP	51	1940-1947	14		26	6.9		
	Columbia CO	401	1887-1947	−2	19			10	8.0
	Columbia AP	227	1939-1947	9					
	Greenville CO	1006e	Up to 1946	−5				10	8.4
S. D.	Huron CO	1342	1881-1938	−43	−26			−20	10.7
	Huron AP	1287	1938-1947	−30					
	Rapid City CO	3309	1888-1947	−34	−21			−20	8.0
	Rapid City AP	3220	1939-1947	−27					
Tenn.	Chattanooga CO	952	1879-1947	−10	9			10	7.7
	Chattanooga AP	675	1940-1947	6		19	6.2		
	Knoxville CO	1024	1871-1942	−16	2			0	7.2
	Knoxville AP	1007	1942-1947	1					
	Memphis CO	348	1872-1941	−9	9			0	9.3
	Memphis AP	267	1941-1947	1		19	8.9		
	Nashville CO	714	1871-1947	−13	5			0	9.8
	Nashville AP	610	1939-1947	−15		14	7.3		
Texas	Abilene CO	1748	1885-1944	−6				15	10.1
	Abilene AP	1756	1940-1947	−9		20	10.4	−5	
	Amarillo CO	3686	1892-1941	−16	0			−10	12.1
	Amarillo AP	3595	1941-1947	−7		11	12.9		
	Austin CO	625	1897-1942	−1				20	8.3
	Austin AP	625	1942-1947	13					
	Brownsville CO	140	1922-1943	12	29			30	10.4
	Brownsville AP	25	1943-1947	30					
	Corpus Christi CO	21	1887-1942d	11				20	11.0
	Corpus Christi AP	45	1943-1946d	23					
	Dallas CO	732	1913-1940	−3	13			0	10.6
	Dallas AP	520	1940-1947	5		23	8.8		
	Del Rio CO	1020	1905-1947	12				15	8.0
	El Paso CO	3792	1880-1942	−5	16			10	9.0
	El Paso AP	3956	1939-1947	11		26	8.6		
	Fort Worth CO	708	1898-1939d	−8	12			10	10.5
	Fort Worth AP	728	1940-1947	4					
	Galveston CO	128	1871-1947	8				20	11.2
	Galveston AP	9	1939-1947	14					
	Houston CO	198	1888-1947	5	22			20	10.5
	Houston AP	73	1932-1947	5		33	9.2		
	Palestine CO	555	1881-1947	−6				15	8.0
	Port Arthur CO	64	1917-1947	11				20	10.7
	Port Arthur AP	21	1944-1947	24					
	San Antonio CO	770e	1885-1941	4	21			20	8.3
	San Antonio AP	800	1942-1947	17		32	7.6		
	Waco AP	513	1931-1947	7		26	11.6		
	Wink AP	2811	1935-1939d			23	8.5		
Utah	Milford AP	5095	1935-1939d	−17		−2	7.7		
	Modena CO	5472	1901-1947	−32	−15			−15	9.0
	Salt Lake City CO	4346	1874-1947	−20	−2			−10	7.8
	Salt Lake City AP	4254	1928-1947	−30		7	7.4		
Vt.	Burlington CO	409	1884-1943	−29	−17			−10	11.6
	Burlington AP	335	1943-1944	−23					
Va.	Cape Henry CO	24	1874-1947	5				10	11.0
	Lynchburg CO	644	1874-1944	−7	8			5	8.1
	Lynchburg AP	951	1944-1947	7					
	Norfolk CO	91	1871-1947	2	15			15	12.1
	Richmond CO	180	1897-1947	−3	10			15	8.1
	Richmond AP	172	1929-1947	−12		15	7.1		
	Roanoke AP	1194	1935-1939d			21	8.2		
Wash.	Ellensburg AP	1731	1935-1939d			1	3.8		
	North Head CO	199	1884-1947	11	24			20	16.1
	Seattle CO	104	1890-1947	3	20			15	9.8
	Seattle AP	47	1928-1947	3		24	6.3		
	Spokane CO	2030	1881-1941	−30	−5			−15	6.2
	Spokane AP	1974	1941-1947	−7		4	5.1		
	Tacoma CO	279	1897-1947	7				15	8.0
	Tatoosh Is. CO	110	1883-1947	7				15	18.9
	Yakima CO	1160	1938-1946	−24				−5	4.1
	Yakima AP	1066	1944-1947	−4					
W. Va.	Elkins CO	1969e	1898-1944	−28	−8			−10	6.2
	Parkersburg CO	685	1888-1947	−27	−1			−10	7.2

TABLE 1. WINTER CLIMATIC CONDITIONS[a]—(CONCLUDED)

COL. 1 STATE	COL. 2 STATION[b]	COL. 3 ELEVATION[c] FT	COL. 4 PERIOD OF RECORD[d]	COL. 5 LOWEST TEMP. ON RECORD[d] °F	COL. 6 Average ANNUAL MIN. TEMP.[e] °F	COL. 7 DESIGN DRY-BULB TEMP. ON TAC 97½% BASIS[f] °F	COL. 8 WIND VEL. AT DESIGN TEMP.[g] MPH	COL. 9 DESIGN DRY-BULB TEMP. IN COMMON USE[h] °F	COL. 10 AVG. WIND VEL.-DEC. JAN.-FEB.[i] MPH
Wisc.	Green Bay . ..... CO	598	1886-1947	−36	−18			−20	10.5
	La Crosse ... . CO	725	1872-1947	−43	−21			−25	9.3
	La Crosse... .. AP	677	1943-1947	−28		−17	6.9		
	Madison.. ....CO	1008	1858-1947	−29				−15	10.1
	Madison . ..... .AP	884	1935-1939[d]			−8	9.1		
	Milwaukee .... CO	744	1870-1947	−25	−12			−15	12.1
	Milwaukee .... AP	707	1927-1947	−29		−6	11.9		
Wyo. ...	Cheyenne.... ......CO	6144	1873-1935	−38	−18			15	13.3
	Cheyenne . ... AP	6161	1935-1947	−34		−3	11.1		
	Lander . . .. CO	5448	1891-1916	−40	−12			−18	3.9
	Lander . .. .AP	5568	1946-1947	−14					
	Rock Springs.. ..AP	6746	1932-1942	−33		−7	9.1		
Alta . ..	Edmonton	2219[c]	Up to 1943	−57	−41			−40	7.5
B. C....	Vancouver .	22[c]	Up to 1943	2	13			10	4.5
	Victoria	228[c]	Up to 1943	−2	19			5	12.6
Man.. ..	Winnipeg	786[c]	Up to 1943	−54	−38			−35	10.1
N B ..	Fredericton	164[c]	Up to 1943	−35	−25			−20	9.1
N S.. ...	Yarmouth .. .	136[c]	Up to 1943	−12	0			−5	14.3
Ont. .	London . . .	912[c]	Up to 1943	−27	−14			−5	10.3
	Ottawa	294[c]	Up to 1943	−35	−24			−20	8.4
	Port Arthur	644[c]	Up to 1943	−40	−29			−30	8.0
	Toronto .	379[c]	Up to 1943	−26	−11			−10	13.6
P E. I. .	Charlottetown .	186[c]	Up to 1943	−27	−13			−10	9.8
Que. ..	Montreal .. .	187[c]	Up to 1943	−29	−18			−15	`11.3
	Quebec ..	296[c]	Up to 1943	−34	−23			−20	13.3
Sask.. ..	Prince Albert .. .	1414[c]	Up to 1943	−70	−47			−45	5.1
Y T .	Dawson	1062[c]	Up to 1943	−68	−54			−45	3.7
Newf.	St. Johns	428[c]	Up to 1943	−21	−5			−10	12

[a] United States Data compiled from U. S. Weather Bureau Records for years indicated, and Canadian data from Meteorological Service of Canada corrected to 1946.

[b] Col. 2. The stations followed by letters AP are airport stations, all others are city office stations and are followed by letters CO.

[c] Col. 3. The elevations marked c are ground elevations of the station. All other elevations given are the actual elevations of the thermometer bulb above mean sea level.

[d] Col. 4. The periods of record indicated apply only to the lowest temperature ever recorded shown in Col. 5, and generally extend from a summer month of the first year indicated through the spring months of the last year indicated. The periods marked by d terminated in December of the year indicated.

[e] Average of readings of one lowest temperature obtained for each year.

[f] It should be noted that Col. 7 applies only to airports, as these data for city stations are not available at this time. The temperature shown is the minimum hourly out-door temperature which has been equalled or exceeded 97½ per cent of the total hours in December, January and February for the period of record. It is pointed out that in most cases the airport stations are outside of the city and these data would apply primarily to rural areas.

[g] Col. 8 indicates the average wind velocity which occurred at temperatures the same as, and lower, than the temperatures shown in Col. 7.

[h] Col. 9 records design temperatures in use by A.S.H.V.E. Members as reported by Chapter Secretaries for the various stations. Where these were not available the design temperatures from an ACRMA publication and various other sources have been inserted.

[i] The wind velocities indicated in Col. 10 were furnished by the U. S. Weather Bureau and corrected through Feb. 1948.

[j] The bulletin published by A.S.H.V.E. for annual weather data of Detroit indicates 6 as design temperature.

[k] Computed for Reading by Karl Shelley and O. F. Smith.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures usually specified, vary in accordance with the use to which the building is to be put and Table 2 presents values which conform to good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 12. In other words, a person may feel warm or cool at the same dry-bulb temperature.

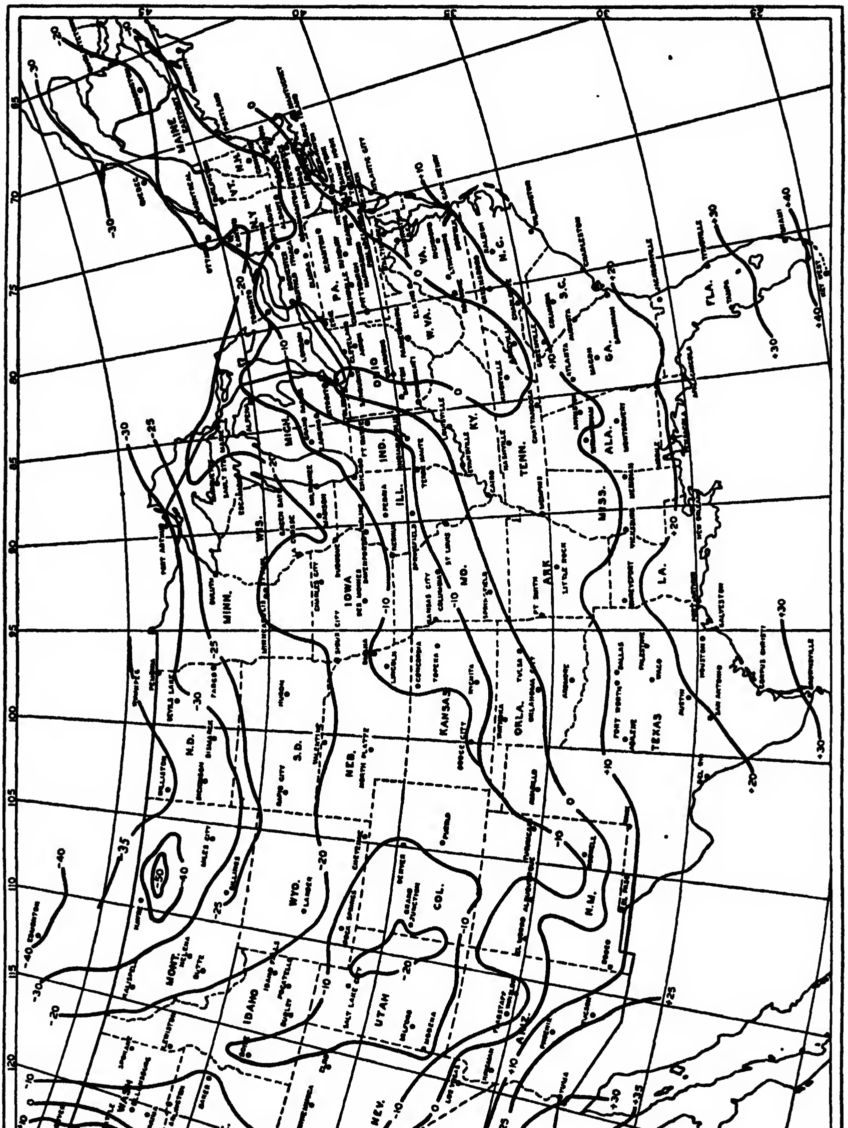


Fig. 1. Isotherms of Winter Outdoor Design Temperature

depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.

As explained in Chapter 12 for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However,

even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 2, consideration should be given to the actual relative humidity to be maintained, if provision is to be made for humidification. If no humidification is to be provided, the higher temperatures may not even produce comfort on cold days; if humidity is to be maintained at 50 per cent the lower temperatures will apply.

In rooms having large glass areas, when sun is not shining, or in rooms with walls having a high transmission coefficient, the lowered surface temperature will cause a feeling of coolness even though the air temperature in the room is at or above the temperatures indicated in the table. In rooms

TABLE 2. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg F	Type of Building	Deg F
SCHOOLS—		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–63
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a] The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature* (See Chapter 12.) When relative humidity is not controlled separately, optimum dry-bulb temperature for comfort will be slightly higher than shown in Table 2.

of this character, it is desirable to design for even higher temperatures than those listed, unless a counter-acting, high temperature surface is installed to offset the low temperature surfaces.

The inside temperatures specified in Table 2 will not necessarily apply to panel heated rooms due to the fact that the panels usually maintain a higher wall or glass surface temperature, thus producing comfort at lower air temperatures.

Temperature at Proper Level: In making the actual heat loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 2 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature

at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level.

Temperature at Ceiling: The air temperature at the ceiling is generally higher than at the breathing level due to stratification of air resulting from the tendency of the warmer or less dense air to rise. An allowance for this fact should be made in calculating ceiling heat losses, particularly in the case of high ceilings. However, the exact allowance to be made may be somewhat difficult to determine as it depends on many factors, including (1) the type of heating system, (2) ceiling height, and (3) the inside-outside temperature differential. The type of heating system is particularly important, as the temperature gradient from floor to breathing-level to ceiling may depend to a large extent on whether direct radiation,

Table 3. Approximate Temperature Differentials Between Breathing Level and Ceiling, Applicable to Certain Types of Heating Systems[a]

Ceiling Height (Ft)	Breathing Level Temperature (5 ft Above Floor)									
	60	65	70	72	74	76	78	80	85	90
10	3.0	3.3	3.5	3.6	3.7	3.8	3.9	4.0	4.3	4.5
11	3.6	3.9	4.2	4.3	4.4	4.6	4.7	4.8	5.1	5.4
12	4.2	4.6	4.9	5.0	5.2	5.3	5.5	5.6	6.0	6.3
13	4.8	5.2	5.6	5.8	5.9	6.1	6.2	6.4	6.8	7.2
14	5.4	5.9	6.3	6.5	6.7	6.8	7.0	7.2	7.7	8.1
15	6.0	6.5	7.0	7.2	7.4	7.6	7.8	8.0	8.5	9.0
16	6.1	6.6	7.1	7.3	7.5	7.7	7.9	8.1	8.6	9.1
17	6.2	6.7	7.2	7.4	7.6	7.8	8.0	8.2	8.7	9.2
18	6.3	6.8	7.3	7.5	7.7	7.9	8.1	8.3	8.8	9.3
19	6.4	6.9	7.4	7.6	7.8	8.0	8.2	8.4	8.9	9.4
20	6.5	7.0	7.5	7.7	7.9	8.1	8.3	8.5	9.0	9.5
25	7.0	7.5	8.0	8.2	8.4	8.6	8.8	9.0	9.5	10.0
30	7.5	8.0	8.5	8.7	8.9	9.1	9.3	9.5	10.0	10.5
35	8.0	8.5	9.0	9.2	9.4	9.6	9.8	10.0	10.5	11.0
40	8.5	9.0	9.5	9.7	9.9	10.1	10.3	10.5	11.0	11.5
45	9.0	9.5	10.0	10.2	10.4	10.6	10.8	11.0	11 5	12.0
50	9.5	10.0	10.5	10.7	10.9	11.1	11.3	11.5	12.0	12.5

[a] The figures in this table are based on an increase of 1 per cent per foot of height above the breathing level (5 ft) up to 15 ft and $\frac{1}{10}$ of one degree for each foot above 15 ft. This table is generally applicable to forced air types of heating systems. For direct radiation or gravity warm air, increase values 50 per cent to 100 per cent.

unit heaters or warm air is used, and in the latter case, whether the circulation is by gravity, auxiliary fan or forced air. Although with properly adjusted air flow the temperature differential with unit heaters can be reduced to a minimum, it is possible with improper adjustment that it may be increased over that which would normally result without mechanical circulation of the air.

It would be difficult from present available information to establish rules for determining the temperature difference to use in all cases However, for residences and other structures having ceiling heights under 10 ft, the comparatively small temperature differential between the breathing level and ceiling may generally be neglected without serious error. For higher ceilings where specific test data are not available, an allowance of approximately 1 per cent per foot of height above the breathing level may be made for ceiling heights up to 15 ft and approximately $\frac{1}{10}$ of 1 deg per foot of height above this level. The values in Table 3 are calculated

on this basis. For direct radiation and gravity warm air systems, the allowance should be increased from 50 per cent to 100 per cent over those given in Table 3. These rules should, however, be used with considerable discretion.

Temperature at Floor Level: According to the University of Illinois Research Residence tests[3], the temperature at the floor level ranged from about $2\frac{1}{2}$ to 6 deg below that at the breathing level, or somewhat greater than the difference between the breathing level and ceiling temperatures. Tests at the University of Wisconsin[4] indicated a somewhat smaller differential between the floor and breathing level temperatures. As a general rule, if the breathing level to ceiling temperature differential is neglected (as with ceiling heights under 10 ft), the breathing level-floor differential may also be neglected as the two are somewhat compensating, especially where both floor and ceiling heat losses are calculated for the same space. In other cases, the 10 ft temperature differentials in Table 3 may be used in arriving at the floor heat loss, these differentials to be subtracted from the breathing level temperature.

ATTIC TEMPERATURES

Frequently it is necessary to estimate the attic temperature, and in such cases Equation 1 can be used for this purpose:

$$t_a = \frac{A_cU_ct_i + t_o(A_rU_r + A_wU_w + A_gU_g)}{A_cU_c + A_rU_r + A_wU_w + A_gU_g} \tag{1}$$

where

t_a = attic temperature, Fahrenheit degrees.

t_i = inside temperature near top floor ceiling, Fahrenheit degrees.

t_o = outside temperature, Fahrenheit degrees.

A_c = area of ceiling, square feet.

A_r = area of roof, square feet.

A_w = area of net vertical attic wall surface, square feet.

A_g = area of attic glass, square feet.

U_c = coefficient of transmission of ceiling, based on surface conductance of 2.20 (upper surface, see Chapter 6). 2.20 = reciprocal of one-half the air space resistance.

U_r = coefficient of transmission of roof, based on surface conductance of 2.20 (lower surface, see Chapter 6).

U_w = coefficient of transmission of vertical wall surface.

U_g = coefficient of transmission of glass.

Example 1. Calculate the temperature in an unheated attic, assuming the following conditions: t_i = 70; t_o = 10; A_c = 1000; A_r = 1200; A_w = 100; A_g = 10; U_r = 0.50; U_c = 0.40; U_w = 0.30; U_g = 1.13.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1000 \times 0.40) + (1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

Equation 1 neglects the effect of any interchange of air such as would take place through attic vents or louvers intended to preclude attic con-

densation. However, according to tests[5], such venting of attics by means of small louvers or other small openings does not appreciably reduce the attic temperature and may be neglected without serious error. The attic temperature may be calculated in the usual manner by means of Equation 1, allowing the full value of the roof. The error resulting from this assumption will generally be considerably less than if the roof were neglected (as is sometimes the practice) and the attic temperature assumed to be the same as the outside temperature. When relatively large louvers are installed as is customary in the southern states, the attic temperature is often assumed as the average between inside and outside temperatures.

For a shorter, approximate method of calculating heat losses through attics, the combined ceiling and roof coefficient may be used as described in Chapter 6.

TEMPERATURES IN UNHEATED SPACES

The heat loss from heated rooms into unheated rooms or spaces must be based on the estimated or assumed temperature in such unheated spaces. This temperature will generally range between the inside and outside temperatures, depending on the relative areas of the surfaces adjacent to the heated room and those exposed to the outside. If the respective surface areas adjacent to the heated room and exposed to the outside are approximately the same, and if the coefficients of transmission are approximately equal, the temperature in the unheated space may be assumed to be the mean of the inside and outside design temperatures. If, however, the surface areas and coefficients are unequal, the temperature in the unheated space should be estimated by means of Equation 2.

$$t_u = \frac{t(A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.}) + t_o(A_aU_a + A_bU_b + A_cU_c + \text{etc.})}{A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.} + A_aU_a + A_bU_b + A_cU_c + \text{etc.}} \qquad (2)$$

where

t_u = temperature in unheated space, Fahrenheit degrees.

t = inside design temperature of heated room, Fahrenheit degrees.

t_o = outside design temperature, Fahrenheit degrees.

A_1, A_2, A_3, etc. = areas of surface of unheated space adjacent to heated space, square feet.

A_a, A_b, A_c, etc. = areas of surface of unheated space exposed to outside, square feet.

U_1, U_2, U_3, etc. = coefficients of transmission of surfaces of A_1, A_2, A_3, etc.

U_a, U_b, U_c, etc. = coefficients of transmission of surfaces A_a, A_b, A_c, etc.

Example 2. Calculate the temperature in an unheated space adjacent to a heated room having surface areas (A_1, A_2, and A_3) in contact therewith of 100, 120, and 140 sq ft and coefficients (U_1, U_2, and U_3) of 0.15, 0.20, and 0.25 respectively. The surface areas of the unheated space exposed to the outside (A_a and A_b) are respectively 100 and 140 sq ft and the corresponding coefficients are 0.10 and 0.30. The sixth surface is on the ground and is neglected in this example. Assume $t = 70$ and $t_o = -10$.

Solution. Substituting in Equation 2:

$$t_u = \frac{70[(100 \times 0.15) + (120 \times 0.20) + (140 \times 0.25)] + -10[(100 \times 0.10) + (140 \times 0.30)]}{(100 \times 0.15) + (120 \times 0.20) + (140 \times 0.25) + (100 \times 0.10) + (140 \times 0.30)}$$

$$t_u = \frac{4660}{126} = 37 \text{ F.}$$

The temperatures in unheated spaces having large glass areas and with two or more surfaces exposed to the outside (such as sleeping porches and sun parlors), are generally assumed to be the same as outside.

GROUND TEMPERATURES

Ground temperatures to be assumed for estimating basement heat losses will usually differ in the case of basement walls and floors, the temperatures under the floors being generally higher than those adjacent to walls.

Temperatures Adjacent to Basement Walls

Ground temperatures near the surface and under open spaces vary with the climate, the season of the year and the depth below the surface. The nearer the surface (during the cold weather) the lower the temperature. Frost will penetrate to a depth of over 4 ft in some localities if not protected by snow. A thick blanket of snow will result in a higher ground temperature near the surface. Consequently ground temperatures near the surface may be higher in cold climates where the snow remains on the ground for a greater length of time than in more moderate climates where the snow melts away periodically during the winter.

Complete data for various localities are not as yet available but in estimating heat losses through vertical walls below grade, it is advisable not to assume average ground temperatures above 32 F in northern climates when estimating heat losses from heated basements. This is for the mean height of the basement wall. Since the recommended wall coefficient for basement walls in contact with the soil is only 0.10, any small variation in the assumed ground temperature will not materially affect the calculated heat loss.

Temperatures Under Basement Floors

The temperature under basement floors[6] is influenced by the heat from the basement or protected from the influence of atmospheric conditions by the basement. In computing losses through basement floors the ground temperatures may be assumed the same as the approximate water temperatures at depths of 30 to 60 ft given in Fig. 3, Chapter 37. Test results indicate that the heat losses through basement floors are frequently over-estimated[7].

BASEMENT TEMPERATURES AND HEAT LOSS

The allowance to be made for basement heat loss depends on whether the basement is to be heated or not.

If the basement is *heated* and a specified temperature is to be maintained, the heat loss should be estimated in the usual manner, based on the proper wall and floor coefficients (see Chapter 6) and the outside air and ground temperatures. Heat loss through windows and walls above grade should be based on outside temperatures and the proper air-to-air coefficients. Heat loss through basement walls below grade should be based on the floor and wall coefficients for surfaces in contact with the soil and on the proper ground temperature.

If a basement is completely below grade and is *not heated*, the temperature in the basement will normally range between that in the rooms above and the ground temperature. Basement windows will of course lower the basement temperature when it is colder outside and any heat

given off by the heating plant will increase the basement temperature. In any case, the exact basement temperature is likely to be a somewhat indeterminate quantity, if the basement is not heated. Since the basement temperature will generally be lower than that of the rooms above, an allowance should theoretically be made for the loss from the rooms above through the floor over the basement.

The temperature in crawl spaces below floors will vary greatly depending on the number and size of wall vents, the quantity of heating pipes and the type of insulation. It is therefore necessary to analyze the conditions and select an appropriate temperature by judgment.

TRANSMISSION HEAT LOSS

The basic formula for the loss of heat by transmission through any surface is given in Equation 3:

$$H_t = AU\ (t - t_o) \tag{3}$$

where

H_t = heat loss transmitted through the wall, roof, ceiling, floor, or glass, Btu per hour.

A = area of wall, glass, roof, ceiling, floor, or other exposed surface, square feet.

U = coefficient of transmission, air to air, Btu per (hour) (square foot) (Fahrenheit degree temperature difference) (Chapter 6).

t = inside temperature near surface involved which may not necessarily be the so-called breathing line temperature, Fahrenheit degrees.

t_o = outside temperature, or temperature of adjacent unheated space or of the ground, Fahrenheit degrees.

Example 3. Calculate the transmission loss through an 8 in. brick wall having an area of 150 sq ft if the inside temperature (t) is 70 F and the outside temperature (t_o) is − 10 F.

Solution. The coefficient of transmission (U) of a plain 8 in. brick wall is 0.50 (Chapter 6, Table 7). The area (A) is 150 sq ft. Substituting in Equation 3:

$$H_t = 150 \times 0.50 \times [70 - (-10)] = 6000 \text{ Btu per hour.}$$

Transmission Loss Through Ceilings and Roofs

The transmission heat loss through top floor ceilings, attics, and roofs may be estimated by either of two methods:

1. By substituting in Equation 3 the ceiling area (A), the inside-outside temperature difference ($t - t_o$) and the proper value of (U):

   a. *Flat roofs.* Select the coefficient of transmission of the ceiling and roof from Tables 14 or 15, Chapter 6, or use appropriate coefficients in Equation 1 if side walls extend appreciably above the ceiling of the floor below.

   b. *Pitched roofs.* Select the combined roof and ceiling coefficient from Table 17, Chapter 6 or calculate the combined roof and ceiling coefficient by means of Equation 5, Chapter 6, where this formula is applicable as explained in Chapter 6.

2. By estimating the attic temperature (based on the inside and outside design temperatures) by means of Equation 1, and substituting for t_o in Equation 3, the value of t_a thus obtained, together with the ceiling area (A) and the ceiling coefficient (U). This applies to *pitched roofs*. In the case of *flat roofs* it is not necessary to

calculate the attic temperatures as the ceiling-roof heat loss can be determined as per paragraph 1a.

INFILTRATION HEAT LOSS

The infiltration heat loss includes (1) the *sensible heat loss* or the heat required to warm the outside air entering by infiltration and (2) the *latent heat loss* or the heat equivalent of any moisture which must be added.

Sensible Heat Loss

The formula for the heat required to warm the outside air which enters a room by infiltration to the temperature of the room, is given in Equation 4:

$$H_s = 0.240\, Qd\, (t - t_o) \tag{4}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t, Btu per hour.

0.240 = specific heat of air.

Q = volume of outside air entering building, cubic feet per hour (see Chapter 8).

d = density of air at temperature t_o, pounds per cubic foot.

It is sufficiently accurate to use $d = 0.075$ in which case Equation 4 reduces to

$$H_s = 0.018\, Q\, (t - t_o) \tag{4a}$$

The volume of outside air entering per hour (Q) depends on the wind velocity and direction, the width of crack or size of openings, the type of openings and other factors, as explained in Chapter 8. Where the crack method is used for estimating leakage, it is more convenient to express the air leakage heat loss in terms of the crack length:

$$H_s = B\, L\, (t - t_o) \tag{4b}$$

where

B = air leakage per (hour) (foot of crack) (Chapter 8) for the wind velocity and type of windows or door crack involved multiplied by 0.018.

L = length of window or door crack to be taken into consideration, feet.

Example 4. What is the infiltration heat loss per hour through the crack of a 3 x 5 ft average, double-hung, non-weatherstripped, wood window, based on a wind velocity of 15 mph? Assume inside and outside temperatures to be 70 F and zero respectively.

Solution. According to Table 2, Chapter 8, the air leakage through a window of this type (based on $\frac{1}{16}$ in. crack and $\frac{3}{64}$ in. clearance) is 39 cu ft per foot of crack per hour. Therefore, $B = 39 \times 0.018 = 0.70$. The length of crack ($L$) is $(2 \times 5) + (3 \times 3)$, or 19 ft; $t = 70$ and $t_o = 0$. Substituting in Equation 4b,

$$H_s = 0.70 \times 19 \times (70 - 0) = 931 \text{ Btu per hour.}$$

Number of Air Changes to be Used for Computations

Since a certain amount of judgment regarding quality of construction, weather conditions, use of room and other factors is required in estimating infiltration by any method, some designers base infiltration upon an estimated number of air changes rather than upon the length of window cracks.

Table 4 of Chapter 8 indicates air changes commonly used but should be taken only as a guide. For further discussion of the method see section on Air Change Method in Chapter 8.

Crack Length to be Used for Computations

For designers who prefer to use the crack method the basis of calculation is as follows: The amount of crack used for computing the infiltration heat loss should not be less than half of the total crack in the outside walls of the room. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building. In a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack.

The total infiltration loss of a building having partitions will not be equal to the sum of the infiltration losses of the various rooms, since at any given time infiltration will take place only on the windward side or sides and not on the leeward side. Therefore, if a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Qd\left(\frac{m_i - m_o}{7000}\right) h_{fg} \tag{5}$$

where

H_l = heat required to increase moisture content of air leaking into building from m_o to m_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
m_i = vapor density of inside air, grains per pound of dry air.
m_o = vapor density of outside air, grains per pound of dry air.
h_{fg} = latent heat of vapor at m_i, Btu per pound.

If the latent heat of vapor (h_{fg}) is assumed to be 1060 Btu per pound, Equation 5 reduces to

$$H_l = 0.0114\, Q\, (m_i - m_o) \tag{5a}$$

Equations 4a, 4b and 5a may also be used for determining the sensible and latent heat gains due to infiltration in cooling load computations.

SELECTION OF WIND VELOCITIES

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.

2. Wind movement materially increases the infiltration of cold air through the cracks around doors and windows, and even through the building materials themselves.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to compute the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified.

Therefore, since Table 1 lists the average velocity of winds occurring at temperatures equalled or exceeded 97½ per cent of the winter period for each locality, this value should be the basis for estimating infiltration losses. When using the air change method it will not be necessary to consider the wind velocities. Designers employing the crack method generally use values corresponding to a 15-mile wind. Due to the small effect of the wind velocity on the transmission coefficient, the values in Chapter 6, based on a 15-mile wind may be used at all times.

Exposure Factors

Many designers use empirical exposure factors to increase the calculated heat loss of rooms or spaces on the side or sides of the building exposed to the prevailing winds. However, according to a survey made in 1943, many GUIDE users have found that the use of exposure factors is not necessary as the GUIDE method of calculating heat losses provides an ample heat loss allowance. Therefore exposure factors may be regarded as *factors of safety* for the rooms or spaces exposed to the prevailing winds, to allow for additional capacity for these rooms or spaces, or to *balance the radiation*, particularly in the case of multi-story buildings. Although the exposure allowance is frequently assumed to be 15 per cent, the actual allowance to be made, if any, must to a large extent be a matter of experience and judgment of the designer, since there are at present no authentic test data available from which rules could be developed for the many conditions encountered in practice.

As stated previously, the value of U in the tables of Chapter 6 is based on a wind velocity of 15 mph and the surface resistance for this wind velocity (0.17) is sufficiently low so that higher wind velocities will decrease the surface resistance to a negligible degree and therefore have only a slight effect on the average over-all coefficient. On the other hand, infiltration losses vary almost directly as the wind velocity, as will be apparent from the factors in Table 2 of Chapter 8. The more exact method therefore would be to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities for the infiltration losses on the various sides.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only

Table 5. Heat Loss Calculation Sheet for Uninsulated Residence (Fig. 2)

A	B	C	D	E	F	G
Room or Space	Part of Structure or Infiltration Air Changes	Net Area or Air Volume	Coefficient	Temp. Diff.[a]	Heat Loss (Btu per hour)	Totals (Btu per hour)
Bedroom A and Closet	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Ceiling	252 sq ft	0.69	39.8[d]	6910	
	Infiltration (¾)[g]	1510 cfh[b]	0.018[c]	80	2180	15,860
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Ceiling	170 sq ft	0.69	39.8[d]	4660	
	Infiltration (¾)[g]	1020 cfh[b]	0.018	80	1470	11,060
Bedroom C and Closet	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Ceiling	129 sq ft	0.69	39.8[d]	3540	
	Infiltration (¾)[g]	874 cfh[b]	0.018[c]	80	1260	8,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0.45	80	720	
	Ceiling	110 sq ft	0.69	39.8[d]	3020	
	Floor over garage	110 sq ft	0.25	35[e]	960[p]	
	Infiltration (¾)[g]	660 cfh[b]	0.018[c]	80	950	8,300
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0.45	80	500	
	Ceiling	55 sq ft	0.69	39.8[d]	1510	
	Infiltration (1)[g]	440 cfh[b]	0.018[c]	80	630	3,310
Bathroom 2	Walls	79 sq ft	0.26	80	1640	
	Glass	9 sq ft	0.45	80	320	
	Ceiling	35 sq ft	0.69	39.8[d]	960	
	Floor over garage	35 sq ft	0.25	35[e]	310[p]	
	Infiltration (1)[g]	280 cfh[b]	0.018[c]	80	400	3,630
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[f]	35[e]	1280[p]	
	Glass	50 sq ft	0.45	80	1800	
	Floor	291 sq ft				
	Infiltration (1½)[h]	3745 cfh[b]	0.018	80	5400	14,460
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	1.13	80	3160	
	Glass (windows)	20 sq ft	0.45	80	720	
	Floor	168 sq ft				
	Infiltration (1½)[i]	2140 cfh[b]	0.018[c]	80	3080	10,680
Kitchen and Entrance to Garage	Walls	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[f]	35[e]	700[p]	
	Glass	18 sq ft	0.45	80	650	
	Door	17 sq ft	0.51	35	300	
	Floor	125 sq ft				
	Infiltration (1½)[j]	1595 cfh[b]	0.018[c]	80	2300	6,100
Lavette and Vestibule	Walls	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[f]	35[e]	1160[p]	
	Glass	9 sq ft	0.45	80	320	
	Door	19 sq ft	0.51	80	780	
	Floor	30 sq ft				
	Infiltration (1½)[k]	383 cfh[b]	0.018[c]	80	550	4,650
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Ceiling[r]	87 sq ft	0.69	39.8[d]	2390	
	Infiltration (2)[l]	1110 cfh[b]	0.018[c]	80	1600	5,500
Garage	Walls	167 sq ft	0.28	45[e]	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	41 sq ft	0.51	45	1010	
	Infiltration (1½)[m]	2360 cfh[b]	0.018[c]	45	1910	
	Floor (heat gain)	185 sq ft	0.10	−15	−280[o]	
	Gain adjoining rooms				−4410[p]	3,040
Recreation Room[q]	Walls	220 sq ft	0.10	38	810	
	Glass	8 sq ft	1.13	80	720	
	Floor	287 sq ft	0.10	20	570	
	Infiltration (1)[n]	2010 cfh[b]	0.018[c]	80	2890	5,020
					TOTAL	99,940

Table 6. Summary of Heat Losses of Uninsulated Residence (*Btu Per Hour*)

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door[a]	Infiltration	Totals
Bedroom A	5330	6910	....	1410	2180	15.860
Bedroom B	3190	4660	....	1440	1170	11,060
Bedroom C	2560	3540	....	970	1260	8.330
Bedroom D	2650	3020	960	720	950	8.300
Bathroom 1	670	1510	....	500	630	3,310
Bathroom 2	1610	960	310	320	400	3.630
Living Room	7260	....	....	1800	5400	14,460
Dining Room	3720	....	....	3880	3080	10,680
Kitchen	2850	....	....	950	2300	6.100
Lavette	3000	....	....	1100	550	4,650
Entrance Hall	870	2390	....	640	1600	5.500
Garage	−1030[a]	−1270[b]	−280[c]	3710	1910	3,010
Recreation	840	....	570	720	2890	5.020
Design Totals	33,850	21.720	1.560	18.190	24,620	99.940
Operating Totals[d]	33.850	21,720	1,560	18,190	12,310	87,630
Percentages[e]	38.6	24.8	1.8	20.7	14.1	100.0

[a] Wall heat loss of 2110 Btuh minus wall heat gains of 1280, 700 and 1160 Btuh. [b] Heat gains of 960,310 Btuh. [c] Heat gain. [d] Based on ½ computed infiltration. [e] Based on Operating Totals.

wind velocity. Inside temperature from Table 2 is assumed to be 70 F. The attic is unheated. Assume ground temperature to be 50 F (see Fig. 3, Chapter 37) under basement and garage floors and 32 F adjoining basement walls. Estimate infiltration losses by the air change method. No wall, ceiling or roof insulation is to be considered in this problem, but all first and second floor windows, except in the garage, are to have storm sash. The building is constructed as follows (heat transmission coefficients U are in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0.28). Walls of dormer over garage, same except wood siding in place of brick veneer (0.26).

Attic Walls: Brick veneer, building paper, wood sheathing on studding (0.42).

Basement Walls: 10 in. concrete (0.10).

Roof: Asphalt shingles on wood sheathing on rafters (0.53).

Ceiling (Second floor): Metal lath and plaster (0.69).

Windows: Double-hung wood windows with storm sash (0.45). Steel casement sash in basement (1.13).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

Floor (Basement and Garage): 4 in. stone concrete on 3 in. cinder concrete (0.10).

Solution: The calculations for this problem are given in Table 5, and a summary of the results in Table 6. The values in column F of Table 5 were obtained by multiplying together the figures in columns C, D, and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 5 for further explanation of data.

Attention is called to the summary of heat losses (Table 6) for the uninsulated residence. As storm windows are used in this instance the *glass* and *door* transmission heat losses of 20.7 per cent are relatively small. The infiltration losses of 14.1 per cent are also comparatively small because the storm windows are equivalent to weatherstripping. In this problem, the wall, ceiling and floor transmission losses comprise 65.2 per cent of the total.

Example 6. Calculate the heat loss of residence shown in Fig. 2 based on the same conditions as in *Example 5* but having construction improved or insulated to obtain coefficients as follows:

Walls, 0.13; *Walls of Dormer over Garage*, 0.12; *Attic Walls*, 0.28; *Walls Adjoining Garage*, 0.18; *Basement Walls (Recreation Room)*, 0.10.

Roof, 0.53.

Ceiling (Second Floor), 0.15.

Windows (Same as in *Example 5*).

Floor (Bedroom D), 0.18.

Solution: The procedure for calculating the heat losses is similar to that for *Example 5*. A summary of the results is given in Table 7.

TABLE 1. TYPICAL COMMERCIAL DESIGN ROOM CONDITIONS FOR SUMMER AVERAGE PEAK LOAD IN COMFORT AIR CONDITIONING[a]

TYPE OF INSTALLATION	DRY-BULB TEMP	WET-BULB TEMP[b]	RELATIVE HUMIDITY PER CENT	GRAINS PER LB[b]	EFFECTIVE TEMP[c]
Deluxe Application	78	65	50	72.7	72.2
Normal Application	80	67	51	78.5	74.0
15 to 40 min Occupancy	82	68	49	80.0	75.3

[a] Values in Table 1 are for *peak load* conditions. It is general practice to operate a system at approximately 76 F and 50 per cent relative humidity at other than peak load.

[b] Psychrometric data for standard barometric pressure.

[c] Fig. 10, Chapter 12, air movement 15 to 25 fpm.

sky radiation and from outdoor-indoor temperature differential for glass areas and exterior, walls and roofs, modified by periodic heat flow or lag factors depending on the type of structure. (2) Load due to heat gain through interior partitions, ceilings and floors. (3) Load due to ventilation either natural or mechanical. (4) Load due to heat sources within the conditioned space such as people, lights, power equipment and appliances. (5) Load due to moisture transfer through permeable building materials. (6) Miscellaneous heat sources.

C. *Determination of Air Quantity and Apparatus Dew-Point.*

These factors will be discussed in turn. The material presented leads to an illustrative procedure for a cooling-load calculation, and a numerical example is given to demonstrate the calculations involved.

DESIGN CONDITIONS

Indoor Conditions

Indoor air conditions for human health and comfort have been and continue to be the subject of much discussion and research.

The effective temperature index, explained in Chapter 12 is probably the best available source of design criteria for comfort air conditioning systems for buildings or enclosures in which the air and inside surface temperatures remain substantially equal; a condition that can safely be assumed for most ordinary comfort air conditioning installations. Other sources of design specifications are to be found in the requirements of codes and ordinances, and in the varied long-term experiences of manufacturers, contractors, and engineering specialists.

Past experience, cumulative over many years, indicates that indoor design conditions for which summer air-conditioning equipment is selected, should not exceed a temperature of 80 F or a relative humidity of 50 per cent for the average job in the United States. If these conditions are exceeded, complaints of discomfort may be expected, especially with continuous occupancy. For very brief occupancy only, a slightly higher peak-load design temperature may be employed. In regard to the lower limit of humidity, complaints are not encountered for store installations operated down to 35 per cent relative humidity or, for office jobs, somewhat lower. These observations apply to normal commercial practice in this country only; for extremes such as tropical or very hot regions it is regarded as more practicable to design for a peak-load outdoor-indoor temperature difference of about 15 to 20 F.

Table 1 offers typical design conditions for average requirements encountered. The *deluxe* figures would also apply in general for localities having a summer outdoor design temperature of 90 F or less; and the *15 to 40 min* occupancy values or even somewhat higher dry-bulb tem-

TABLE 2. ILLUSTRATIVE TEMPERATURES AND RELATIVE HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING[a]

Classification	Materials, Location or Process	Temperature F	Relative Humidity %
Employe Efficiency	General Machine Shop Work	78-80	50
	Drafting Rooms	78-80	50
	Offices	73-80	50
Storage Prior to Manufacturing	Rough Castings	80	50
	Ceramic Materials	60-80	50
	Pharmaceutical Powders	70-80	30-35
	Sugar	80	35
	Paper	75-80	35
	Electrical Goods	60-80	35-50
	Flour	60-75	55-65
	Rubber	60-75	40-50
	Grains	60	30-45
	Hardened Aluminum Alloys	0 to −30	
Manufacturing Process	Machine Tool Oil Cooling	70-90	
	Precision Parts Honing Machinery	75-80	40-55
	Ceramic Molding	80	60
	Manufacturing of Electrical Wiring	60-80	35-50
	Assembly Line	65-80	40-50
	Gage Rooms	78	50
	Instrument Calibration	68	50-55
	Match Manufacturing	72-74	50
Research and Development	Paper Testing Laboratory	60-80	55-65
	Textile Testing Laboratory	70	65
	Special Process Temperature Boxes	−100 to +170	
	Chemical Laboratories	78	50
	Fibres and Plastics	70-75	50-65
	Drafting	78-80	45-50
	Temperature Shock Tests	−80 to +150	

[a] Taken from the article, "Indoor Climate and Refrigeration for Post-War Industry," by E. K. Heglin *Cleveland Engineering*, Vol. 40, No. 27, July 3, 1947, p. 5.

peratures would indicate acceptable conditions for very hot localities. Table 1 *is to be used with good judgment*, for there is no universal rule which may be applied to indoor design conditions.

Guarantees of conditions to be maintained for summer operation are based upon a definite set of load conditions. At other than the guarantee load, the conditions produced by a system are determined by the balance of imposed load and equipment capacity and by the method adopted for regulating the system operation. Complete specifications of indoor design conditions would include part-load and overload operation, particularly from the viewpoint of economy.

In the field of *industrial air conditioning*, indoor design conditions are established by the requirements of goods and processes, in addition to the comfort and efficiency of the workers. No generally-applicable specifications are possible, as each job has its own special requirements. Table 2 offers illustrative general information.

The indoor design conditions suggested have had reference to conditions to be maintained at the level of occupancy. For extremely high ceilings in public or industrial buildings, only the zone from 10 to 15 ft above the floor may be cooled to the full extent. The air temperature at the ceiling would be much higher, and this should be kept in mind when calculating the convective portion of the roof heat gain. A reduction of outdoor-to-indoor air temperature differential may be assumed in such instances; radiation from the inner roof surface is not diminished.

Outdoor Conditions

Summer climatic conditions and suggested design wet-bulb and dry-bulb temperatures are given in Table 3 for various locations in the United

Table 3. Summer Climatic Conditions[a] (Continued)
Suggested Design Wet-Bulb and Dry-Bulb Temperatures

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] °F	Col. 6 Design Dry-Bulb Temp. on T.A.C 2½% Basis[e] °F	Col. 7 Design Dry-Bulb Temp. in Common Use[f] °F	Col. 8 Design Wet-Bulb Temp. in Common Use[f] °F	Col. 9 Average Summer Wind Velocity[g] Mph
Ill.	Peoria AP	660	1935–1939	111	94	96	76	8.2
	Springfield CO	603	1879–1947	110	....	98	77	....
	Springfield AP	608	1930–1947	109	96	....	....	....
Ind.	Evansville CO	464	1897–1940	108	....	95	78	7.0
	Fort Wayne CO	885	1911–1941[d]	106	....	95	75	....
	Helmer AP	970	1935–1939	107	89	....	....	....
	Indianapolis CO	816	1871–1946	106	....	95	76	8.9
	Indianapolis AP	800	1932–1946	107	91	....	....	....
	Terre Haute CO	1146	1893–1946	110	....	95	78	....
	Terre Haute AP	589	1941–1946	103	....	....	....	....
Iowa	Davenport CO	648	1872–1947	111	....	95	78	....
	Des Moines AP	979	1935–1939	111	95	95	78	8.6
	Dubuque CO	740	1874–1947	110	....	95	78	....
	Keokuk CO	637	1872–1946	113	....	95	78	....
	Sioux City CO	1093[e]	1889–1944[d]	111	....	95	78	....
	Sioux City AP	1098	1940–1946	108	....	....	....	....
Kans	Concordia CO	1425	1885–1947	116	....	95	78	....
	Dodge City CO	2515	1874–1942[d]	109	....	95	78	....
	Dodge City AP	2599	1942–1947	109	....	....	....	....
	Topeka CO	991	1887–1947	114	....	100	78	....
	Topeka AP	883	1946–1947	108	....	....	....	....
	Wichita CO	1497	1888–1939	114	....	100	75	11.8
	Wichita AP	1423	1939–1947	109	100	....	....	....
Ky.	Louisville CO	563	1871–1947	107	....	95	78	7.2
	Louisville AP	544	1937–1947	103	93	....	....	....
La.	New Orleans CO	85	1874–1947	102	....	95	80	6.9
	New Orleans AP	8	1937–1947	100	93	....	....	....
	Shreveport AP	179	1935–1939	109	98	100	78	7.0
Maine	Eastport CO	100	1873–1947	93	....	90	70	....
	Portland CO	185	1885–1910	103	....	90	73	8.7
	Portland AP	65	1940–1947	99	....	....	....	....
Md.	Baltimore CO	114	1871–1947	107	....	95	78	7.4
	Baltimore AP	43	1935–1939	105	91	....	....	....
Mass.	Boston CO	356	1870–1935	104	....	92	75	12.5
	Boston AP	45	1936–1947	101	87	....	....	....
	Nantucket CO	45	1886–1947	92	....	95	75	....
	Nantucket AP	48	1946–1947	82	....	....	....	....
Mich.	Alpena CO	615	1874–1946	104	....	95	75	....
	Detroit CO	1000	1873–1933	104	....	95	75	9.5
	Detroit AP	632	1934–1947	105	89	....	73[h]	....
	Lansing CO	861	1910–1947	102	....	95	75	....
	Lansing AP	863	1940–1947	98	....	....	....	....
	Marquette CO	721	1874–1947	108	....	93	73	....
Minn.	Duluth CO	1133	1874–1947	106	....	93	73	....
	Duluth AP	1413	1941–1947	95	....	....	....	....
	Minneapolis CO	945	1890–1947	108	....	95	75	10.2
	Minneapolis AP	873	1938–1947	104	....	....	....	....
	St. Paul CO	951	1871–1933	104	....	95	75	....
	St. Paul AP	708	1937–1947	104	91	....	....	....
Miss.	Meridian CO	410	1889–1947	105	....	95	79	4.6
	Meridian AP	298	1939–1947	105	....	....	....	....
	Vicksburg CO	316	1874–1947	104	....	95	78	6.1
	Vicksburg AP	266	1941–1947	104	....	....	....	....
Mo.	Columbia CO	739	1889–1947	111	....	100	78	....
	Columbia AP	787	1939–1947	102	....	....	....	....
	Kansas City AP	780	1935–1939	112	101	100	76	9.1
	St. Louis CO	646	1871–1947	110	....	95	78	9.5
	St. Louis AP	597	1930–1947	111	97	....	....	....
	Springfield AP	1270	1935–1939	105	96	....	....	8.7
Mont.	Billings AP	3584	1935–1947	106	92	90	66	....
	Butte AP	5538	1931–1947	100	85	....	....	....
	Havre CO	2498	1880–1947	108	....	95	70	....
	Helena CO	4175	1880–1940	103	....	95	67	8.1
	Kalispell CO	3004	1897–1947	101	....	95	65	....
	Miles City AP	2629	1935–1939	108	97	....	....	....
Nebr.	Lincoln CO	1189	1887–1947	115	....	95	78	9.7
	Lincoln AP	1185	1933–1947	115	101	....	....	....
	North Platte CO	2815	1874–1947	109	....	95	78	8.1
	North Platte AP	2788	1935–1939	109	98	....	....	....
	Omaha CO	1219	1873–1935[d]	111	....	95	78	....
	Omaha AP	1009	1935–1947	114	98	....	....	....
	Valentine CO	2627	1889–1947	110	....	95	78	....
Nev.	Elko AP	5079	1935–1939	102	92	....	....	....
	Las Vegas AP	1882	1937–1947	117	108	....	....	....
	Reno CO	4588	1905–1912	106	....	95	65	7.2
	Reno AP	4417	1940–1947	105	93	....	....	....
	Winnemucca CO	4293	1871–1947	108	....	95	65	....
N. H.	Concord CO	343	1871–1941[d]	102	....	90	73	4.9
	Concord AP	359	1941–1947	99	....	....	....	....

Table 3. Summer Climatic Conditions[a] (Continued)
Suggested Design Wet-Bulb and Dry-Bulb Temperatures

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] °F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[e] °F	Col. 7 Design Dry-Bulb Temp. in Common Use[f] °F	Col. 8 Design Wet-Bulb Temp. in Common Use[f] °F	Col. 9 Average Summer Wind Velocity[g] Mph
N. J.	Atlantic City CO	45	1874-1947	104	......	95	78	......
	Camden AP	20	1935-1939	105	91	......	......	......
	Newark AP	15	1931-1947	104	89	95	75	......
	Trenton CO	144	1866-1946	106	......	95	78	8.8
N. M.	Albuquerque CO	5022	1931-1933[d]	99	......	95	70	7.8
	Albuquerque AP	5319	1933-1947	101	93	......	......	......
	El Morro AP	7120	1935-1939	92	84	......	......	......
	Rodeo AP	4116	1935-1939	104	97	......	......	......
	Roswell CO	3643	1905-1947[d]	107	......	95	70	......
	Tucumcari AP	4054	1935-1939	107	97	......	......	......
N. Y.	Albany CO	114	1874-1947	104	......	93	75	7.5
	Albany AP	280	1938-1947	99	88	......	......	......
	Binghamton CO	915	1891-1946	103	......	95	75	......
	Binghamton AP	836	1942-1947	97	......	......	......	......
	Buffalo AP	726	1935-1939	95	86	93	73	12.1
	Canton CO	438	1906-1947	99	......	90	73	8.2
	Elmira AP	948	1935-1939	96	88	......	......	......
	New York CO	425	1871-1947	102	......	95	75	12.5
	Oswego CO	363	1871-1947	100	......	93	73	......
	Rochester CO	609	1872-1947	102	......	95	75	......
	Rochester AP	560	1935-1939	98	89	......	......	......
	Syracuse CO	467	1902-1940	102	......	93	75	......
	Syracuse AP	404	1940-1947	97	88	......	......	......
N. C.	Ashville CO	2250	1902-1947	99	......	93	75	5.6
	Charlotte CO	809	1878-1947	103	......	95	78	......
	Charlotte AP	757	1939-1947	103	93	......	......	......
	Greensboro AP	896	1928-1947	101	91	95	78	......
	Raleigh CO	405	1887-1947	104	......	95	78	6.3
	Raleigh AP	446	1944-1947	102	93	......	......	......
	Wilmington CO	78	1871-1947	103	......	95	78	8.4
N. D.	Bismarck CO	1675	1875-1940	114	......	95	73	9.5
	Bismarck AP	1655	1940-1947	109	96	......	......	......
	Devils Lake CO	1481	1904-1947	112	......	95	70	......
	Dickinson AP	2599	1935-1939	112	94	......	......	......
	Fargo AP	900	1935-1939	115	93	95	75	......
	Pembina AP	830	1935-1939	109	92	......	......	......
	Williston CO	1919	1879-1947	110	......	95	73	......
Ohio	Akron AP	104	1935-1939	101	88	95	75	......
	Cincinnati CO	772	1870-1947	108	......	95	78	5.6
	Cincinnati AP	488	1931-1947	108	94	......	......	......
	Cleveland CO	669	1871-1946	100	......	95	75	11.1
	Cleveland AP	813	1930-1946	107	90	......	......	......
	Columbus CO	812	1878-1946	106	......	95	76	......
	Columbus AP	820	1939-1947	100	90	......	......	......
	Dayton CO	1086	1883-1943[d]	108	......	95	78	......
	Dayton AP	1002	1940-1947	99	......	......	......	......
	Sandusky CO	608	1878-1946	105	......	95	75	......
	Toledo CO	668	1871-1947	105	......	95	75	......
	Toledo AP	626	1940-1947	100	91	......	......	......
Okla.	Ardmore AP	762	1935-1939	110	99	......	......	......
	Oklahoma City CO	1264	1890-1947	113	......	101	77	9.8
	Oklahoma City AP	1311	1939-1947	109	99	......	......	......
	Tulsa AP	686	1932-1947	109	100	101	77	......
	Waynoka AP	1529	1935-1939	115	103	......	......	......
Ore.	Arlington AP	881	1935-1939	111	95	......	......	......
	Baker CO	3501	1889-1947	104	......	90	66	......
	Baker AP	3374	1939-1947	103	90	......	......	......
	Eugene CO	366	1890-1942	104	......	90	68	......
	Eugene AP	368	1942-1947	105	88	......	......	......
	Medford CO	1428	1911-1929	110	......	95	70	......
	Medford AP	1343	1929-1947	115	95	......	......	......
	Portland CO	98	1874-1947	107	......	90	68	6.5
	Portland AP	25	1940-1947	105	87	......	......	......
	Roseburg CO	523	1877-1947	109	......	90	66	......
Pa.	Curwensville AP	2219	1943-1947	90	82	......	......	......
	Erie CO	771	1873-1946	98	......	93	75	......
	Erie AP	736	1935-1939	96	85	......	......	......
	Harrisburg AP	339	1935-1939	103	91	95	......	......
	Philadelphia CO	200	1871-1947	106	......	95	78	9.7
	Philadelphia AP	18	1940-1947	100	......	......	......	......
	Pittsburgh CO	929	1875-1947	103	......	95	75	8.9
	Pittsburgh AP	1284	1935-1947	102	88	......	......	......
	Reading CO	311	1913-1947	105	......	95	75	......
	Scranton CO	877	1901-1947	103	......	95	75	......
	Sunbury AP	448	1935-1939	101	89	......	......	......
R. I.	Block Island CO	46	1881-1947	93	......	95	75	......
	Providence CO	77	1904-1947	100	......	93	75	9.5
S. C.	Charleston CO	59	1871-1947	104	......	95	78	9.8
	Charleston AP	51	1940-1947	103	91	......	......	......

TABLE 5. PROPOSED STANDARD VALUES OF I_n, DIRECT SOLAR RADIATION RECEIVED AT NORMAL INCIDENCE, AT THE EARTH'S SURFACE[a, b]

Solar Altitude, β Deg	I_n Btu per (Hr) (Sq Ft)	Solar Altitude, β Deg	I_n Btu per (Hr) (Sq Ft)	Solar Altitude, β Deg	I_n Btu per (Hr) (Sq Ft)
5	65	30	234	60	276
10	122	35	245	70	283
15	165	40	253	80	289
20	196	45	260	90	294
25	219	50	266		

[a] Calculated using the following assumptions: barometric pressure of 760 mm Hg (29.921 in.); depth of precipitable water of 20 mm (0.787 in.); dust particles, by counting, 300 per cc; partial pressure of the ozone ayer in the atmosphere of 2.8 mm Hg (0.110 in.). This is representative of a *clear* summer day.

[b] For sea level. As an approximate altitude correction, *add* 1 per cent for each 1000 ft altitude.

determined once the indoor and outdoor design conditions are fixed. Calculations will be discussed subsequently.

INSTANTANEOUS HEAT LOAD

The total cooling load is frequently divided for convenience into two components; *sensible heat* and *latent heat.* While this subdivision is not imperative, past practice has found it convenient.

A gain of *sensible heat* is considered to occur when there is a direct addition of heat to the enclosure by any one or all of the mechanisms of conduction, convection, and radiation. A gain of *latent heat* is considered to occur when there is an addition of water vapor to the air of the enclosure. For example, when the humidity in an enclosure is increased by water vapor emitted by human occupants, or by water vapor resulting from a process such as cooking, the heat required to vaporize the water does *not* come from the air. Maintenance of a constant humidity ratio in a sealed enclosure requires the condensation of water vapor in the cooling apparatus at a rate equal to its rate of addition within the enclosure. The rate of heat removal from this condensing vapor would be substantially equal to the product of the rate of condensation and the latent heat of condensation; this product, expressed in Btu per hour, would be called a *latent heat* load.

As a further example, the infiltration of outdoor air with a high dry-bulb temperature and a high humidity ratio and the corresponding escape of room air at a lower dry-bulb temperature and a lower humidity ratio would increase both the *sensible heat load* and the *latent heat load.*

SOLAR AND SKY RADIATION AND TRANSMISSION LOSSES

Magnitude of Solar Radiation—Calculation Tables

If a plane surface were set perpendicular to the rays of the sun (*e.g.*, for *normal incidence*) outside the earth's atmosphere, it would receive solar

TABLE 6. APPROXIMATE RATIO OF DIRECT SOLAR RADIATION TO SKY RADIATION RECEIVED ON A HORIZONTAL SURFACE ON CLEAR DAYS IN EASTERN STATES[a]

Solar Altitude, β Deg	Ratio e	Solar Altitude, β Deg	Ratio e	Solar Altitude, β Deg	Ratio e
0	0	40	3.84	70	5.63
10	1.40	50	4.55	80	5.90
20	2.30	60	5.20	90	6.10
30	3.10				

[a] For rough estimates assume that the sky radiation on a *vertical* surface is one half of that on a horizontal surface. Sky radiation may be assumed independent of vertical-surface orientation.

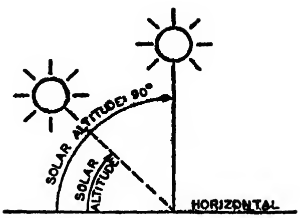


FIG. 1. DEFINITION OF SOLAR ALTITUDE

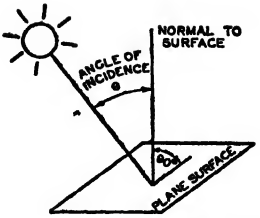


FIG. 2. DEFINITION OF ANGLE OF INCIDENCE

radiation of about 420 Btu per (hr) (sq ft). A similar receiving surface on the surface of the earth would receive radiant energy at a considerably lower rate because a large part of the radiation entering the atmosphere is *scattered* in passing through the air, moisture, smoke, and dust which comprise the earth's envelope. Also, some of the atmospheric constituents, notably water vapor, carbon dioxide, and ozone, *absorb* radiant energy. This absorption and scattering cause different proportionate reductions from outer-atmosphere radiation intensity with different wave-lengths. An exact analysis of these phenomena is beyond the practical purposes of air-conditioning load estimates; the important principle to remember is that the radiation reaching the surface of the earth is the sum of I_n and I_s.

I_n = The *direct radiation* (at normal incidence), which is the *transmitted fraction* of the net sun radiation received by the outer atmosphere, and

I_s = The *sky* or *diffuse* radiation coming *from the atmosphere itself* as a consequence of the scattering and absorption which give rise, in part, to a re-radiation to the earth. The diffuse radiation does not strike only at normal incidence; it strikes at *all* angles from which the sky *sees* the surface in question.

Standardized, practical-purpose values of the *direct* solar radiation incident upon a plane perpendicular to the sun's rays at the earth's surface have been proposed by P. Moon.[2] Table 5 gives these data; they are representative of a *clear* summer day at about sea-level elevation. (For industrial areas, I_n values will be slightly less than Table 5, with the greatest decrease occurring at low solar altitudes towards evening.)

Practical design data on *sky* radiation are meager. Table 6 presents a basis of estimates for *clear* summer days in terms of the direct solar radiation to sky radiation ratio.

The *solar altitude* is the angle (see Fig. 1) between the sun's rays and the horizontal.

In the usual application the receiving surface (*e.g.*, building roof or wall) will not be perpendicular to the rays of the sun. The intensity of the direct radiation incident upon a surface, Btu per (hour) (square foot of absorbing surface), which is oriented with an angle of incidence θ for the sun's rays, is

$$I_d = KI_n, \tag{1}$$

where I_d = Intensity of incident direct radiation, Btu per (hour) (square foot of absorbing surface).

I_n = Intensity of direct radiation on a plane normal to the sun's rays, Btu per (hour) (square foot), (from Table 5).

K = Cosine of the angle of incidence θ.

The angle of incidence (see Fig. 2) is the angle between the sun's rays and the normal to the absorbing surface.

For *horizontal* surfaces the magnitude of K is determined by the time of year, the time of day (sun's position), and the latitude of the location concerned. For *vertical* surfaces, a fourth factor is needed: the *azimuth* of the surface. The azimuth (see Fig. 3) is the angle, measured clockwise, from the south to the exterior side of the wall in question.

Complete tabulated calculations are available for magnitudes of the factor K, extending over all latitudes, all azimuths, all months of the year, and all hours of the day.[8] Illustrative excerpts are given in Tables 7 and 8. Data of this type are valuable for all manner of problems involving solar radiation, and not only for cooling-load calculations.

The reader is warned that values of K from Tables 7 and 8 include only the *direct* radiation; *sky* radiation must be calculated separately and added to the direct radiation (see Table 6).

The use of Tables 5, 6, 7, and 8 requires that the relation between solar

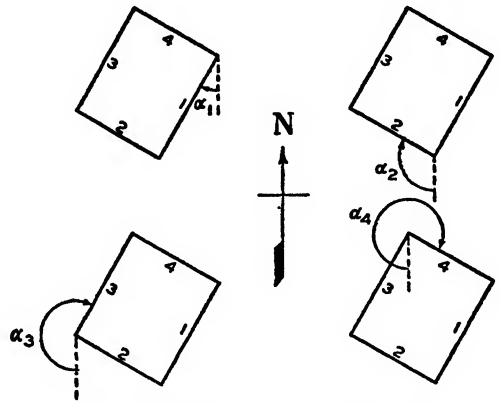


FIG. 3. DEFINITION OF AZIMUTH

The azimuth angle α is always measured clockwise from the south to the exterior side of the wall in question.

altitude and mean sun time be available in convenient form. Table 9 provides this information.

The preceding discussion has been concerned only with *incident* radiation. When radiation is incident upon a surface, part is reflected, part is absorbed, and, if the material transmits radiant energy, part is transmitted. Moreover, the building surfaces themselves *send out* radiant energy to the sky and to the surroundings. The heat balance on an outer surface, for a unit time interval, may be expressed as follows:

Heat entering outer surface	Incident direct solar radiation	Incident sky radiation	Reflected direct radiation
	Reflected sky radiation	Radiation emitted to sky by surface	Air-to-surface convection (2)

The convective heat transfer may be either to or from the surface, depending upon whether the air temperature is higher or lower than the surface temperature. No term has been included to represent radiation exchange with other objects in the surroundings, as this effect is not usually of significant magnitude relative to the others.

In practical calculations, the *difference* between incident and reflected radiation is computed by multiplying the incident radiation by a factor

TABLE 7. VALUES OF K FOR HORIZONTAL PLANES IN NORTH LATITUDES[a] DURING THE PERIOD MAY 2–AUGUST 10

Local Mean Sun Time[b] A.M.	P.M.	North Latitude—Degrees 25	30	35	40	45	50
5	7	......	......	......	.034	.070	.106
6	6	.145	.171	.196	.220	.242	.262
7	5	.365	.382	.395	.406	.414	.418
8	4	.570	.578	.581	.580	.574	.564
9	3	.747	.746	.740	.729	.712	.689
10	2	.882	.876	.863	.843	.817	.785
11	1	.967	.957	.940	.915	.884	.845
12		.996	.985	.966	.940	.906	.866

[a] Figured for solar declination of 20 deg. [b] The relation between local mean sun time and civil time may be obtained from Weather Bureau offices in various localities.

called the absorptivity or the emissivity; see Table 6 and Equation 3, Chapter 5. Convection is calculated as explained in Chapters 5 and 6.

The daytime radiation emitted to the sky by a surface is not usually calculated as a separate item in practical air-conditioning work. Its effect may be roughly accounted for by an adjustment of the magnitude of the absorptivity factor for solar radiation, and this approach is often followed in making emissivity measurements for building materials exposed to the sun and sky.

When a building surface is exposed to a *clear* night sky, there is an appreciable net exchange of low temperature radiant energy from the surface to the sky, but this is not considered for the usual comfort cooling calculation. When peak load conditions occur at night, nocturnal radiation is of particular importance for glass sections. For further information, see Brunts equation as given by Haurwitz.[4]

TABLE 8. VALUES OF K FOR VERTICAL PLANES IN NORTH LATITUDES DURING THE PERIOD MAY 2–AUGUST 10

Latitude[a] Deg	Local Mean Sun Time[b] A.M. →	P.M. →	Azimuth Scales for Values of K in Ordinary Type[c]: 0 / 180	15 / 165	30	45	60	75	90	105	120	135	150	165	180
30	6	6	.940	.831	.666	.455	.213	*.048*	*.296*	*.529*	*.726*	*.874*	*.962*	*.984*	*.940*
	7	5	.908	.832	.699	.518	.303	.066	*.175*	*.404*	*.605*	*.765*	*.873*	*.922*	*.908*
	8	4	.814	.770	.674	.532	.354	.151	*.061*	*.270*	*.460*	*.619*	*.735*	*.802*	*.814*
	9	3	.664	.651	.593	.495	.363	.207	.036	*.137*	*.301*	*.444*	*.557*	*.632*	*.664*
	10	2	.470	.482	.462	.410	.331	.229	.111	*.015*	*.139*	*.254*	*.352*	*.425*	*.470*
	11	1	.243	.276	.289	.283	.258	.217	.158	.089	.015	*.061*	*.132*	*.194*	*.243*
	12		.000	.045	.087	.123	.150	.168	.174	.168	.150	.123	.087	.045	.000
45	5	7	.908	.770	.579	.349	*.095*	*.165*	*.414*	*.635*	*.812*	*.934*	*.993*	*.984*	*.908*
	6	6	.940	.845	.693	.493	.260	.010	*.242*	*.477*	*.679*	*.835*	*.935*	*.970*	*.940*
	7	5	.908	.859	.751	.592	.393	.167	*.070*	*.302*	*.514*	*.691*	*.821*	*.893*	*.908*
	8	4	.814	.809	.750	.639	.485	.298	.090	*.123*	*.329*	*.512*	*.660*	*.763*	*.814*
	9	3	.664	.701	.689	.631	.530	.392	.228	.048	*.135*	*.309*	*.461*	*.583*	*.664*
	10	2	.470	.540	.574	.568	.524	.444	.334	.201	.054	*.096*	*.240*	*.368*	*.470*
	11	1	.243	.338	.411	.455	.468	.449	.400	.323	.225	.111	*.011*	*.131*	*.243*
	12		.000	.109	.211	.299	.366	.408	.423	.408	.366	.299	.211	.109	.000
	Time: P.M. →		360	345	330	315	300	285	270	255	240	225	210	195	180
	Time: A.M. →		180	195	210	225	240	255	270	285	300	315	330	345	360
			Azimuth Scales for Values of K in Italicized Type												

[a] Values for 30 deg latitude may be employed over the range 25 deg to 35 deg. Values for 45 deg latitude may be employed over the range 40 deg to 50 deg. Interpolation will yield values for the range 35 deg to 40 deg latitude.

[b] The relation between local mean sun time and civil time may be obtained from Weather Bureau offices in various localities.

[c] Values in ordinary type are for azimuths 0 to 180 deg. Values in italic type are for azimuths 180 to 360 deg.

The Sol-Air Temperature

A convenient way of combining the effects of solar and sky radiation, solar absorptivity, temperature, and air movement is to use the concept of sol-air temperature. The sol-air temperature, t_e, is the temperature of outdoor air which, in contact with the weather side of a material that is receiving no solar or sky radiation, would give the same rate of heat entry into that surface as would exist with the actual combination of incident solar and sky radiation and convective heat transfer.

In order to calculate the rate of heat transfer into an outside building surface for any instant of time, it is necessary to consider:

1. The intensity of direct solar radiation striking the surface.
2. The absorptivity of the surface for direct solar radiation.
3. The intensity of sky radiation striking the surface.

Table 9. Relation Between Local Mean Sun Time and Solar Altitude During the Period May 2–August 10 for North Latitudes

Local Mean Sun Time		Solar Altitude					
		North Latitude—Degrees					
A.M.	P.M.	25	30	35	40	45	50
5	7	0	0	0	2.	4.	6.
6	6	8.5	10.	11.	12.5	14.	15.
7	5	21.5	22.5	23.5	24.	24.5	24.5
8	4	35.	34.5	35.5	35.5	35.	34.5
9	3	48.5	48.	47.5	47.	45 5	43.5
10	2	62.	61.	59.5	57.5	55.	51.5
11	1	75.5	73.	70.	66.	62.	57 5
12		85.	80.	75.	70.	65.	60.

4. The absorptivity of the surface for sky radiation.
5. The rate at which the surface emits radiation to the sky.
6. The temperature of the surrounding air.
7. The temperature of the outer building surface.
8. The unit convective conductance for heat transfer between the air and the building surface. The magnitude of the convective conductance depends upon the position of the surface and the velocity of the wind or air currents.

The simultaneous consideration of all these effects is too complex for practical application and therefore the sol-air temperature is developed as follows:

1. The equation for the rate of heat transfer into the weather side of a sunlit building material at any instant is written:

$$\left(\frac{q}{A}\right)_L = \quad . + f_o(t_o - t_L) \text{ Btu per (hour) (square foot).} \qquad (3)$$

where b = Absorptivity of weather side of material for incident solar and sky radiation, dimensionless.

I_t = Rate of incidence of solar and sky radiation, Btu per (hour) (square foot).

t_o = Outside air temperature, Fahrenheit degrees.

t_L = Temperature of weather surface of the material, Fahrenheit degrees.

f_o = Unit convective conductance of outside surface, Btu per (hour) (square foot) (Fahrenheit degree).

2. The sol-air temperature is defined as:

$$t_e = t_o + \frac{bI_t}{f_o} \tag{4}$$

3. Then, the instantaneous rate of heat entry into the weather side of the structure becomes:

$$\left(\frac{q}{A}\right)_L = f_o(t_e - t_L) \text{ Btu per (hour) (square foot)} \tag{5}$$

Example 1. If $t_a = 90$ F, $b = 0.7$, $I = 200$ Btu per (hour) (square foot), and $f_o = 4$, find the sol-air temperature, t_e. Substituting these values in Equation 4:

$$t_e = 90 + \frac{0.7\ (200)}{4} = 125 \text{ F.}$$

Thus the instantaneous rate of entry of heat into the weather side of this material is precisely the same as if the air temperature were 125 F with no solar and sky radiation exchange with the surface.

The preceding sol-air concept is intended for non-glass building areas. For glass surfaces a similar method could be developed, with the addition of terms to account for absorption within the material and outward radiation from the interior space. It is customary, however, to treat glass separately; see section on Glass Areas in this chapter.

The sol-air temperature is a composite quantity the magnitude of which is influenced by each of the variables entering its defining equation. Establishing its magnitude for design calculations is part of the broad problem of determining weather-design data. Sufficient studies have been completed, however, to produce some sol-air data of practical value.

Data of the U. S. Weather Bureau for the 10-year period from 1932 through 1941 have been studied for New York, N. Y.,[5] and Lincoln, Nebr.[6] Only simultaneous values of the air temperature and solar and sky radiation have been combined in determining design values of the sol-air temperature for various surfaces at different times of day in these localities. Since the 24-hour average of the sol-air temperature is greater in July than for any other month at both stations, the sol-air temperature at each hour in July, which was equalled or exceeded at that hour only 16 times in 310 observations, was chosen as the *design* sol-air temperature.

Summer design sol-air temperatures are given in Table 10 for New York, N. Y.; Table 11 gives similar data for Lincoln, Nebr. These data may be taken as representative for similar places in northern latitudes in the United States.

Tables 10 and 11, referring to New York for an industrial area and to Lincoln, Nebr., for a non-industrial area, can be used to determine sol-air temperatures for other locations and conditions by applying corrections as explained in the following paragraphs (using the same symbols as given in Tables 10 and 11 but indicated by (′) for other conditions):

1. To adjust the data in Tables 10 and 11 for the variation of I_t with latitude:

a. Compute I'_t for the latitude in question from the data presented previously in this chapter. (For smoky industrial areas, decrease the direct radiation from Table 5 by 15 to 20 per cent for afternoon hours.)

b. Determine the difference $(I'_t - I_t)$ and multiply this by 0.25. Find $I_t = (f_o/b)(t_e - t_o)$ from Tables 10 and 11.

c. Add (algebraically) the difference $0.25\ (I'_t - I_t)$ to the data tabulated.

TABLE 10. SUMMER DESIGN SOL-AIR TEMPERATURES FOR NEW YORK, N. Y. (NORTH LATITUDE 40°46′; ELEVATION 180 FT)

Mean Sun Time	Sol-Air Temperature, t_e Fahrenheit Degrees					
	Any Surface[b]	Horizontal	North	East	South	West
Ratio[a]: $\frac{b}{f_o}$	0	0.25	0.25	0.25	0.25	0.25
12 midnight	79	79	79	79	79	79
1 a.m.	78	78	78	78	78	78
2	77	77	77	77	77	77
3	77	77	77	77	77	77
4	76	76	76	76	76	76
5	76	76	76	76	76	76
6	76	81	80	89	77	77
7	80	96	85	106	82	82
8	82	110	85	114	86	85
9	86	127	90	120	97	90
10	88	137	92	114	104	92
11	90	148	94	106	111	94
12 noon	92	155	97	97	115	98
1 p.m.	93	154	98	98	117	108
2	94	152	99	99	112	126
3	94	144	99	99	102	136
4	94	136	99	99	99	145
5	93	121	103	97	97	140
6	90	106	106	93	93	134
7	88	93	102	90	90	115
8	85	85	85	85	85	85
9	83	83	83	83	83	83
10	82	82	82	82	82	82
11	81	81	81	81	81	81
24-hr avg, t_m	84.8	106.4	88.5	92.3	90.7	96.5

[a] b = surface absorptivity, dimensionless.
f_o = unit convective conductance, Btu per (hr) (sq ft) (F deg)
[b] Values in this column are magnitudes of t_o, the outdoor air temperature.

TABLE 11. SUMMER DESIGN SOL-AIR TEMPERATURES FOR LINCOLN, NEBR. (NORTH LATITUDE 40°50′; ELEVATION 1225 FT)

Mean Sun Time	Sol-Air Temperature, t_e, Fahrenheit Degrees					
	Any Surface[b]	Horizontal	North	East	South	West
Ratio[a]: $\frac{b}{f_o}$	0	0.25	0.25	0.25	0.25	0.25
12 midnight	89	89	89	89	89	89
1 a.m.	88	88	88	88	88	88
2	86	86	86	86	86	86
3	84	84	84	84	84	84
4	84	84	84	84	84	84
5	82	82	82	82	82	82
6	81	87	88	100	82	82
7	82	103	93	125	85	85
8	88	124	94	137	92	92
9	93	143	98	142	104	98
10	96	160	102	138	115	102
11	100	172	106	129	125	106
12 noon	102	178	108	115	130	108
1 p.m.	104	180	110	110	132	119
2	106	178	112	112	131	137
3	107	170	113	113	126	150
4	107	158	112	112	117	158
5	106	142	113	110	110	157
6	105	126	117	108	108	149
7	102	109	113	104	104	128
8	98	99	98	98	98	98
9	94	94	94	94	94	94
10	92	92	92	92	92	92
11	90	90	90	90	90	90
24-hr avg, t_m	94.4	121.6	98.6	105.9	102.1	106.6

[a] b = surface absorptivity, dimensionless.
f_o = unit convective conductance, Btu per (hr) (sq ft) (F deg).
[b] Values in this column are magnitudes of t_o, the outdoor air temperature.

2. To adjust the data in Tables 10 and 11 for variations in t_o:
a. Establish the magnitude of t'_o at the locality in question.
b. Determine the difference $(t'_o - t_o)$.
c. Add (algebraically) the difference $(t'_o - t_o)$ to the data tabulated.

3. To adjust the data in Tables 10 and 11 to other magnitudes of b/f_o than 0.25:
a. For New York or Lincoln, merely interpolate or extrapolate the tabulated data by direct proportion, using the column for $b/f_o = 0$.
b. For other localities, first determine $t'_e = t'_o + b/f_o\, I'_t$ for $b/f_o = 0$ and $b/f_o = 0.25$. Then interpolate or extrapolate by direct proportion as before.

Sol-air data for a particular locality may be adjusted for different azimuths of the vertical surface in question in a similar manner; although, all things considered, a simple interpolation between the four points of the compass should serve for all but the most accurate calculations.

An example of the use of the tables in determining sol-air temperature is given.

Example 2. Find the summer design sol-air temperature in New York, N. Y., for a west wall which has a solar absorptivity of 0.7, at a sun time of 3:00 p. m.

$$\text{Use } f_o = 4\text{; then } \frac{b}{f_o} \text{ for this wall is } \frac{0.7}{4.0} \text{ or } 0.175.$$

From Table 10: for $\frac{b}{f_o} = 0$, $t_e = 94$ F; for $\frac{b}{f_o} = 0.25$, $t_e = 136$ F; and by linear interpolation for this wall:

$$t_e = 94 + \frac{0.175}{0.25}(136 - 94) = 94 + 29 = 123 \text{ F}.$$

Sol-air temperatures are especially helpful in the calculation of periodic heat transfer, as will be illustrated in the material which follows.

PRINCIPLES OF PERIODIC HEAT FLOW

Calculation principles for periodic heat flow are dealt with briefly in this section; in the section which follows, *practical tables are given to facilitate rapid design estimates.* In addition to the rate of heat entry into the outside building surface these tables take into account the following factors:

1. The thermal conductivity of material.
2. The density, specific heat and character of material.
3. Thickness of material.
4. Room air temperature.
5. Unit convective conductance between the inside surface and room air; and radiant heat transfer between the inside surfaces and other surfaces in the room.

Time Lag

The fundamental analysis of periodic heat flow is complicated when compared with steady-state calculations on account of the time-variable storage of heat from point to point through a wall or roof. The cyclic variation of outdoor conditions produces a related cyclic variation of temperature and heat flow throughout each structural section exposed to the weather. The cyclic variations undergo a progressive *shift in phase* and

decrease in amplitude in going through a wall with constant conditions maintained in the indoor space.

By a *shift in phase* is meant that as the cyclic temperature wave passes through the wall, the time of occurrence of the maximum temperature at any point shifts farther and farther behind the time of the outer-surface maximum for successive positions through the wall. The resultant *time lag* between the outer-surface and inner-surface maximum temperatures is important, for it may be the determining factor in fixing the time of the maximum cooling load.

By a *decrease in amplitude* is meant that as the cyclic temperature wave passes through the wall, the difference between the maximum temperature of a cycle and the mean temperature of the cycle, which is the amplitude of the wave by definition, decreases progressively as the wave passes through the wall. The magnitude of the *temperature amplitude* at the inner wall surface is necessary for the determination of the instantaneous rate of heat transfer to the indoor space.

Practical design data for periodic heat flow comprise a means of determining the time lag and amplitude decrement for different wall constructions and any given outdoor cycle of sol-air temperature. Both analytical and experimental studies have been made on this problem[7]. While the analytical solution has been written, it is far too detailed for direct use in rapid practical work; and the extensive numerical work required to establish a basis for simplified calculations has been only partially completed. The method reported by Mackey and Wright[7] will be adopted as the basis for the design procedure recommended here.

Homogeneous Walls or Roofs, Constant Indoor Temperature

For walls or roofs of a single, homogeneous material, the *instantaneous* rate of heat gain within an enclosure where the indoor air temperature is held constant is, approximately,

$$\frac{q}{A} = U\,(t_m - t_i) + \lambda U\,(t_e^* - t_m) \text{ Btu per (hour) (square foot)} \qquad (6)$$

where t_m = 24-hr average sol-air temperature for the particular value of $\frac{b}{f_o}$, Fahrenheit degrees.

λ = Amplitude decrement factor, a variable that depends upon the thickness, material, and orientation of the wall or roof; see Table 12 for values. The amplitude decrement factor λ as used in this chapter is equivalent to $\left(\frac{1.65 \times \lambda}{U}\right)$ as defined by Mackey and Wright[7] and also used by Stewart[11].

t_e^* = Sol-air temperature at a time earlier than the time for which the heat gain is being found by an amount that is equal to the time lag of the wall or roof, Fahrenheit degrees; see Table 12 for values of time lag.

U = Over-all coefficient of heat transfer of the wall or roof, Btu per (hour) (square foot) (Fahrenheit degree),

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{L}{k}} = \frac{1}{\frac{1}{1.65} + \frac{1}{4} + \frac{L}{k}} = \frac{1}{0.856 + \frac{L}{k}}$$

L = Thickness of building material, feet.

k = Thermal conductivity of building material, Btu per (hour) (square foot) (Fahrenheit degree per foot).

f_i = Unit convective conductance (film coefficient of heat transfer) of indoor air, Btu per (hour) (square foot) (Fahrenheit degree).

f_o = Unit convective conductance (film coefficient of heat transfer) of outdoor air for summer conditions of approximately 7.5 mph wind velocity, Btu per (hour) (square foot) (Fahrenheit degree).

The time at which the *maximum occurs in the rate of heat entry into the outside surface of walls or roofs* is taken as the time at which the peak point occurs in the sol-air temperature cycle (mean *sun time* is used in the sol-air cycles). The corresponding maximum rate of heat entry follows from Equation 6 with $t_e{}^*$ being the maximum temperature of the sol-air cycle.

The time of *maximum heat gain to the room* is obtained by adding the time lag to the time of maximum sol-air temperature (from Tables 10 or 11) for the particular wall or roof.

The magnitude of the second term in Equation 6 relative to the first term indicates the relative portion of the structural heat in-flow assignable

TABLE 12. PERIODIC HEAT FLOW DATA FOR HOMOGENEOUS WALLS OR ROOFS

Material	Thickness, In.	Over-all Coefficient, Btu per (hr) (sq ft) (°F) U[a]	Thermal Resistance of Solid Material, (hr) (sq ft) (°F)/Btu $\frac{L}{k}$	Time Lag, Hr	Factor, λ, in Equation 6			
					Horizontal and North	East	South	West
Stone	8	0.57	0.64	5.5	0.51	0.36	0.48	0.42
	12	0.55	0.96	8.0	0.28	0.19	0.26	0.22
	16	0.47	1.28	10.5	0.17	0.10	0.15	0.13
	24	0.36	1.92	15.5	0.06	0.03	0.05	0.04
Solid Concrete	2	0.98	0.17	1.1	0.93	0.87	0.92	0.89
	4	0.84	0.33	2.5	0.79	0.68	0.76	0.72
	6	0.74	0.50	3.8	0.61	0.46	0.58	0.51
	8	0.66	0.67	5.1	0.49	0.33	0.46	0.39
	12	0.54	1.00	7.8	0.29	0.17	0.26	0.22
	16	0.46	1.33	10.2	0.17	0.09	0.15	0.12
Common Brick	4	0.60	0.80	2.3	0.83	0.75	0.81	0.78
	8	0.41	1.60	5.5	0.51	0.39	0.49	0.44
	12	0.31	2.40	8.5	0.26	0.17	0.25	0.21
	16	0.25	3.20	12.0	0.13	0.08	0.12	0.10
Face Brick	4	0.77	0.44	2.4	0.81	0.70	0.78	0.74
Wood	½	0.68	0.62	0.17	1.0	1.0	1.0	1.0
	1	0.48	1.25	0.45	1.0	0.99	0.99	0.99
	2	0.30	2.50	1.3	0.98	0.91	0.96	0.94
Insulating Board	½	0.42	1.51	0.08	1.0	1.0	1.0	1.0
	1	0.26	3.03	0.23	1.0	1.0	1.0	1.0
	2	0.14	6.05	0.77	1.0	1.0	1.0	1.0
	4	0.08	12.1	2.7	0.83	0.74	0.81	0.76
	6	0.05	18.2	5.0	0.64	0.49	0.61	0.55

[a] Based upon an outdoor combined film coefficient of 4.0 and an indoor combined film coefficient of heat transfer of 1.65 Btu per (hour) (square foot) (Fahrenheit degree).

to periodic heat flow. The periodic term is continually passing through a *cyclic variation* from zero to a positive maximum, to zero, to a negative maximum, to zero again and so on over each 24 hour cycle. Surfaces with different exposures pass through these cycles with maximum points at different times of day.

An example in the use of Tables 10, 11, and 12 follows:

Example 3. Find the instantaneous design rate of heat gain through an 8-in. west wall of common brick ($b = 0.7, f_o = 4$) located in New York, N. Y., at 9:30 p.m. sun time, when the temperature of the indoor air is constant at 80 F.

From Table 12, $U = 0.41$, the time lag is 5.5 hr, and $\lambda = 0.44$.

By linear interpolation on the basis of the value of b/f_o in Table 10,

$$t_m = 84.8 + \frac{0.175}{0.25}(96.5 - 84.8) = 93 \text{ F.}$$

Table 13. Summer Design Sol-Air Temperatures Used for Tables 14 and 15

Mean Sun time	Sol-Air Temperature t_e Fahrenheit Degrees								
	Any Surface[b]	Horiz.	North	East		South		West	
Ratio[a]: $\frac{b}{f_o}$	0	0.225	0	0.225	0.125	0.225	0.125	0.225	0.125
12 Midnight	77	77	77	77	77	77	77	77	77
1 AM	76	76	76	76	76	76	76	76	76
2	76	76	76	76	76	76	76	76	76
3	75	75	75	75	75	75	75	75	75
4	74	74	74	74	74	74	74	74	74
5	74	74	74	75	80	74	74	74	74
6	74	76	74	110	93	74	74	74	74
7	75	91	75	123	100	75	75	75	75
8	77	106	77	126	103	82	78	77	77
9	80	119	80	125	104	93	86	80	80
10	83	129	83	117	100	102	93	83	83
11	87	137	87	108	96	110	99	89	87
12 Noon	90	142	90	92	92	114	104	96	92
1 PM	93	144	93	93	93	115	105	110	102
2	94	140	94	95	94	111	104	124	111
3	95	132	95	95	95	104	100	135	119
4	94	120	94	94	94	99	96	141	120
5	93	107	93	93	93	95	94	139	118
6	91	96	91	91	91	91	91	125	111
7	87	90	87	87	87	88	87	103	94
8	85	85	85	85	85	85	85	85	85
9	83	83	83	83	83	83	83	83	83
10	81	81	81	81	81	81	81	81	81
11	79	79	79	79	79	79	79	79	79
24 Hr Avg t_m	83.1	100.5	83.1	93.0	88.4	89.0	86.2	93.0	88.4

[a] b = surface absorptivity, dimensionless: roof = 0.9; dark walls = 0.9, and light walls = 0.5. f_o = unit convective conductance = 4.0 Btu per (hr) (F deg).
[b] values in this column are magnitudes of t_o, the outdoor air temperature.

The design sol-air temperature at a time earlier than 9:30 p.m. by the time lag (at 4:00 p.m.) for a west wall in New York, N. Y., is, from Table 10,

$$t'_e = 94 + \frac{0.175}{0.25}(145 - 94) = 129.7 \text{ F}$$

From Equation 6, the instantaneous design rate of heat gain is

$$\frac{q}{A} = 0.41\,[(93 - 80) + 0.44\,(129.7 - 93)]$$

$$= 11.9 \text{ Btu per hour for each square foot of sunlit surface.}$$

From Table 12, the time lag is 5.5 hr. From Table 10, the time of maximum rate of heat entry for a west wall is 4:00 p.m. The time of maximum instantaneous rate of heat gain by the enclosure, as far as the west wall is concerned, is then 4:00 p.m. plus 5.5 hr or 9:30 p.m. (This is *sun time*.) The computed rate is therefore the maximum.

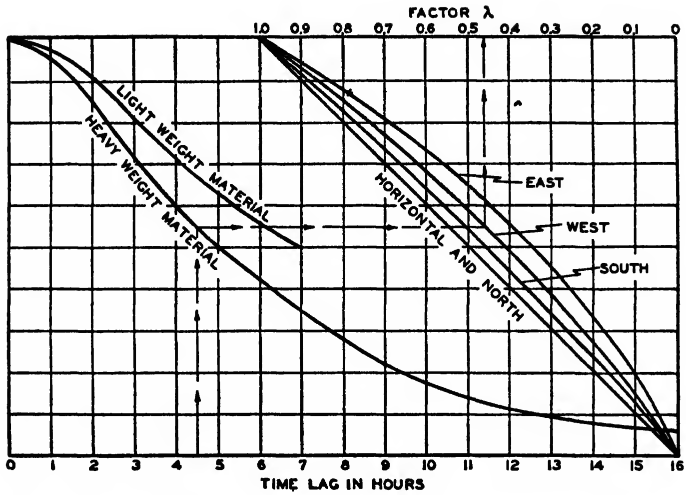


FIG. 4. APPROXIMATE VALUE OF THE AMPLITUDE DECREMENT FACTOR λ FOR USE IN EQUATION 6

Composite Walls or Roofs, Constant Indoor Temperature

A composite wall or roof is made up of two or more layers of different materials. Since the analytical solution for this type of construction has not been reduced to simple and practical terms, it is necessary at present to utilize approximate procedures. In accord with the results of comparative calculations, the following procedures are suggested[7].

To find the time lag for a composite construction:

a. Find the time lag for each layer from Table 12.

b. Add the individual time lags, recognizing that this sum will always be *less* than the true time lag for the actual composite wall.

c. To the sum from (b), *add* an *arbitrary* additional lag of ½ to 1 hr to obtain the estimated lag for the actual construction. For two-layer and light construction walls, the ½-hr value will be suitable while, for walls of three or more layers, or very heavy construction, the 1-hr value is preferred. For intermediate conditions, individual judgment is the only guide. (Computed time lags should not be considered to be accurate closer than to about the nearest hour by this method.)

To find the amplitude decrement factor, λ, *for composite construction:*

Having determined the time lag and the orientation, use Fig. 4. (Note also that the factor λ for a composite construction should never exceed the *product* of the factors for the individual layers.)

One valuable result of the analytical studies made to date on composite walls has been the demonstration of the effect of the *order* of the materials. Other factors remaining the same, the use of the material of lower density on the weather side will increase the time lag and decrease the instantaneous maximum rate of heat gain.

Two examples are given to show how to estimate, roughly, the instantaneous rate of heat gain through sunlit composite walls or roofs.

Example 4. Estimate the maximum instantaneous design rate of heat gain from a horizontal roof in New York, N. Y., that is made up of black built-up roofing on the

weather side ($b = 1$; $f_o = 4$; thermal resistance = 0.28), 1 in. of insulating board, and 4 in. of concrete with no ceiling, when the temperature of the indoor air is 80 F.

The over-all coefficient of heat transfer for this construction is:

$$U = \frac{1}{0.25 + 0.28 + 3.03 + 0.33 + 0.61} = 0.22 \text{ Btu per (hour) (square foot) (F deg).}$$

If the time lag of the built-up roofing be ignored, the sum of the time lags of the individual layers is, from Table 12, (0.23 + 2.5) or 2.73 hr.

Actually, the time lag will be between 0.5 hr and 1.0 hr greater than this, so assume a time lag of 3.5 hr.

From Table 12, the homogeneous concrete roof having a time lag of 3.5 hr would have a value of λ of about 0.65; use this value for the composite roof.

From Fig. 4, λ is approximately 0.65.

With values of t_m and $t_e{}^*$ found from Table 10, as in previous examples, use Equation 6 and find the maximum *design instantaneous rate* of heat gain as:

$$\frac{q}{A} = 0.22\,[(106.4 - 80) + 0.65\,(155 - 106.4)] = 13 \text{ Btu per (hr) (sq ft).}$$

The maximum instantaneous rate of heat gain from this roof would occur at about 3:30 p.m., sun time.

Example 5. Estimate the maximum instantaneous design rate of heat gain from a south wall in Lincoln, Nebr., consisting of 4 in. of face brick ($b = 0.7$; $f_o = 4$), 4 in. of common brick, furred, with an air space (thermal resistance = 0.75), and finished on the inside with ¾ in. of plaster on metal lath (thermal resistance = 0.23); the temperature of the indoor air is constant at 80 F.

The over-all coefficient of heat transfer for this construction is:

$$U = \frac{1}{0.25 + 0.44 + 0.80 + 0.75 + 0.23 + 0.61} = 0.32 \text{ Btu per (hr) (sq ft) (F deg).}$$

From Table 12, the sum of the time lags for the face brick and the common brick is (2.4 + 2.3) or 4.7 hr. The actual time lag will be slightly greater than this, and a value of 5.5 hr will be assumed.

From Fig. 4, λ is approximately 0.45.

By interpolation in Table 11, $t_m = 99.8$ F, and $t_e{}^* = 123.6$ F. From Equation 6 the maximum instantaneous design rate of heat gain is:

$$\frac{q}{A} = 0.32\,[(99.8 - 80) + 0.45\,(123.6 - 99.8)] = 9.8 \text{ Btu per (hr) (sq ft).}$$

The time of this heat gain is about 6:30 p.m., sun time.

Those concerned with a further study of the details of cooling-load estimates in particular relation to periodic heat flow will find much of value and interest in the reports of experimental studies of these problems[7, 8, 9, 10].

PRACTICAL TABLES FOR CALCULATING SOLAR HEAT GAIN THROUGH WALLS AND ROOFS

Use of Equivalent Temperature Differentials

The preceding paragraphs have explained the principles and methods used in estimating solar heat gain by use of sol-air temperature. This method is rather tedious and is not convenient for every-day use. Some new practical tables[11] have therefore been developed using the basic method reported by Mackey and Wright[7]. These new tables utilize *equivalent temperature differentials* which may be multiplied by the overall heat transmission coefficient U to give directly the total heat transmission,

Btu per square foot, from solar radiation and from temperature difference between outside and room air.

These tables were prepared from sol-air data as shown in Table 13 which is approximately the same as New York data in Table 10. It is suggested that Tables 14 and 15 be used for general estimating purpose.

These analytical procedures as well as those using Tables 10, 11 and 12 presented here yield generally *higher* rates of heat gain than reported for Pittsburgh in early A.S.H.V.E. experimental studies. Current authoritative opinion is that the analytical calculations are to be preferred, all factors considered.

Tables 14 and 15 are based on an equation re-arranged from Equation 6 to read:

$$\frac{q}{A} = U\,[t_m + \lambda\,(t_e^* - t_m) - t_i]. \tag{7}$$

Let $t_m + \lambda\ (t_e^* - t_m) = t_p$, a net equivalent outdoor temperature for combined periodic and mean heat flow. Magnitudes of t_p will vary cyclically with time. Then,

$$\frac{q}{A} = U\,(t_p - t_i) \tag{8}$$

which is a simple form analogous to the steady state equations of Chapters 5 and 6. The rate of heat flow is obtained by multiplying the overall heat transmission coefficient of the structure by the equivalent temperature differential obtained from the tables.

Tables 14 and 15 were developed by using an outside film conductance of 4.0 and an inside film conductance of 1.65 Btu (hr) (sq ft) (F deg). A reduction was made in the temperature differentials for roofs amounting to some 20 per cent of solar radiation as explained by Stewart[11]. This was to compensate for several factors, one of which is the radiant heat lost to the sky which is not included in the Mackey and Wright method. The *temperature differentials* for roofs were based on an inside film conductance of 1.65 because the charts prepared by Mackey and Wright[7] used this value and it was not considered practicable to repeat their work using a different film coefficient. An examination of the values given in their paper indicates that the *temperature differential* would be changed very little even if a value 1.20 were used instead of 1.65. But to obtain the heat flow rates through roofs, more accurate values will be obtained if the *overall heat transmission coefficient is calculated using* 1.2 as the inside film conductance of heat transfer in summer.

The roof coefficients of transmission for summer shown in Table 16 are based on film conductances f of 4.0 for an outside roof surface and 1.20 for an inside ceiling surface. The outside conductance 4.0 is used for summer because it corresponds to a wind velocity of approximately 7.5 mph averaged for rough and smooth surfaces and is more representative of summer wind velocities. Also the lower wind velocity should be used in order to be on the safe side in determining the sol-air temperature. The inside conductance 1.20 is used because the convective portion of the film conductance factor of downward heat flow from a horizontal surface is appreciably less than the winter conductance which applies when heat is flowing upward.

Since there is little difference in *wall transmission coefficients* for summer, based on conductances of 4.0 and 1.65, and the winter coefficients, based

Table 14. Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Roofs

Description of Roof Construction[a]	Sun Time								
	A.M.			P.M.					
	8	10	12	2	4	6	8	10	12
Light Construction Roofs—Exposed to Sun									
1″ Wood[b] or 1″ Wood[b] + 1″ or 2″ Insulation	12	38	54	62	50	26	10	4	0
Medium Construction Roofs—Exposed to Sun									
2″ Concrete or 2″ Concrete + 1″ or 2″ Insulation or 2″ Wood[b]	6	30	48	58	50	32	14	6	2
2″ Gypsum or 2″ Gypsum + 1″ Insulation; 1″ Wood[b] or 2″ Wood[b] or 2″ Concrete or 2″ Gypsum + 4″ Rock Wool in Furred Ceiling	0	20	40	52	54	42	20	10	6
4″ Concrete or 4″ Concrete with 2″ Insulation	0	20	38	50	52	40	22	12	6
Heavy Construction Roofs—Exposed to Sun									
6″ Concrete	4	6	24	38	46	44	32	18	12
6″ Concrete + 2″ Insulation	6	6	20	34	42	44	34	20	14
Roofs Covered with Water—Exposed to Sun									
Light Construction Roof with 1″ Water	0	4	16	22	18	14	10	2	0
Heavy Construction Roof with 1″ Water	−2	−2	−4	10	14	16	14	10	6
Any Roof with 6″ Water	−2	0	0	6	10	10	8	4	0
Roofs with Roof Sprays—Exposed to Sun									
Light Construction	0	4	12	18	16	14	10	2	0
Heavy Construction	−2	−2	2	8	12	14	12	10	6
Roofs in Shade									
Light Construction	−4	0	6	12	14	12	8	2	0
Medium Construction	−4	−2	2	8	12	12	10	6	2
Heavy Construction	−2	−2	0	4	8	10	10	8	4

[a] Includes ⅜ in. felt roofing with or without slag. May also be used for shingle roof. [b] Nominal thickness of the wood.

NOTES FOR TABLE 14

Explanation: {Total heat transmission from solar radiation and temperature difference between outside and room air. Btu per (hr) (sq ft) of roof area} = {Equivalent temperature differential from above table} × {Heat transmission coefficient for summer Btu per (hr) (sq ft) F deg}

1. *Source.* Calculated by Mackey and Wright method (see reference list) and adjusted after studying ASHVE original test data. Estimated for July in 40 deg north latitude. (For Sol-air temperatures used in calculations see Table 13.) For typical design day where the maximum outdoor temperature is 95 F and minimum temperature at night is approximately 75 F (daily range of temperature, 20 F) mean 24 hr temperature 84 F for a room temperature of 80 F. All roofs have been assumed a dark color which absorbs 90 per cent of solar radiation, and reflects only 10 per cent.

2. *Application.* These values may be used for all normal air conditioning estimates; usually without correction, in latitude 0 deg to 50 deg north or south when the load is calculated for the hottest weather. Note 5 explains how to adjust the temperature differential for other room and outdoor temperatures.

3. *Peaked Roofs.* If the roof is peaked and the heat gain is primarily due to solar radiation, use for the area of the roof, the area projected on a horizontal plane.

4. *Attics.* If the ceiling is insulated and if a fan is used in the attic for positive ventilation, the total temperature differential for a roof exposed to the sun may be decreased 25 per cent.

5. *Corrections. For temperature difference when outdoor maximum design temperature minus room is different from 15 deg.* If the outdoor design temperature minus room temperature is different from the base of 15 deg, correct as follows: When the difference is greater (or less) than 15 deg add the excess to (or subtract the deficiency from) the above differentials.

For outdoor daily range of temperature other than 20 deg. If the daily range of temperature is less than 20 deg, add 1 deg for every 2 deg lower daily range; if the daily range is greater than 20 deg, subtract 1 deg for every 2 deg higher daily range. For example, the daily range in Miami, Florida is 12 deg or 8 deg less than 20 deg, therefore, the correction is + 4 deg at all hours of the day.

Light Colors. Credit should not be taken for light colored roofs except where the permanence of the light color is established by experience, as in rural areas or where there is little smoke. When the exterior surface of roof exposed to the sun is a light color, such as white or aluminum (which absorb approximately 50 per cent and reflect 50 per cent of the solar radiation) add to the temperature differential for roof in shade 55 per cent of the difference between the roof in sun and roof in shade. When the roof exposed to the sun is a medium color such as light grey, blue or green, or bright red, add 80 per cent of this difference.

For solar transmission in latitudes other than 40 deg north; and in other months. The table values of temperature differentials will be approximately correct for a roof in the following months:

North Latitude		South Latitude	
Latitude (deg)	Months	Latitude (deg)	Months
0	All Months	0	All Months
10	All Months	10	All Months
20	All Months except Nov, Dec, Jan	20	All Months except May, June, July
30	Mar, Apr, May, June, July, Aug, Sept	30	Sept, Oct, Nov, Dec, Jan, Feb, Mar
40	April, May, June, July, Aug	40	Oct, Nov, Dec, Jan, Feb
50	May, June, July	50	Nov, Dec, Jan

For other months, the total temperature differential (t_x) may be approximated by the use of the following formula:

$$t_x = t_s + \frac{I_x}{I_y}(t_v - t_s)$$

where t_s = temperature differential for the same wall in shade for desired time of day; obtained from Table 14.

I_y = maximum solar transmission through glass, Btu per (hr) (sq ft) for flat skylight in July, 40 deg north latitude (*Note:* this is maximum value irrespective of time).

I_x = same as I_y except use the maximum value for flat skylight, for month, and latitude desired for t_x.

t_v = temperature differential for particular roof exposed to sun for the desired time of day from Table 14.

(*Note that this makes adjustment only for solar radiation and that there may be additional correction for outdoor temperature.*)

on 6.0 and 1.65, it is recommended that the overall coefficient U, *for walls*, be taken directly from the tables in Chapter 6 in which they are based on an outside film conductance of 6.0, corresponding to a 15 mph wind velocity.

Advantages of Equivalent Temperature Differential Method

1. The total sensible heat flow is obtained by multiplying the overall heat transmission coefficient, U, and the equivalent temperature differential indicated in Tables 14 and 15.

2. The temperature differentials listed for a few representative types of construction may be used on all classes of walls and roofs, even though the overall heat transmission coefficient is different, provided the structure has thermal and physical properties similar to one of those listed in Tables 14 and 15.

3. Adjustments can be made, according to instructions given in the footnotes for room and outdoor conditions different from those on which the tables are based.

Examples of Use of Equivalent Temperature Tables

Example 6. Given: A roof is constructed of 6 in. of stone concrete with 2 in. of insulating board and tar felt roofing ⅜ in. thick; and is exposed to the sun. The location is the central part of the United States. Find the rate of heat flow into building

Table 15. Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Walls

North Latitude Wall Facing	Sun Time																		South Latitude Wall Facing
	A.M.						P.M.												
	8		10		12		2		4		6		8		10		12		
	Exterior color of Wall—D = dark, L = light																		
	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	
	Frame																		
NE	22	10	24	12	14	10	12	10	14	14	14	14	10	10	6	4	2	2	SE
E	30	14	36	18	32	16	12	12	14	14	14	14	10	10	6	6	2	2	E
SE	13	6	26	16	28	18	24	16	16	14	14	14	10	10	6	4	2	2	NE
S	−4	−4	4	0	22	12	30	20	26	20	16	14	10	10	6	6	2	2	N
SW	−4	−4	0	−2	6	4	26	22	40	28	42	28	24	20	6	4	2	2	NW
W	−4	−4	0	0	6	6	20	12	40	28	48	34	22	22	8	8	2	2	W
NW	−4	−4	0	−2	6	4	12	10	24	20	40	26	34	24	6	4	2	2	SW
N (Shade)	−4	−4	−2	−2	4	4	10	10	14	14	12	12	8	8	4	4	0	0	S (Shade)
	4 In. Brick or Stone Veneer + Frame																		
NE	−2	−4	24	12	20	10	10	6	12	10	14	14	12	12	10	10	6	4	SE
E	2	0	30	14	31	17	14	14	12	12	14	14	12	12	10	8	6	6	E
SE	2	−2	20	10	28	16	26	16	18	14	14	14	12	12	10	8	6	6	NE
S	−4	−4	−2	−2	12	6	24	16	26	18	20	16	12	12	8	8	4	4	N
SW	0	−2	0	−2	2	2	12	8	32	22	36	26	34	24	10	8	6	6	NW
W	0	−2	0	0	4	2	10	8	26	18	40	28	42	28	16	14	6	6	W
NW	−4	−4	−2	−2	2	2	8	6	12	12	30	22	34	24	12	10	6	6	SW
N (Shade)	−4	−4	−2	−2	0	0	6	6	10	10	12	12	12	12	8	8	4	4	S (Shade)
	8 In. Hollow Tile or 8 In. Cinder Block																		
NE	0	0	0	0	20	10	16	10	10	6	12	10	14	12	12	10	8	8	SE
E	4	2	12	4	24	12	26	14	20	12	12	10	14	12	14	10	10	8	E
SE	2	0	2	0	16	8	20	12	20	14	14	12	14	12	12	10	8	6	NE
S	0	0	0	0	2	0	12	6	24	14	26	16	20	14	12	10	8	6	N
SW	2	0	2	0	2	0	6	4	12	10	26	18	30	20	26	18	8	6	NW
W	4	2	4	2	4	2	6	4	10	8	18	14	30	22	32	22	18	14	W
NW	0	0	0	0	2	0	4	2	8	6	12	10	22	18	30	22	10	8	SW
N (Shade)	−2	−2	−2	−2	−2	−2	0	0	6	6	10	10	10	10	10	10	6	6	S (Shade)
	8 In. Brick or 12 In. Hollow Tile or 12 In. Cinder Block																		
NE	2	2	2	2	10	2	16	8	14	8	10	6	10	8	10	10	10	8	SE
E	8	6	8	6	14	8	18	10	18	10	14	8	14	10	14	10	12	10	E
SE	8	4	6	4	6	4	14	10	18	12	16	12	12	10	12	10	12	10	NE
S	4	2	4	2	4	2	4	2	10	6	16	10	16	12	12	10	10	8	N
SW	8	4	6	4	6	4	8	4	10	6	12	8	20	12	24	16	20	14	NW
W	8	4	6	4	6	6	8	6	10	6	14	8	20	16	24	16	24	16	W
NW	2	2	2	2	2	2	4	2	6	4	8	6	10	8	16	14	18	14	SW
N (Shade)	0	0	0	0	0	0	0	0	2	2	6	6	8	8	8	8	6	6	S (Shade)
	12 In. Brick																		
NE	8	6	8	6	8	4	8	4	10	4	12	6	12	6	10	6	10	6	SE
E	12	8	12	8	12	8	10	6	12	8	14	10	14	10	14	8	14	8	E
SE	10	6	10	6	10	6	10	6	10	6	12	8	14	10	14	10	12	8	NE
S	8	6	8	6	6	4	6	4	6	4	8	4	10	6	12	8	12	8	N
SW	10	6	10	6	10	6	10	6	10	6	10	8	10	8	12	8	14	10	NW
W	12	8	12	8	12	8	10	6	10	6	10	6	10	6	12	8	16	10	W
NW	8	6	8	6	8	4	8	4	8	4	8	4	8	6	10	6	10	6	SW
N (Shade)	4	4	2	2	2	2	2	2	2	2	2	2	2	2	4	4	6	6	S (Shade)
	8 In. Concrete or Stone or 6 In. or 8 In. Concrete Block																		
NE	4	2	4	0	16	8	14	8	10	6	12	8	12	10	10	8	8	6	SE
E	6	4	14	8	24	12	24	12	18	10	14	10	14	10	12	10	10	8	E
SE	6	2	6	4	16	10	18	12	18	12	14	12	12	10	12	10	10	8	NE
S	2	1	2	1	4	1	12	6	16	12	18	12	14	12	10	8	8	6	N
SW	6	2	4	2	6	2	8	4	14	10	22	16	24	16	22	16	10	8	NW
W	6	4	6	4	6	4	8	6	12	8	20	14	28	18	26	18	14	10	W
NW	4	2	4	0	4	2	4	4	6	6	12	10	20	14	22	16	8	6	SW
N (Shade)	0	0	0	0	0	0	2	2	4	4	6	6	8	8	6	6	4	4	S (Shade)

TABLE 15—*Concluded.* TOTAL EQUIVALENT TEMPERATURE DIFFERENTIALS FOR CALCULATING HEAT GAIN THROUGH SUNLIT AND SHADED WALLS

North Latitude Wall Facing	Sun Time																		South Latitude Wall Facing
	A.M.						P.M.												
	8		10		12		2		4		6		8		10		12		
	Exterior color of Wall—D = dark, L = light																		
	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	
	12 In. Concrete or Stone																		
NE	6	4	6	2	6	2	14	8	14	8	10	8	10	8	12	10	10	8	SE
E	10	6	8	6	10	6	18	10	18	12	16	10	12	10	14	10	14	10	E
SE	8	4	8	4	6	4	14	8	16	10	16	10	14	10	12	10	12	10	NE
S	6	4	4	2	4	2	4	2	10	6	14	10	16	12	14	10	10	8	N
SW	8	4	8	4	6	4	6	4	8	6	10	8	18	14	20	14	18	12	NW
W	10	6	8	6	8	6	10	6	10	6	12	8	16	10	24	14	22	14	W
NW	6	4	6	2	6	2	6	4	6	4	8	6	10	8	18	12	20	14	SW
N (Shade)	0	0	0	0	0	0	0	0	2	2	4	4	6	6	8	8	6	6	S (Shade)

NOTES FOR TABLE 15

Explanation: Total heat transmission from solar radiation and temperature difference between outside and room air Btu per (hr) (sq ft wall area) = Equivalent temperature differential from above table × {Heat transmission coefficient for wall Btu per (hr) (sq ft) (F deg)}

NOTES:

1. *SOURCE.* Same as Table 14. A north wall has been assumed to be a wall in the shade; this is practically true. Dark colors on exterior surface of walls have been assumed to absorb 90 per cent of solar radiation and reflect 10 per cent; white colors absorb 50 per cent and reflect 50 per cent. This includes some allowance for dust and dirt since clean, fresh white paint normally absorbs only 40 per cent of solar radiation.

2. *APPLICATION.* These values may be used for all normal air conditioning estimates, usually without corrections, when the load is calculated for the hottest weather. Correction for latitude (Note 3) is necessary only where extreme accuracy is required. There may be jobs where the indoor room temperature is considerably above or below 80 F or where the outdoor design temperature is considerably above 95 F, in which case it may be desirable to make correction to the temperature differentials shown. The solar intensity on all walls other than east and west varies considerably with time of year.

3. *CORRECTIONS. Outdoor minus room temperature.* If the outdoor maximum design temperature minus room temperature is different from the base of 15 deg, correct as follows. When the difference is greater (or less) than 15 deg add the excess to (or subtract the deficiency from) the above differentials.

Outdoor daily range temperature. If the daily range of temperature is less than 20 deg add 1 deg to every 2 deg lower daily range; if the daily range is greater than 20 deg, subtract 1 deg for every 2 deg higher daily range. For example, the daily range in Miami, Florida is 12 deg, or 8 deg less than 20 deg; therefore, the correction is + 4 deg.

Color of exterior surface of wall. Use temperature differentials for light walls only where the permanence of the light wall is established by experience. For cream colors use the values for light walls. For medium colors interpolate half way between the dark and light values. Medium colors are medium blue, medium green, bright red, light brown, unpainted wood, natural color concrete, etc. Dark blue, red, brown, green, etc., are considered dark colors.

For latitudes other than 40 deg north; and in other months. These table values will be approximately correct for the east or west wall in any latitude (0 deg to 50 deg North or South) during the hottest weather. In the lower latitudes when the maximum solar altitude is approximately 80 deg to 90 deg (the maximum occurs at noon) the temperature differential for either a south or north wall will be approximately the same as a north, or shade wall. The temperature differential (t_x) for any wall facing, and for any latitude for any month may be approximated as follows:

$$t_x = t_s + \frac{I_2}{I_1} \times (t_w - t_s)$$

where t_s = temperature differential for the same wall in shade for desired time of day; obtained from Table 15.

I_1 = maximum solar radiation intensity transmitted through glass, Btu per (hr) (sq ft) for particular wall facing, in July, 40 deg north latitude (*note:* this is maximum value irrespective of time).

I_2 = same as I_1 except use the maximum value for wall facing, for month, and latitude desired for t_x.

t_w = temperature differential for particular wall facing, for the desired time of day from above table.

(*Note that this makes adjustment only for solar radiation and that there may be additional correction for outdoor temperature.*)

4. *FOR INSULATED WALLS* use same temperature differentials as used for uninsulated

at 2:00 P.M. during July for an outdoor design temperature 95 F and an inside temperature 80 F.

Solution: From Table 14 in 2 P.M. column for 6 in. concrete plus 2 in. insulation, find the total equivalent temperature differential 34 deg. The overall heat transmission coefficient for summer is taken from Table 16 and is found to be 0.13. The heat flow rate equals 34 × 0.13 = 4.42 Btu per (hr) (sq ft).

Example 7: For the conditions of Example 6, find the rate of heat flow into building at 2:00 P.M. during July for design temperatures of 105 F (outdoor) and 78 F (indoor). Daily range of temperature 30 degrees, i.e., outdoor temperature minimum of 75 F which occurs at 4:00 or 5:00 A.M.; this being 30 deg less than the maximum.

Solution: Make correction in equivalent temperature differential in accordance with Note 5 in Table 14 as follows:

The correction for 27 deg design temperature difference is $(27 - 15) = +12$

The correction for 30 deg daily range is $\left(-\frac{30 - 20}{2}\right) = -5$

Net total correction is $+12 - 5 = +7$

The heat flow rate at 2:00 P.M. therefore is $(34 + 7) \times 0.13 = 5.32$ Btu per (hr) (sq ft).

A method of determining *heat flow rates,* when structure is not given in Tables 14 or 15 is illustrated in *Example 8.*

Example 8: A 4 in. stone concrete roof covered with an average depth of 4 in. cinder concrete ($k = 4.9$) on which is placed a ⅜ in. thick felt roof with ½ in. pitch and slag surface is exposed to the sun. The location is the central part of the United States. Design temperatures are: outdoor 95 F; daily range 20 deg; indoor temperature 80 F. Find the heat flow rate at 2:00 P.M. for a day in July.

Solution: For the purpose of selecting the *equivalent temperature differential,* this construction is assumed to be equal approximately to an uninsulated 6 in. concrete roof for which the equivalent temperature is found to be 38 deg in the 2:00 P.M. column of Table 14. Calculate the overall heat transmission coefficient U of the roof as follows:

$$U = \frac{1}{\frac{1}{1.2} + \frac{4}{12} + \frac{4}{4.9} + \frac{0.375}{1.33} + \frac{0.50}{1.00} + \frac{1}{4.0}} = 0.33.$$

The heat flow rate is then 38 × 0.33 equals 12.5 Btu per (hr) (sq ft).

GLASS AREAS—DESIGN TABLES

In order to clearly set forth the principles involved in calculating heat transfer through glass areas the general instantaneous heat-balance relation will be presented. Fig. 5 shows this schematically. The net heat gain for the indoor space is the result of several contributing phenomena.

In discussing the heat gain through glass, there are three dimensionless quantities which require definition:

a = Absorptivity of glass, or fraction of incident radiation intensity which is absorbed within the glass itself.

τ = Transmissivity of glass, or fraction of incident radiation intensity which is transmitted through the glass.

r = Reflectivity of glass, or fraction of incident radiation intensity which is reflected at the surface.

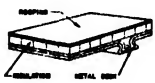
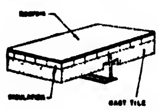
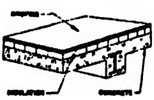
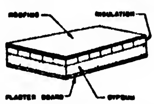
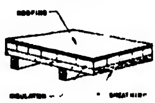
It is necessary that

$$a + \tau + r = 1 \tag{9}$$

These quantities vary with wave length and angle of incidence, primarily; and they are determined by the properties and thickness of the glass material concerned. Data are obtainable from glass manufacturers for their various products. The dependence of these quantities upon wave

Table 16. Summer Coefficients of Heat Transmission U of Flat Roofs Covered With Built-Up Roofing[a]

Btu per (hour) (square foot) (F deg difference between the air on the two sides)

Type of Roof Deck Ceiling not shown	Thickness of Roof Deck (Inches)		Insulation on Top of Deck (Covered With Built-Up Roofing)									
			No Ceiling—Underside of Roof Exposed					Furred Ceiling with Air Space, Metal Lath and Plaster				
			No Insulation	Insulating Board[d] Thickness, In.				No Insulation	Insulating Board[d] Thickness, In.			
				½	1	1½	2		½	1	1½	2
Flat Metal Roof Deck 	4 Ply Felt Roof		0.73	0.35	0.23	0.17	0.13	0.40	0.25	0.18	0.14	0.12
	Ditto + ½ in. Slag		0.54	0.30	0.20	0.16	0.13	0.34	0.22	0.16	0.13	0.11
Precast Cement Tile 	4 Ply Felt Roof	1½	0.67	0.33	0.22	0.17	0.13	0.38	0.24	0.18	0.14	0.12
	Ditto + ½ in. Slag	1½	0.50	0.28	0.20	0.15	0.12	0.32	0.21	0.17	0.13	0.11
Concrete 	4 Ply Felt Roof	2 4 6	0.65 0.59 0.54	0.33 0.31 0.30	0.22 0.21 0.20	0.16 0.16 0.16	0.13 0.13 0.13	0.37 0.36 0.33	0.24 0.23 0.22	0.18 0.17 0.17	0.14 0.13 0.13	0.12 0.12 0.11
	Ditto + ½ in. Slag	2 4 6	0.49 0.46 0.42	0.28 0.27 0.26	0.20 0.19 0.19	0.15 0.15 0.14	0.12 0.12 0.12	0.31 0.30 0.29	0.21 0.21 0.20	0.16 0.16 0.16	0.13 0.13 0.13	0.11 0.11 0.10
Gypsum and Wood Fiber[b] on ½" Gypsum Board 	4 Ply Felt Roof	2½ 3½	0.34 0.28	0.23 0.20	0.17 0.15	0.13 0.12	0.12 0.11	0.25 0.21	0.18 0.16	0.14 0.13	0.12 0.11	0.097 0.094
	Ditto + ½ in. Slag	2½ 3½	0.29 0.25	0.20 0.18	0.16 0.14	0.13 0.12	0.11 0.10	0.22 0.19	0.16 0.15	0.13 0.13	0.11 0.10	0.093 0.090
Wood[c] 	4 Ply Felt Roof	1 1½ 2 3	0.43 0.33 0.29 0.22	0.26 0.22 0.20 0.16	0.19 0.17 0.16 0.13	0.15 0.13 0.13 0.11	0.12 0.11 0.11 0.09	0.29 0.24 0.22 0.17	0.20 0.18 0.16 0.13	0.15 0.14 0.13 0.12	0.13 0.12 0.11 0.10	0.11 0.097 0.094 0.085
	Ditto + ½ in. Slag	1 1½ 2 3	0.35 0.29 0.26 0.20	0.23 0.20 0.19 0.15	0.17 0.15 0.14 0.12	0.14 0.12 0.12 0.10	0.11 0.10 0.10 0.09	0.25 0.21 0.20 0.16	0.18 0.17 0.15 0.13	0.14 0.13 0.13 0.11	0.12 0.11 0.10 0.09	0.10 0.093 0.090 0.081

[a] The summer coefficients are considered temporary and have been calculated with an outdoor wind velocity of 8 mph. For summer an inside surface conductance of 1.2 has been used instead of the regular 1.65 value. In all of these roofs a 4 ply felt roof has been assumed ⅜ in. thick, thermal conductivity = 1.33. Pitch and slag has been assumed as an additional thickness of ½ in. and has been assigned thermal conductivity = 1.0. In both cases thermal conductivity refers to one *inch* thickness.

[b] 87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board. This is a poured roof.

[c] Nominal thickness of wood is specified but actual thickness was used in calculations.

[d] If corkboard insulation is used the coefficient U may be decreased 10 per cent.

length has important practical consequences; for example, common window glass transmits a large portion of incident *solar* radiation, whereas it transmits outward only a very small portion of the indoor radiation

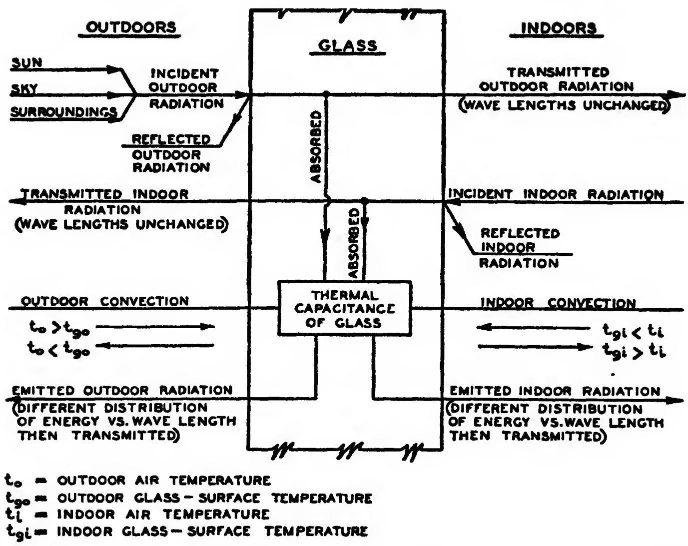


FIG. 5. INSTANTANEOUS HEAT-BALANCE CONDITIONS ON A GLASS SECTION

falling upon the inside surface because this latter is principally in the longer wave lengths.

A recent analysis has served to demonstrate the effects of the various controlling variables upon the heat gain from glass in relation to air-conditioning problems[12]. The practical results for a single sheet of common window glass will be cited here. For average conditions, the design equation for the instantaneous rate of heat transfer from the indoor glass surface to the conditioned space (for single sheets of common window glass only) in Btu per (hour) (square foot of sunlit surface) was found to be:

$$1.04\,(t_o - t_i) + 0.022\,I_d + 0.0165\,I_s + \tau_d I_d + 0.778\,I_s. \qquad (10)$$

where t_o Outdoor air temperature, Fahrenheit degrees.

t_i = Indoor air temperature, Fahrenheit degrees.

I_d = Intensity of solar or direct radiation on the outer glass surface, Btu per (hour) (square foot of sunlit surface).

I_s = Intensity of sky radiation on the outer glass surface.

τ_d = Transmissivity of the glass for solar radiation, dimensionless, given as a function of the angle of incidence in Table 17.

Similar equations with other numerical constants apply to other types of

TABLE 17. TRANSMISSIVITY OF SINGLE-SHEET COMMON WINDOW GLASS FOR DIRECT SOLAR RADIATION

Angle of Incidence[a] Deg	Transmissivity τ_d	Angle of Incidence[a] Deg	Transmissivity τ_d	Angle of Incidence[a] Deg	Transmissivity τ_d
0	0.87	50	0.83	80	0.41
20	0.86	60	0.77	90	0
40	0.85	70	0.65		

[a] Of Direct Solar Radiation.

glass or glass areas in other arrangements. (Heat-absorbing glass or glass in multiple layers, for example).

Equation 10 may be clarified further by explanations of the numerical constants involved:

1.04 = Over-all unit conductance for heat transfer under *summer* conditions, Btu per (hour) (square foot) (Fahrenheit degree). (Note that this is the transmittance U taken at 1.13 for winter conditions in Chapter 6.)

0.022 = Fraction of I_d which is absorbed and then transferred to the indoor space from the indoor glass surface, dimensionless. (Note that, of the total absorption, part goes inside and part goes outside.)

0.0165 = Fraction of I_s which is absorbed and then transferred to the indoor space from the indoor glass surface, dimensionless.

0.778 = transmissivity of the glass for sky radiation, dimensionless.

Magnitudes of the quantities I_d, I_s and t_o entering into Equation 10 would depend upon the time of the day, time of the year, atmospheric conditions, latitude of the receiving surface, and the orientation of the receiving surface. Principles and data needed for the calculation of I_d and I_s have been given previously in this chapter. Weather data, such as the sol-air temperatures in Tables 10 and 11, give data on t_o throughout a design day.

Table 18 has been prepared to expedite practical calculations of the heat gain through glass areas; tabulated values are magnitudes of the sum $(0.022\ I_d + 0.0165\ I_s + \tau_d I_d + 0.778\ I_s)$ from Equation 10 for a solar declination of 18 deg. This declination corresponds to a nominal August 1 design day, although the values tabulated may be used with safety to represent average conditions from July 15th to August 15th.

It is possible to use the heat-gain data of Table 18 for other single-thickness glass materials than common window glass through the introduction of approximate correcting factors. Table 19 gives such factors. The range of transmissivities in Table 19 extends from high-transmission crystal-like glass to low-transmission heat-absorbing glass. The transmissivity of 0.87 is that of common window glass.

By using two or more layers of glass separated by air spaces, the absorptivity is increased and the transmissivity decreased. Values for some combinations appear in the literature[12]. A rough approximation which holds fairly well until the angle of incident radiation becomes large is that the transmissivity of the combination is the product of the transmissivities of the component layers, while the absorptivity of the combination is the absorptivity of the outer glass plus the product of the absorptivity of the inner glass and the transmissivity of the outer.

Heat-gain quantities from Table 18 may also be used for other times of the year than August 1 if a correction is made for solar declination. Solar heating calculation for glass areas can also be based upon these data. Table 20 gives solar declinations for various times of the year. The rule for procedure is the following:

To find the radiation heat gain for some declination of the sun other than 18 deg use the latitude in Table 18 equal to that of the locality in question plus (18 deg— solar declination involved).

For example, consider that the instantaneous heat gain is required for Philadelphia in the middle of June. The latitude of Philadelphia is 40 deg North. The solar declination in mid-June is about 23.5 deg. The section of Table 18 to be used is then that for a latitude of $40 + (18 - 23.5) = 34.5$ deg; say 35 deg.

Caution: This method gives only approximate values and its use should be limited. It may be used for Eastern and Western exposures for all hours 8:00 a.m. to 4:00 p.m.

Table 18. Instantaneous Rates of Heat Gain Due to Solar and Sky Radiation for Single Sheets of Unshaded Common Window Glass[a, b]

Computed for Solar Declination of 18 Deg—August 1

Note: *To determine the total instantaneous rate of heat gain add the term 1.04* (t_o-i *to the values shown in the table (see Equation 10).*

Sun Time	Solar Altitude β Deg	Instantaneous Rate of Heat Gain, Btu per Hour for Each Square Foot of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal
						25 Deg North Latitude				
6 a.m.	7.5	19	77	84	40	3	3	3	3	12
7	20.5	26	146	173	98	10	10	10	10	68
8	34.0	18	148	193	121	13	13	13	13	143
9	47.5	15	118	172	120	16	15	15	15	205
10	61.5	16	67	124	95	22	16	16	16	252
11	74.5	16	25	57	57	25	16	16	16	282
12	83.0	16	16	16	25	28	25	16	16	293
1 p.m.	74.5	16	16	16	16	25	57	57	25	282
2	61.5	16	16	16	16	22	95	124	67	252
3	47.5	15	15	15	15	16	120	172	118	205
4	34.0	18	13	13	13	13	121	193	148	143
5	20.5	26	10	10	10	10	98	173	146	68
6	7.5	19	3	3	3	3	40	84	77	12
						30 Deg North Latitude				
6 a.m.	9.0	22	88	97	47	4	4	4	4	15
7	21.5	23	146	176	105	11	11	11	11	74
8	34.5	16	140	194	130	14	14	14	14	144
9	47.5	15	104	171	133	20	15	15	15	205
10	60.0	16	53	126	112	33	16	16	16	248
11	72.0	16	19	56	74	42	16	16	16	277
12	78.0	16	16	16	34	45	34	16	16	288
1 p.m.	72.0	16	16	16	16	42	74	56	19	277
2	60.0	16	16	16	16	33	112	126	53	248
3	47.5	15	15	15	15	20	133	171	104	205
4	34.5	16	14	14	14	14	130	194	140	144
5	21.5	23	11	11	11	11	105	176	146	74
6	9.0	22	4	4	4	4	47	97	88	15
						35 Deg North Latitude				
6 a.m.	10.0	21	97	109	53	4	4	4	4	19
7	22.5	19	143	179	110	11	11	11	11	79
8	34.5	14	133	194	140	15	14	14	14	144
9	46.5	15	90	170	144	28	15	15	15	200
10	58.5	16	38	126	128	47	16	16	16	243
11	68.5	16	16	56	91	62	17	16	16	269
12	73.0	16	16	16	45	68	45	16	16	279
1 p.m.	68.5	16	16	16	17	62	91	56	16	269
2	58.5	16	16	16	16	47	128	126	38	243
3	46.5	15	15	15	15	28	144	170	90	200
4	34.5	14	14	14	14	15	140	194	133	144
5	22.5	19	11	11	11	11	110	179	143	79
6	10.0	21	4	4	4	4	53	109	97	19
						40 Deg North Latitude				
5 a.m.	1.5	7	18	17	6	1	1	1	1	2
6	11.5	23	106	120	62	5	5	5	5	24
7	23.0	15	141	181	118	11	11	11	11	82
8	34.5	14	122	194	147	19	14	14	14	145
9	45.5	15	76	172	156	42	15	15	15	196
10	56.0	16	30	125	144	66	16	16	16	235
11	64.5	16	16	53	110	85	22	16	16	261
12	68.0	16	16	16	62	94	62	16	16	269
1 p.m.	64.5	16	16	16	22	85	110	52	16	261
2	56.0	16	16	16	16	66	144	125	30	235
3	45.5	15	15	15	15	42	156	172	76	196
4	34.5	14	14	14	14	19	147	194	122	145
5	23.0	15	11	11	11	11	118	181	141	82
6	11.5	23	5	5	5	5	62	120	106	24
7	1.5	7	1	1	1	1	6	17	18	2

TABLE 18. INSTANTANEOUS RATES OF HEAT GAIN DUE TO SOLAR AND SKY RADIATION FOR SINGLE SHEETS OF UNSHADED COMMON WINDOW GLASS[a, b] (CONCLUDED)

45 Deg North Latitude

Sun Time	Solar Altitude β Deg	Instantaneous Rate of Heat Gain, Btu per Hour for Each Square Foot of Unshaded Glass: N	NE	E	SE	S	SW	W	NW	Horizontal
5 a.m.	2.0	9	23	23	8	.1	1	1	1	2
6	12.5	22	111	129	68	6	6	6	6	28
7	23.0	13	135	182	121	11	11	11	11	82
8	33.5	14	116	192	153	24	14	14	14	141
9	44.0	15	63	168	166	55	15	15	15	189
10	53.0	16	22	123	154	88	16	16	16	225
11	60.0	16	16	56	127	113	30	16	16	249
12	63.0	16	16	16	76	119	76	16	16	256
1 p.m.	60.0	16	16	16	30	113	127	56	16	249
2	53.0	16	16	16	16	88	154	123	22	225
3	44.0	15	15	15	15	55	166	168	63	189
4	33.5	14	14	14	14	24	153	192	116	141
5	23.0	13	11	11	11	11	121	182	135	82
6	12.5	22	6	6	6	6	68	129	111	28
7	2.0	9	1	1	1	1	8	23	23	2

50 Deg North Latitude

Sun Time	Solar Altitude β Deg	N	NE	E	SE	S	SW	W	NW	Horizontal
5 a.m.	4.5	18	48	48	17	2	2	2	2	5
6	13.5	22	119	139	75	6	6	6	6	32
7	23.5	11	131	183	127	11	11	11	11	84
8	33.0	13	103	190	161	30	13	13	13	138
9	42.0	14	51	165	173	69	14	14	14	179
10	50.0	16	19	122	166	109	16	16	16	214
11	56.0	16	16	55	138	133	41	16	16	235
12	58.0	16	16	16	92	140	92	16	16	242
1 p.m.	56.0	16	16	16	41	133	138	55	16	235
2	50.0	16	16	16	16	109	166	122	19	214
3	42.0	14	14	14	14	69	173	165	51	179
4	33.0	13	13	13	13	30	161	190	103	138
5	23.5	11	11	11	11	11	127	183	131	84
6	13.5	22	6	6	6	6	75	139	119	32
7	4.5	18	2	2	2	2	17	48	48	5

[a] Table compiled from solar radiation transmission data developed by A.S.H.V.E. Research Laboratory[2] and direct intensity values developed by Moon[1], recently enlarged and revised by the A.S.H.V.E. Research Laboratory.

[b] For relatively clear atmosphere at sea level. For hazy atmosphere values may be reduced 10 per cent. Above sea level add one per cent per 1000 ft elevation.

inclusive, and for all exposures from 10:00 a.m. to 2:00 p.m. inclusive. For Southern exposures at 8:00 a.m. or at 4:00 p.m. the error may be ±15 per cent. At noon this method gives accurate results; the error becomes progressively larger as the time differs from noon.

In regard to *glass blocks*, data on the over-all unit conductances for heat transfer by convection and conduction are given in Table 18, Chapter 6; these will serve also for cooling load calculations for shaded or north-

TABLE 19. RATIO OF INSTANTANEOUS RATE OF HEAT GAIN THROUGH SINGLE THICKNESS OF HEAT ABSORBING SUNLIT GLASS TO HEAT GAIN FOR COMMON GLASS AS GIVEN IN TABLE 18

Transmissivity[a]	Approximate Ratio	Transmissivity[a]	Approximate Ratio	Transmissivity[a]	Approximate Ratio
0.90	1.02	0.70	0.85	0.40	0.60
0.87	1.0	0.60	0.76	0.30	0.52
0.80	0.93	0.50	0.68	0.20	0.45

[a] For direct solar radiation at normal incidence.

TABLE 20. APPROXIMATE SOLAR DECLINATIONS IN DEGREES

Date	Declination	Date	Declination	Date	Declination
April 1	4.5	June 1	22.0	Aug. 1	18.0
April 15	10.0	June 15	23.5	Aug. 15	14.0
May 1	15.0	July 1	23.0	Sept. 1	8.5
May 15	19.0	July 15	21.5	Sept. 15	3.0

exposure walls. (Steady-state calculations as suggested there take no account of the time lag present with periodic cycles of air temperature.)

The results of experimental observations[13] on heat gains through glass-block sections are given in Table 21. Tabulated heat-gain rates are based upon a constant indoor-air temperature of 78 F and a maximum outdoor-air temperature of 95 F for the design day chosen. The values given are approximate, and intended for estimates only; they are *averages* for *four* representative glass block designs, two with smooth exterior faces and two with ribbed exterior faces.

Shading of Glass Areas—Design Tables

The effects and possibilities of shading should be carefully investigated whenever the heat gain from glass is a large portion of the cooling load.

Vertical glass which is not mounted in the plane of the building surface is partially shaded by the *setback*. If a vertical window of height l and width w be set back from the plane of the building a distance s, the fraction of the total area of the window which receives *direct* solar radiation is:

$$G_f = 1 - r_1 \tan \beta - r_2 \tan \gamma + r_1 r_2 \tan \beta \tan \gamma \tag{11}$$

where

$r_1 = s/l$, $r_2 = s/w$, β = solar altitude, and γ is the difference between the azimuth angles of the wall in question and of the horizontal projection of the sun's rays. (γ is always equal to or less than 90 deg.)

Solar altitudes are given in Table 18 for various latitudes and an August

TABLE 21. TOTAL INSTANTANEOUS RATES OF HEAT GAIN FOR GLASS BLOCKS[14] ON DESIGN DAY OF AUGUST 1

(*Values tabulated include solar and sky radiation, convection, and conduction averaged for four types of blocks*)

Mean Sun Time	Total Instantaneous Rate of Heat Gain, Btu per Hour for Each Square Foot of Sunlit Glass Block — Vertical Surface Facing					
	East	West	South			
North Latitude, Deg	30 to 45	30 to 45	30	35	40	45
7 a.m.	61	......	−4.5	−2.0	−0.5	1.0
8	78	......	0.0	2.0	4.0	5.0
9	74	5.0	5.0	7.0	10	12
10	58	6.5	11	15	18	21
11	45	7.5	17	22	26	32
12 noon	37	11	22	28	34	41
1 p.m.	30	22	25	32	39	46
2	24	35	26	32	39	47
3	20	55	24	30	37	45
4	16	77	20	26	32	41
5	13	86	15	20	25	34
6	11	55	9.5	14	18	26
7	8	19	3.5	7.0	11	18

Table 22. Values of the Azimuth-Difference Angle, γ, Degrees, for Window-Reveal Shading Calculations (See Equation 11)
Computed for solar declination of 18 deg—August 1

N. Latitude Degrees	Mean Sun Time	Window Orientation						
		NE	E	SE	S	SW	W	NW
30	6 a.m.	61	74	29	Shade	Shade	Shade	Shade
	7	54	81	36	....	....	....	....
	8	47	88	43	....	....	....	....
	9	39	84	51	6	....	....	....
	10	28	73	62	17	....	....	....
	11	7	52	83	38	....	....	....
	12 N	Shade	0	45	90	45	0	....
	1 p.m.	....	Shade	Shade	38	83	52	7
	2	....	....	....	17	62	73	28
	3	....	....	....	6	51	84	39
	4	....	....	....	Shade	43	88	47
	5	....	....	....	....	36	81	54
	6	....	....	....	....	29	74	61
40	6 a.m.	59	76	31	Shade	Shade	Shade	Shade
	7	50	85	40	....	....	....	....
	8	40	85	50	5	....	....	....
	9	27	74	61	16	....	....	....
	10	14	59	76	31	....	....	....
	11	Shade	35	80	55	....	....	....
	12 N	....	0	45	90	45	0	....
	1 p.m.	....	Shade	10	55	80	35	....
	2		....	Shade	31	76	59	14
	3	....	....	....	16	61	74	27
	4	....	....	....	5	50	85	40
	5	....	....	....	Shade	40	85	50
	6		....	....	....	31	76	59
50	6 a.m.	57	78	33	Shade	Shade	Shade	Shade
	7	45	90	45	0	....	....	....
	8	33	78	57	12	....	....	....
	9	20	65	70	25	....	....	....
	10	3	48	87	42	....	....	....
	11	Shade	26	71	65	19	....	....
	12 N	....	0	45	90	45	0	....
	1 p.m.		Shade	19	65	71	26	....
	2		....	Shade	42	87	48	3
	3		....	....	25	70	65	20
	4			....	12	57	78	33
	5	....	....	....	0	45	90	45
	6	....	....	....	Shade	33	78	57

1 design day. Values of the angle γ are given in Table 22 for the August 1 design day. Special cases not covered by the tabulated data may be solved analytically[14]; however, the design conditions chosen will yield a satisfactory approximation if used without correction for estimates at any time during the summer period.

Example 9. Estimate the instantaneous rate of sky and solar radiation heat gain from a west window 3 ft wide by 5 ft high with a setback of 6 in. for August 1 and a north latitude of 40 deg at 3 p. m. (sun time).

From Table 18, the instantaneous rate of sky and solar radiation heat gain per square foot of sunlit glass is 172 Btu per hour. From the same table, the solar altitude is 45.5 deg. From Table 22, the angle γ is 16 deg, from Equation 11, the fraction of the total window area that is receiving direct solar radiation is:

$$G_t = 1 - 0.1 \tan 45.5 - 0.167 \tan 16 + 0.0167 \tan 45.5 \tan 16$$
$$= 1 - 0.102 - 0.048 + 0.005 = 0.855.$$

Although the *sky* radiation is not reduced in proportion to the shaded portion, since the sky radiation is small this fact may be neglected, and hence instantaneous sky- and solar-radiation heat gain for this window is:

$$q = 3 \times 5 \times 0.855(172) = 2206 \text{ Btu per hour.}$$

A window such as the one in Example 9 above would customarily be provided with an additional shading means for use particularly when directly

sunlit. Conventional shading devices include *awnings*, *shades*, and *screens* of various types.

Experimental work conducted at the A.S.H.V.E. Research Laboratory[14] and other research[15] to determine the effectiveness of various types of window shades have been used as the basis for the recommended ratios in column 3 of Table 23. A study of absorptivity of the shade to solar radiation and heat transfer from the shade to the outdoors and indoors was used to determine these ratios.

There are a number of variables affecting these ratios such as color, fit, solar altitude, and angle of incidence of the solar radiation. These values, therefore, must be considered as approximate, only, and will have to be used with considerable judgment. An inside shade is effective to the extent of its reflectivity, since the portion of the solar radiation directly transmitted by the glass that is absorbed by the shade is transferred by

Table 23. Effect of Shading upon Instantaneous Solar Heat Gain Through Single Thickness of Common Window Glass

Type of Shading	Finish on Side Exposed to Sun	Fraction of Gain Through Unshaded Window
Canvas Awning	Dark	0.25–0.35
Inside Roller Shade, Fully Drawn[a]	White	0.45
Inside Roller Shade, Fully Drawn[a]	Medium color	0.63
Inside Roller Shade, Fully Drawn[a]	Dark color	0.80
Inside Roller Shade, Half Drawn[a]	White	0.72
Inside Roller Shade, Half Drawn[a]	Medium color	0.81
Inside Roller Shade, Half Drawn[a]	Dark color	0.90
Inside Venetian Blind, Slats set at 45 deg[b]	White	0.62
Inside Venetian Blind, Slats set at 45 deg[b]	Medium	0.74
Inside Venetian Blind, Slats set at 45 deg[b]	Aluminum	0.70
Inside Venetian Blind, Slats set at 45 deg[b]	Dark color	0.86
Outside Venetian Blind, Slats set at 45 deg[b]	Cream	0.30
Outside Venetian Blind, Slats set at 45 deg, extended as an awning[c]	Any color	0.40
Outside Shading Screen[d], solar altitude 0–20 deg	Dark color	0.75–0.43
Outside Shading Screen[d], solar altitude 20–40 deg	Dark color	0.43–0.22
Outside Shading Screen[d], solar altitude, above 40 deg	Dark color	0.22

[a] Roller shades are assumed to be opaque. Some white shades may transmit considerable solar radiation. For white translucent shades fully drawn use 0.55 and for half drawn use 0.77.

[b] Venetian Blinds are fully drawn and cover window. It is assumed that the occupant will adjust slats to prevent direct rays from passing between slats

[c] Commercial shade with wide slats.

[d] Metal slats 0.05 inches wide spaced 0.063 inches apart and set at 17 degree angle with horizontal. At solar altitudes below 38 deg some direct solar rays are allowed to pass between slats, and this amount becomes progressively greater at low solar altitudes.

convection to the room air and by radiation to the solid surfaces of the room.

Instantaneous Heat Gains vs. Instantaneous Cooling Loads

The difference between instantaneous heat gain and instantaneous cooling load has been mentioned previously; its practical importance is sufficient to warrant further consideration.

Fig. 6 offers a simplified schematic illustration showing how the *radiative* part of the instantaneous heat gain is first absorbed by solid objects and is not encountered by the conditioning equipment as a cooling load until some later time when it finally appears in the air-stream entering the equipment. While it is true that some lag also is inherent in convective heat transfer and the time required to change the air in the conditioned space, this is usually of the order of a few minutes to perhaps half an hour. Heat storage in the interior furnishings and structure increases according

to the proportion of the instantaneous heat gain which is in the form of radiation, and also as the thermal capacitance of the objects and materials involved is increased.

Constituents of the total instantaneous heat gain which have appreciable radiation components include those due to: glass areas, exposed walls and roofs, lighting, appliances, and people.

No comprehensive data are presently available for use in design load estimates to evaluate the interior load-lag effect, but several investigators[7, 15], have made a study of the problem and have presented much useful data. Tables 14, 15 and 18 are all based on instantaneous rates of heat transfer. Hence, practical judgment and experience offer the only basis of procedure. Until the needed data become available, it is recommended that the non-continuous load be averaged over two or three hours during the time of maximum load, when determining the total instantaneous cooling load where a large portion of the heat gain is radiant. This suggestion applies only to conditions near the time of maximum heat gain, as the heat stored within the structure would necessarily appear in the cooling load eventually, but if it appears at a time when the gain from outside

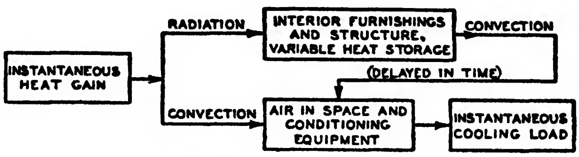


Fig. 6. Origin of the Difference Between the Magnitudes of the Instantaneous Heat Gain and Instantaneous Cooling Load

The radiation absorbed by the interior furnishings and structure reaches the conditioning equipment after a considerable delay in time.

is relatively low the equipment will be able to maintain satisfactory conditions within the range of maximum capacity.

LOAD FROM INTERIOR PARTITIONS, CEILINGS, AND FLOORS

Whenever a conditioned space is adjacent to another space in which a different temperature prevails, the transfer of heat through the separating structural section must be considered. Calculations are made according to the relation

$$q = U_i A_i (t_b - t_i) \text{ Btu per hour.} \tag{12}$$

where U_i = Coefficient of over-all heat transfer between the adjacent and the conditioned space, Btu per (hour) (square foot) (Fahrenheit degree).

A_i = Area of separating section concerned, square feet.

t_b = Air temperature in adjacent space, Fahrenheit degrees.

t_i = Air temperature in conditioned space, Fahrenheit degrees.

Magnitudes of U_i may be obtained from Chapter 6. The temperature t_x may have any value over a considerable range, according to conditions in the adjacent space. The temperature in a kitchen or boiler room may be as much as 15 to 50 deg above the outdoor air temperature. It is recommended that actual temperatures be measured in adjoining spaces wherever practicable. Where nothing is known except that the adjacent space is of conventional construction and contains no heat sources, it is recommended that the difference $(t_b - t_i)$ be taken as the difference between the out-

door-air and conditioned-space design dry-bulb temperatures *minus* 5 Fahrenheit degrees. In some cases it may be that the air temperature in the adjacent space will correspond closely to the outdoor air temperature at all times. Under these latter conditions, the heat gain through the partition will be periodic in nature and the value of a shaded wall should be used from Table 15.

For floors directly in contact with the ground, or over an underground basement that is neither ventilated nor warmed, the heat transfer may be neglected for cooling-load estimates.

LOAD FROM OUTSIDE AIR-VENTILATION AND INFILTRATION

Ventilation. Data for determining the necessary ventilation rate have been presented previously in this chapter. Ventilation required is primarily dependent upon the number of occupants and upon the materials and apparatus within the space which may give off odors. For spaces having ceiling heights 10 ft or less, the total requirement should be checked against the volume, and in no case should the ventilation air rate be less than one air change per hour. In spaces having ceilings higher than 10 ft where the occupant load is low, a check calculation can be made against the volume of the space below an assumed 10 ft ceiling.

Infiltration must never be counted upon to provide ventilation because on still days there will be little or no infiltration.

Infiltration. The principles of infiltration calculations have been discussed in Chapter 8 and 14, with emphasis on the heating season. For the cooling season, infiltration calculations are usually limited to doors and windows.

To compute cooling-load infiltration for windows by the crack method use the data of Table 2, Chapter 8, for a wind velocity of 10 mph. Note that for double-hung windows the length of crack is three times the width plus twice the height; while for metal-sash windows the crack length is the total perimeter of the movable or ventilating sections. In calculating window infiltration for an entire structure, it is not necessary to consider the total crack length on all sides of the building, for the wind would not act simultaneously on all sides at once. In no case, however, should less than half of the total crack length be figured. A knowledge of the prevailing wind direction will aid judgment in this consideration.

Cooling-load infiltration for doors[16] may be obtained from Table 3, Chapter 8. For conditions other than those covered, the notes appended to Table 3 will provide a basis for estimates. The tabulated data may also be used as the basis of estimates for interior doors between an air-conditioned and a non-air-conditioned space.

Infiltration load must be included whenever the new air introduced through the system is not sufficient to maintain excess pressure within the enclosure to prevent the infiltration. Whenever economically feasible it is desirable to introduce sufficient outdoor air through the air-conditioning equipment to maintain a constant outward escape of air and thus eliminate the infiltration portion of the load. The pressure maintained must, of course, be sufficient to overcome wind pressure through cracks and door openings. When this condition prevails it is not necessary to include any infiltration load. When the quantity of new air introduced through the cooling equipment is not sufficient to build up the required pressure to offset infiltration, the entire infiltration load should be included in the cooling load calculations.

Total Outside Air Load. To determine the design cooling load caused by the introduction of outside air, the maximum rate of outside-air entry is first established. In some applications the use of special exhausters from the conditioned space may add to the outdoor-air requirements in determining the maximum rate. Once this design quantity is established, and with the design indoor and outdoor air states known, the cooling load may be computed. There are several methods in use, the more accurate of these require rather detailed calculations. Refer to Chapter 3, under section on Cooling Load and Chapter 43 in section on Apparatus Dew Point. The following equations are considered to be sufficiently accurate for use at usual design conditions as their accuracy is within 1 per cent.

$$\textit{Sensible Load}\quad q_s = Q \times 60 \times 0.244 \times 0.075\left(1 - \frac{0.0\text{[illegible]}3}{0.62}\right)(t_o - t_i) \tag{13}$$

$$= Q \times 1.08\,(t_o - t_i),\ \text{Btu per hour}$$

$$\textit{Latent Load}\quad q_e = Q \times 60 \times 0.075 \times 1076\,(W_o - W_i)$$

$$= Q \times 4840\,(W_o - W_i),\ \text{Btu per hour} \tag{14}$$

$$\textit{Total Load}\quad q_t = q_s + q_e \tag{15}$$

where Q = Rate of entry of outside air, cubic feet per minute.

t_o = Outdoor dry-bulb temperature, Fahrenheit degrees.

t_i = Indoor dry-bulb temperature, Fahrenheit degrees.

W_o = Outdoor humidity ratio, pounds moisture per pound of dry air.

W_i = Indoor humidity ratio, pounds moisture per pound of dry air.

0.075 = Standard air density, pounds per cubic foot.

0.244 = A constant approximating the specific heat of dry air corrected for moisture Btu per (pound) (Fahrenheit degree).

1076 = A factor approximating the average Btu released in condensing one pound of water vapor from air.

Standard air weight (0.075 lb per cu ft) is recommended for use in all calculations as this is the basis for rating fans and its consistent use keeps all parts of the calculations in conformity.

HOW OUTSIDE AIR LOAD AFFECTS ROOM LOAD

Actually the outdoor air used for ventilation would pass through the conditioning equipment and be cooled and dehumidified to a lower temperature and humidity ratio than room conditions before entering the room; but for heat-balance purposes the cooling load chargeable to the outdoor air is that corresponding to the difference between the outdoor and indoor air conditions.

One important purpose of the cooling load estimate is to determine the conditions and quantity of air supplied to the space. All the various sensible and latent heat loads within the space must be included. Infiltration must be included in the space load since this air enters the doors and windows and its heat and moisture load must be offset by the introduction of cooler, dryer air to the space. However since ventilation air is taken through the conditioning equipment and cooled, this portion does not become a part of the space load. To determine the total load on the refrigeration machine, the ventilation air load must be included in the grand total load.

Example 10. For outdoor design conditions of 95 F dry-bulb and 75 F wet-bulb and indoor design conditions of 80 F dry-bulb and 67 F wet-bulb and for the supply

of outdoor air at the rate of 1000 cfm and the exhaust of room air at the corresponding rate, calculate the total, sensible and latent heat gains.

Solution: Substituting in Equation 13:

$$q_s = 1000 \times 1.08\ (95 - 80) = 16{,}200 \text{ Btu per hr.}$$

From psychrometric data $W_o = 0.01413$, $W_i = 0.01122$.

Substituting in Equations 14 and 15.

$$q_e = 1000 \times 4840\ (0.01413 - 0.01122) = 14{,}100 \text{ Btu per hr.}$$

$$q_t = q_s + q_e = 30{,}300 \text{ Btu.}$$

Many cooling coil manufacturers publish tables giving psychrometric data based on the *average* conditions of the leaving air for various coil temperatures, air velocities, and entering dry-bulb and wet-bulb conditions. When these tables are used, it is necessary to calculate the mixed air condition entering the coil and determine from the tables what coil and air velocity will produce the desired leaving air conditions as required for

TABLE 24. RATES OF HEAT GAIN FROM OCCUPANTS OF CONDITIONED SPACES[a]

Degree of Activity	Typical Application	Total Heat Adults, Male Btu/Hr	Total Heat Adjusted[b] Btu/Hr	Sensible Heat Btu/Hr	Latent Heat Btu/Hr
Seated at Rest	Theater-Matinee	390	330	180	150
	Theater-Evening	390	350	195	155
Seated, Very Light Work	Offices, Hotels, Apartments	450	400	195	205
Moderately Active Office Work	Offices, Hotels, Apartments	475	450	200	250
Standing, Light Work; or Walking Slowly	Department Store, Retail Store Dime Store	550	450	200	250
Walking; Seated; Standing; Walking Slowly	Drug Store Bank	550	500	200	300
Sedentary Work	Restaurant[c]	490	550	220	330
Light Bench Work	Factory	800	750	220	530
Moderate Dancing	Dance Hall	900	850	245	605
Walking 3 mph; Moderately Heavy Work	Factory	1000	1000	300	700
Bowling[d]	Bowling Alley				
Heavy Work	Factory	1500	1450	465	985

[a] *Note:* Tabulated values are based on 80 F room dry-bulb temperature. For 78 F room dry-bulb, the total heat remains the same, but the sensible heat values should be increased by approximately 10 per cent and the latent heat values decreased accordingly.

[b] *Adjusted total heat gain* is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85 per cent of that for an adult male and that the gain from a child is 75 per cent of that for an adult male.

[c] Adjusted total heat value for *sedentary work, restaurant,* includes 60 Btu per hour for food per individual (30 Btu sensible and 30 Btu latent).

[d] For *bowling* figure one person per alley actually bowling and all others as sitting (400 Btu per hour) or standing (550 Btu per hour).

the space to be conditioned. When cooling coils are listed as 80 to 95 per cent efficient, the manufacturer indicates that 20 to 5 per cent of the air passes through the coil without being cooled. If data of this nature are used, the uncooled portion of the air must be added to the space load before determining the effective air quantity.

HEAT SOURCES WITHIN THE CONDITIONED SPACE

People. The rates at which heat and moisture are given off by human beings under different states of activity are given in Table 24. In many applications these sensible and latent heat gains become a large fraction of the total load. Appreciable variations in heat-emission rates must be recognized according to the age and sex of the individual, state of activity, environmental influences, and duration of occupancy (since for short

occupancy the extra heat and moisture brought in by people may be a significant factor).

While Chapter 12 should be referred to for detailed information, Table 24 in this chapter summarizes practical data representing conditions commonly encountered.

Lighting. In general, the instantaneous rate of heat gain from electric lighting[17] may be calculated from the following relation:

$$q_{el} = \begin{Bmatrix}\text{total light}\\ \text{wattage}\end{Bmatrix} \times \begin{Bmatrix}\text{use}\\ \text{factor}\end{Bmatrix} \times \begin{Bmatrix}\text{special allow-}\\ \text{ance factor}\end{Bmatrix} \times 3.41, \text{ Btu per hour} \qquad (16)$$

The *total light wattage* is obtained from the ratings of all fixtures installed, both for general illumination and for display use.

The *use factor* is the ratio of the wattage in use, for the conditions under which the load estimate is being made, to the total installed wattage. For commercial applications such as stores, the use factor would be unity.

The *special allowance factor* is introduced to care for fluorescent fixtures and for fixtures which are either ventilated or installed so that only part of their heat goes to the conditioned space. For *fluorescent fixtures,* the special allowance factor is recommended to be taken as 1.20 in order to allow for power consumed in the *ballast.* For *ventilated fixtures, recessed fixtures,* and the like, manufacturer's or other data[17] must be sought to establish the fraction of the total wattage which may be expected to enter the conditioned space.

Power. When equipment of any sort is operated within the conditioned space by electric motors, the heat equivalent of this operation must be considered in the cooling load. The general equation for calculating this load is:

$$q_{em} = \left(\frac{\text{Horsepower Rating}}{\text{Motor Efficiency}}\right) \times \left(\begin{matrix}\text{Load}\\ \text{Factor}\end{matrix}\right) \times 2544, \text{ Btu per hour.} \qquad (17)$$

It is assumed that both the motor and the driven equipment are within the conditioned space. If the motor is without the space, then *do not divide* by the motor efficiency in Equation 17. The *load factor* is merely the fraction of the rated load which is being delivered under the conditions of the cooling-load estimate. *Motor efficiencies* may be approximated as follows: about 50 to 60 per cent at $\frac{1}{8}$ hp rating, increasing to 80 per cent at 1 hp, and to 88 per cent at 10 hp and above.

Appliances. Care must be taken in a cooling-load estimate to properly account for the heat gain from all appliances, electrical, gas, or steam. Table 25 presents recommended data[18]. Note that the *maintaining rate* in Table 25 is the heat input required to maintain the appliance at the normal operating temperature even though it is not being used; *i.e.*, no coffee is being made, no toast is being made, no food is being cooked in the fry kettle, etc. The maintaining rate is useful in setting up a lower limit to the heat gain to a room from the appliance when in operation.

Experienced judgment must be used in the application of data given in Table 25. Consideration must be given to the heat contributed by appliances which are in use at the time of peak load. The quantity of heat will depend upon whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded allowing part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that, when they are properly hooded

TABLE 25. RATE OF HEAT GAIN FROM APPLIANCES WITHOUT HOODS[a, b]

Appliance	Capacity	Over-all Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating Watts	Btu/Hr	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour		
								Sensible	Latent	Total
Restaurant Electrical Appliances										
Coffee Brewer and Warmer	½ gal		M M	Brewer 660 w Warmer 90 w	600 90	2000 300	306	900 230	220 60	1120 290
Coffee Brewer Unit with Tank	½ gal	20 x 30 x 26		2000 w Water heater, 2960 w brewer	4960	17000		4800	1200	6000
Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	A A	Nickel plated Nickel plated	4500 5000	15000 17000	2600 3600	2200 3400	1500 2300	3700 5700
Doughnut Machine		22 x 22 x 57	A	Exhaust System	4700	16000		5000	0	5000
Egg Boiler	2 cups	10 x 13 x 25	M		1100	3750		1200	800	2000
Food Warmer, with Plate Warmer, per sq ft of top surface			A	Insulated, separate heat unit for each pot; plate warmer in base	400	1350	500	350	350	700
Food Warmer, alone, per sq ft of top surface			A		300	1000	400	200	350	550
Fry Kettle	11½ lb fat	12 (Diam.) x 14	A		2600	8900	1100	1600	2400	4000
Fry Kettle	25 lb fat	16 x 18 x 12	A	Area 12 x 14 in.	7000	24000	2000	3800	5700	9500
Griddle, Frying		18 x 18 x 8	A	Area 18 x 14 in.	2350	8000	2800	3100	1700	4800
Griddle, Frying		24 x 20 x 10	A	Area 23 x 18 in.	4000	13500	5000	5300	2900	8200
Grill, Meat		14 x 14 x 10	A	Area 10 x 12 in.	3000	10250	1900	3900	2100	6000
Grill, Sandwich		13 x 14 x 10	A	Area 12 x 12 in.	1650	5600	1900	2700	700	3400
Roll Warmer		23 x 23 x 29	A	Three drawers	1000	3400	900	2400	300	2700
Toaster, continuous	360 slices/hr	15 x 15 x 28	A	2 slices wide	2200	7500	5000	5100	1300	6400
Toaster, continuous	720 slices/hr	20 x 15 x 28	A	4 slices wide	3000	10250	6000	6100	2600	8700
Toaster, pop-up	216 slices/hr	12 x 11 x 9	A	4 slice	2450	8400	2000	4900	900	5800
Waffle Iron	20 waffles/hr	12 x 13 x 10	A	7 in. diam. waffle	750	2500	600	1100	750	1850

Restaurant Gas-Burning Appliances

Coffee Brewer and Warmer	½ gal		M M	Brewer Warmer		3400 500	 500	1350 400	350 100	1700 500
Coffee Brewer Unit with Tank	4½ gal Tank	19 x 30 x 26		4 Brewers and Tank				7200	1800	9000
Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	A A	Nickel plated Nickel plated			3400 4700	2500 3900	2500 3900	5000 7800
Food Warmer, per sq ft of top surface			M	Water bath		2000	900	850	430	1280
Fry Kettle	15 lb fat	12 x 20 x 18	A	Area 10 x 10		14250	3000	4200	2800	7000
Fry Kettle	28 lb fat	15 x 35 x 11	A	Area 11 x 16		24000	4500	7200	4800	12000
Grill		22 x 14 x 17	M	Insulated, Grill surface of 1.4 sq ft Top burner 22,000 Btu/hr Bottom burner 15,000 Btu/hr		37000		14400	3600	18000
Stoves, Short Order Open Top, per sq ft top Closed Top, per sq ft top Fry Top, per sq ft top			 M M M	 Ring type burners Ring type burners Tubular type burners		 14000 11000 12000		 4200 3300 3600	 4200 3300 3600	 8400 6600 7200
Toaster, Continous	360 slices/hr	15 x 15 x 28	A	2 slices wide		12000	10000	7700	3300	11000
Toaster, Continous	640 slices/hr	20 x 15 x 28	A	4 slices wide		20000	14000	12000	5000	17000

Restaurant Steam-Heated Appliances

Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	T T	Nickel plated Nickel plated				2400 3400	1600 2300	4000 5700
Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	M M	Nickel plated Nickel plated				2600 3700	2600 3700	5200 7400
Food Warmer, per sq ft of top surface			T					400	500	900
Food Warmer, per sq ft of top surface			M					450	1150	1600

[a] For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.

[b] When these appliances are hooded and provided with adequate exhaust, use 50 per cent of recommended rate of heat gain from unhooded appliances.

Table 25. Rate of Heat Gain From Appliances WITHOUT HOODS[a, b] (Concluded)

Appliance	Capacity	Over-all Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating Watts	Btu/Hr	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour		
								Sensible	Latent	Total
Miscellaneous Electrical Appliances										
Hair Dryer, Blower Type			M	Fan, 165 w; Low, 915 w; High, 1580 w	1580	5400		2300	400	2700
Hair Dryer, Helmet Type			M	Fan, 80 w; Low, 300 w; High, 710 w	705	2400		1870	330	2200
Permanent Wave Machine			M	60 heaters at 25 w each, 36 in normal use	1500	5000		850	150	1000
Neon Sign, per linear ft of tube				½ in. outside diam. ⅝ in. outside diam.				30 60		30 60
Sterilizer, Instrument			A	For physicians; thermostat cuts off 550 w before boiling	1100	3750		650	1200	1850
Miscellaneous Gas-Burning Appliances										
Burners, Laboratory Small Bunsen Small Bunsen Fishtail Fishtail Large Bunsen		 7/16 in. Barrel 7/16 in. Barrel 7/16 in. Barrel 7/16 in. Barrel 1½ in. Mouth	 M M M M M	 Manufactured Gas Natural Gas Manufactured Gas Natural Gas Adjustable orifice		 1800 3000 3500 5500 6000		 960 1680 1960 3080 3350	 240 420 490 770 850	 1200 2100 2450 3850 4200
Cigar Lighter			M	Continuous Flame		2500		900	100	1000
Hair Dryer, 5 helmets			A	Heater and fan blowing air to helmets		33000		15000	4000	19000
Stoves, Oven				Insulated, modern Not insulated		25000 25000	6000 8500	7200 9200	1800 2300	9000 11500

[a] For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.
[b] When these appliances are hooded and provided with adequate exhaust, use 50 per cent of recommended rate of heat gain from unhooded appliances.

with a positive fan exhaust system through the hood, 50 per cent of the heat will be carried away and 50 per cent dissipated in the space to be conditioned. The same effectiveness of the hood should be figured for both latent and sensible heat.

LOAD FROM MOISTURE TRANSFERRING THROUGH PERMEABLE BUILDING MATERIALS

The diffusion of moisture through all common building materials is a natural phenomenon which is always present to a greater or lesser degree.

The permeability values for various building materials are given in Table 19 of Chapter 6, together with an explanation of moisture transmission through these materials.

In the usual comfort air-conditioning application, it is common practice to neglect moisture transfer through walls, for the actual rate is quite small and the corresponding latent-heat load is hardly significant. So-called *vapor barriers* are frequently employed in modern construction for the purpose of keeping moisture transfer to a minimum, particularly because of the deteriorating and insulation-destroying effects of moisture.

Industrial jobs, on the other hand, frequently call for a low moisture content to be maintained in a conditioned space. Here the matter of moisture transfer cannot be neglected; indeed, it is quite possible to have the latent-heat load accompanying this transfer be of greater magnitude than any other latent-heat load. The equation for computing this load is:

$$\left(\frac{q_m}{A}\right) = \frac{\mu}{7000} \times \left(\begin{array}{c}\text{Vapor Pressure}\\ \text{Difference, In. Hg}\end{array}\right) \times 1076, \text{ Btu per (hr) (sq ft).} \qquad (18)$$

where μ = Permeability grains per (sq ft) (hr) (in. Hg)

7000 = Grains per pound.

The factor 1076 is defined in list of symbols at Equation 15. (Sensible cooling of the water vapor is included in the factor 1076.)

The only means of preventing moisture transfer is to use a vapor proof wall or to apply a special lining, which is vapor proof. All openings in moisture proof construction must be equipped with special gaskets to prevent entrance of moisture.

When moisture transfer contributes an appreciable part of the latent-heat load, it is recommended that estimates should be made intentionally liberal in order to avoid later difficulties with insufficient dehumidifying capacity. Storage spaces, for example, would require sufficient dehumidifying capacity to handle the moisture brought in with goods to be stored in addition to moisture leaking in subsequently.

MISCELLANEOUS HEAT LOADS

This designation is intended to cover the various small heat gains from exposed piping, ducts, work done by circulating fan, and unforeseen contingencies. Where sufficient data are available these various heat gains may be estimated individually. In the majority of cases, however, common practice is to lump these factors together and combine them with a *safety factor* according to the experience and judgment of the estimator. On this basis, a small safety factor is added to the calculated cooling load to compensate for miscellaneous effects. No rules can be given for this procedure as experience in air conditioning is indispensable for application of suitable safety factors.

REQUIRED AIR QUANTITY THROUGH CONDITIONING EQUIPMENT

The procedure for determining the required air quantity is based upon the thermodynamic principles of Chapter 3 and the use of the Mollier diagram supplied with THE GUIDE, or a psychrometric chart. Readers are advised to review these principles, paying particular attention to the illustrative examples of cooling load calculations, and to refer to the section on Apparatus Dew Point in Chapter 43.

Calculation of the cooling load for a conditioned space is equivalent to making, for the space, a heat balance in which all heat, moisture, and infiltration are treated as directly entering the space. As explained in the section, *Load from Outside Air—Ventilation and Infiltration*, the outside air load normally does not become a part of the space load because heat and moisture are removed in the air conditioner before this air gets into the conditioned space. The desired conditions are maintained by considering a certain quantity of air to be withdrawn from the space, passed through the conditioning equipment, and returned to the space with such a temperature and humidity ratio that its net effect will be to counterbalance or *remove* the given entering amounts of heat and water vapor. This quantity of *indoor air* which is considered to be circulated in this manner is called the *required air quantity* and its determination is normally part of every cooling-load estimate. The procedure is as follows:

1. Determine the total sensible and latent heat loads in Btu per hour for the space.

2. Compute the quantity called the heat-moisture ratio of the room load, q_w. Use the following equation:

$$q_w = \left(\frac{\text{Space sensible load} + \text{space latent load}}{\text{Space latent load}}\right) \times 1076 \text{ Btu/lb of moisture difference} \quad (19)$$

Note that the ratio (Space latent load) ÷ 1076 is the equivalent of the required rate of water-vapor removal, in pounds per hour. If the rate of water-vapor removal is known, it may be used directly in Equation 19.

3. Locate the state point of the room air (design wet-bulb and dry-bulb temperatures) on the Goff diagram. From this state point draw a line intersecting the saturation line, using the slope established by the protractor on the Goff diagram for the particular value of q_w prevailing. This line is the *condition line* for the process.

4. Read the temperature where the condition line from step 3 intersects the saturation line. This is called the *apparatus dewpoint*.

5. Compute the *required air quantity* from the relation

$$Q_{ra} = \frac{(\text{Space sensible load})}{1.08\left[\left(\begin{matrix}\text{Space}\\ \text{dry-bulb}\end{matrix}\right) - \left(\begin{matrix}\text{Apparatus}\\ \text{dewpoint}\end{matrix}\right)\right] \times \left(\begin{matrix}\text{Coil}\\ \text{efficiency}\end{matrix}\right)} \quad (20)$$

The magnitude of Q_{ra} is substantially the quantity, cfm, of cooled and dehumidified air for which the distribution system must be designed.

The numerical factor 1.08 is derived from the product 1 cfm × 60 min. × 0.244 × $0.075\left(1 - \frac{0.00923}{0.62}\right) = 1.08$, assuming an average supply air dewpoint of 55 F. Since standard air density (0.075) includes the weight of the water vapor it is desirable to reduce it to the basis of dry air by the last factor where 0.00923 = humidity ratio of air at 55 F dewpoint, and 0.62 = ratio of density of water vapor to dry air at same temperature and pressure. Refer to Chapter 25 for coil selection.

Note that the product [(Space dry-bulb) − (Apparatus dewpoint)] × (Coil efficiency) is equal to the dry-bulb range through which the conditioned air is cooled. Hence, in rare instances when the *condition line* of the process may not intersect the saturation line, any other convenient reference temperature on the *condition line*

may be used instead, provided that the coil efficiency is specified accordingly on the proper basis.

MINIMUM ENTERING AIR TEMPERATURE

Due consideration must be given to the temperature of the air entering the conditioned space in order to prevent objectionable drafts. With ceiling type diffusers or wall grilles with a high aspect ratio (See Chapter 40), many engineers consider 20 deg as the maximum difference for good design under average conditions. This difference can only be exceeded with extremely high ceiling outlets or wall grilles. Thus, if 80 F dry-bulb is to be maintained in a space with average ceiling height, the minimum delivered air temperature would be limited to about 60 F dry-bulb temperature. If the latent heat load is relatively high, it is often necessary to circulate more air with a higher delivered dry-bulb temperature in order to produce a thermodynamic balance. If the temperature difference is known, the required air quantity can be calculated from the formula,

$$Q_{ra} = \frac{q_s}{1.08\,(t_i - t_d)} \tag{21}$$

or the temperature difference t_d can be determined as follows,

$$t_d = t_i - \frac{q_s}{1.08 \times Q_{ra}} \tag{22}$$

EXAMPLE—COOLING LOAD CALCULATION

An effective means of summarizing the calculation procedure will be the use of an illustrative example. While condensed calculation forms are commonly employed for work of this nature, an outline will be used here in order to facilitate explanatory comments.

Example 11: A one-story office building Fig. 7 is located in an eastern state near 40 deg latitude. The adjoining buildings on the north and west are not conditioned and the air temperature within them is known to be substantially equal to the outdoor air temperature at any time of the day.

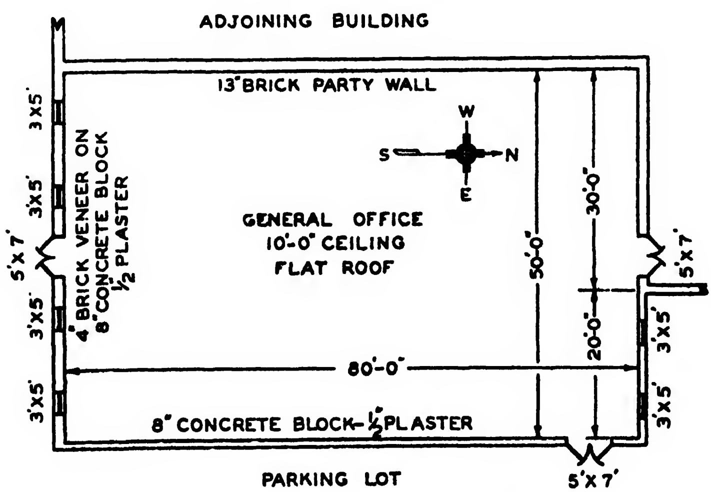


FIG. 7. PLAN OF ONE-STORY OFFICE BUILDING

South wall construction: 8 in. concrete block, 4 in. brick veneer, ¾ in. plaster on walls. (Table 8, Chapter 6, No. 92B, $U = 0.41$.)

East wall and outside north wall construction: 8 in. concrete block, painted white, ¾ in. plaster on walls. (Table 7, Chapter 6, No. 82B, $U = 0.52$.)

West wall and adjoining north party wall construction: 13 in. solid brick, no plaster:

$$\frac{1}{U} = \frac{1}{1.65} + \frac{13}{5} + \frac{1}{1.65} \quad \text{or,} \quad U = 0.263. \quad \text{Use } U = 0.26.$$

Roof construction: 2½ in. flat roof deck of 2 in. gypsum fiber concrete on gypsum board surfaced with built-up roofing. (Table 16, $U = 0.34$ *for summer.*)

Floor construction: 4 in. concrete on ground.

Window: 3 ft x 5 ft, non-opening type, with medium colored Venetian blinds for windows on south wall. Approximately 4 in. reveal on all windows.

Front doors: Two 2 ft-6 in. x 7 ft (glass panels).

Side doors: Two 2 ft-6 in. x 7 ft (½ glass panels).

Rear doors: Two 2 ft-6 in. x 7 ft (glass panels).

Outside design conditions: Maximum dry bulb 95 F, wet bulb 78 F; $W_o = 0.0169$ lbs vapor per lb dry air; $h_o = 41.38$ Btu per lb dry air.

Indoor design conditions: Dry bulb 80 F, wet bulb 65 F; $W_i = 0.0098$ lb vapor per lb dry air; $h_i = 29.95$ Btu per lb dry air.

Occupancy: 85 office workers.

Lights: 12,000 watts, fluorescent; 4000 watts tungsten.

Fan motor: 7½ hp.

Conditioning equipment to be located in adjoining structure to north.

Find: total, sensible, and latent maximum cooling loads and required air quantity through conditioning equipment.

Solution: From Table 4, the recommended ventilation rate is 15 cfm per person. Total necessary = $85 \times 15 = 1275$ cfm or 76,500 cu ft per hr.

As the room volume is 40,000 cu ft the air changes per hour will be 76,500/40,000 = 1.91 which is more than one air change.

Estimated Time of Maximum Cooling Load:

For this job, judgment indicates that the roof will make the greatest single contribution to the cooling load. Hence, the time of maximum cooling load probably will be the time of maximum heat gain through the roof. From Table 14 the maximum temperature differential for a 2 in. gypsum roof of medium weight construction is 54 deg at 4:00 P.M. and 53 deg at 3:00 P.M. Examination of Table 18 (40 deg N Latitude) shows that solar heat gain through glass on the south wall is 19 Btu per (hr) (sq ft) at 4:00 P.M. and 42 Btu at 3:00 P.M. This indicates that the maximum cooling load occurs at approximately 3:00 P.M. Therefore make load calculations at 3:00 P.M. suntime. (This may be slightly different than 3:00 P.M. local time.) In some cases, there would be no clear-cut evidence of this nature and consequently it would be necessary to estimate the load for several successive times and then to select the maximum.

Heat Gain Through Outer Wall and Roof Areas:

From Table 15 the temperature differential for the south wall (8 in. concrete block with 4 in. brick veneer) may be about the same as a 12 in. brick which is 6 deg at 3:00 P.M. for a dark colored wall. From the same table, the temperature differential for the east wall (8 in. concrete block with plaster) will be 11 deg at 3:00 P.M. for a *light colored* wall (interpolating between 2:00 and 4:00 P.M.). Likewise, the temperature differential for the north exposed wall (8 in. concrete block plus plaster) will be 3 deg at 3:00 P.M. (by interpolation) for a light wall.

The party wall of 13 in. brick on the West side and part of the North side may be treated as if it were an outside wall in the shade which has a temperature differential (from Table 15) of 2 deg.

For the door in North Wall estimate $U = 0.59$ from Chapter 6, Table 9, No. 5A. The outdoor temperature at 3:00 P.M. is 95 F. Neglect time lag and any decrement factor. The temperature differential is $(t_o - t_i) = 95 - 80 = 15$ deg. The tabulation of the preceding values at 3 P.M. is given in the following table:

Section	Net Area Sq Ft	Temperature Differential F Deg	Heat Transmission Coefficient (U)	Heat Flow Rate per Hour Btu
Roof	4000	53	0.34	72,000
South Wall	405[a]	6	0.41	995
East Wall	765[a]	11	0.52	4,380
North Exposed Wall	170[a]	3	0.52	265
West & North Party Wall	1065[a]	2	0.26	550
Door in North Wall	35	15	0.59	310
Total				78,500

[a] Calculated from gross wall area less windows and doors.

Heat Gain Through Glass Areas:

In computing the load for 3:00 P.M., only the south windows and doors will be exposed to direct sunlight. Table 18 and Equation 10 will give the total heat gain from the glass areas. The window reveals will shade the south windows slightly; the fraction of window area receiving direct radiation is obtained from Equation 11 by substituting values as follows:

$r_1 = s/l = 4/60$; $r_2 = 4/36$; $\beta = 45.5$ deg, $\tan \beta = 1.02$.

$\gamma = 16$ deg, $\tan \gamma = 0.287$.

$$G_t = 1 - \frac{4}{60}(1.02) - \frac{4}{36}(0.287) + \left(\frac{4}{60}\right)\left(\frac{4}{36}\right)(1.02)(0.287) = 0.902.$$

The south doors will be considered entirely sunlit. The outdoor air temperature is 95 F at 3:00 P.M. From Table 23 the inside Venetian blind factor is taken as 0.74. Referring to Table 18, the instantaneous heat gain due to solar and sky radiation for 40 deg N. latitude for south exposure at 3:00 P.M. is read as 42 Btu per sq ft. These figures are tabulated below. The radiation gain for south windows is $60 \times 0.902 \times 0.74 \times 42 = 1680$ Btu per hr. The normal heat transmission $60 \times 1.04\ (95-80) = 940$. The sum of these heat gains 2620 Btu per hr is totaled in last column. The remaining doors and windows are calculated in a similar manner.

Location	Area Sq Ft	Fraction Sunlit	Inside Venetian Blind Shading Factor	Solar Heat Gain Btu/sq ft	Radiation Heat Gain[a] Btu/hr	Normal Heat Trans. Btu/hr	Total Heat Gain Btu/hr
South Windows	60	0.902	0.74	42	1680	940	2620
South Doors	35	1.00	—	42	1470	550	2020
East[b] (Glass) doors	18	—	—	15	270	—	820
	35	—	—	—	—	550	
North Windows	30	—	—	15	450	470	920
Total							6380

[a] See Equation 10 and the note under caption of Table 18.

[b] Doors are ½ glass. Calculate sky radiation for glass portion and normal transmission for entire door assuming $U = 1.04$ for wood portion as well as glass.

In some jobs it would be desirable to increase (or decrease) the instantaneous radiation heat gain by a load-lag factor. The reason for not doing so in this case is that the solar gain is of a low magnitude and reference to the table indicates that 0.8 of the previous hour would not affect the results materially.

Heat Gain from Ventilation and Infiltration:

Since the necessary ventilation rate 1275 cfm is greater than one air change per hour, it will be satisfactory for determining the ventilation component of the heat gain.

Window infiltration can be taken as negligible since the windows do not open.

Door infiltration requires some judgment. Assume that for each person passing through the double doors, the infiltration will be 100 cu ft of outdoor air, see Chapter

8, Table 3. Assume that the outside doors will be used at the rate of 10 persons per hour and the inside doors at the rate of 30 persons per hour. Total infiltration will then be $40 \times 100 = 4000$ cfh or 67 cfm.

The design rate of entry of outside air is then:

$$Q = 1275 + 67 = 1342 \text{ cfm.}$$

The sensible, latent and total loads are determined from Equations 13, 14 and 15 respectively at 3:00 P.M. (Table 13) $t_o = 95$, $t_i = 80$, $W_o = 0.0169$, $W_i = 0.0098$. All the air entering the room as infiltration becomes a part of the space load.

Infiltration:

$q_s = 67 \times 1.08\ (95-80) = 1085$ Btuh sensible.

$q_e = 67 \times 4840\ (0.0169-0.0098) = 2300$ Btuh latent.

$q_t = q_s + q_e = 1085 + 2300 = 3385$ Btuh total.

Ventilation Air Taken Through Cooling Unit Which Does Not Become a Part of the Space Load:

$q_s = 1275 \times 1.08\ (95-80) = 20{,}700$ Btuh, sensible.

$q_e = 1275 \times 4840\ (0.0169-0.0098) = 43{,}800$ Btuh, latent.

$q_t = q_s + q_e = 20{,}700 + 43{,}800 = 64{,}500$ Btuh, total.

Heat Gain from Sources within the Conditioned Space:

For the occupants, use the data of Table 24 for moderately active office work.

Sensible heat gain $= 85 \times 200 = 17{,}000$ Btu per hr.

Latent heat gain $= 85 \times 250 = 21{,}250$ Btu per hr.

Total $= 38{,}250$ Btu per hr.

For the gain from *lighting*, use Equation 16 with a use factor of unity, and a special allowance factor of 1.20 for the fluorescents and of unity for the tungsten globes.

$$q_{el} = (12{,}000 \times 1.20 + 4000) \times 3.41 = 62{,}700 \text{ Btu per hr.}$$

For the *fan motor*, use Equation 17 with a load factor of unity and omit term *Motor Efficiency* because the motor is not within the space.

$$q_{em} = 7.5 \times 2544 = 19{,}100 \text{ Btu per hr.}$$

Moisture Permeation, Miscellaneous Allowance, and the Load-Lag Estimate

Moisture permeation will be negligible, since this is a comfort job with a good building construction.

There would be some heat gain in the ductwork, but this would not be great because of the short run involved. Practical judgment for this job would suggest that no adjustment for load-lag need be made to the load as computed. (Refer to Fig. 6). While it is true that inside radiation forms an important part of the total heat gain, it is advisable to be conservative in recognizing the effect of the large, flat, hot roof on the *comfort sensations of the occupants*. Radiation from the relatively low ceiling, augmented by heat absorption from the lighting fixtures, would produce a sensation of warmth in excess of the nominal effective temperature (see Chapter 12) established by the wet bulb and dry bulb temperatures. Hence, it is not desirable to take advantage of every small decrease possible in the peak design load, especially since the peak occurs in mid-afternoon when everything would be rather well warmed.

Total Loads and Required Air Quantity through Conditioning Equipment:

The total loads are summarized in Table 26.

Compute the *specific enthalpy of the water* difference *room air and supply air*, from Equation 19.

$$q_w = \frac{(184{,}765 + 23{,}550)}{23{,}550} \times 1076 = 9520.$$

TABLE 26. SUMMARY OF TOTAL LOADS—EXAMPLE 11

LOAD COMPONENT	SENSIBLE Btu/hr	LATENT Btu/hr
All Walls, roof and doors	78,500	. .
Glass areas	6,380	
Infiltration 67 cfm	1,085	2,300
Occupants	17,000	21,250
Lighting	62,700	.
Motor, Fan	19,100	.
Space Load	184,765	23,550
Ventilation 1275 cfm	20,700	43,800
Totals	205,465	67,350
Grand Total Sensible and Latent		272,815

From the Goff diagram, determine that the apparatus dew-point is 54.3 F (refer to Chapters 3 and 43)

In computing the effective air quantity, assume a coil efficiency of 85 per cent. Then,

$$Q_{ra} = \frac{[illegible]}{1.08(} \; [illegible] \; 0.85$$

(Refer to Chapter 25 for coil selection and efficiency.)

From note under Equation 20 the dry-bulb range will be $(80-54.3) \times 0.85 = 21.8$ deg and the dry bulb temperature of air leaving the coil will be $80 - 21.8 = 58.2$ F. The dry-bulb temperature leaving the fan (including the heat supplied by the fan motor), or delivered into the room, will be (from Equation 22):

With good distribution and diffusion, this temperature should not produce objectionable drafts.

EXAMPLE 11: SUMMARY

Outdoor Conditions	95 DB	78 WB	0.0169 Humidity Ratio
Space Conditions..	80 DB	65 WB	0.0098 Humidity Ratio
DIFFERENCE ...	15		0.0071

	Btu/Hr
SENSIBLE LOAD	
Transmission	
Roof 4000 sq ft × 53° × 0.34 =	72,000
S. Wall 405 sq ft × 6° × 0.41 =	995
E. Wall 765 sq ft × 11° × 0.52 =	4,380
N. Wall Ex. 170 sq ft × 3° × 0.52 =	265
N. & W. Party Wall 1065 sq ft × 2° × 0.26	550
Floor None	
Door 35 sq ft × 15 × 0.59 =	310
All Glass 160 sq ft × 15 × 1.04 =	2,510
Solar Radiation	
S. Glass 60 sq ft × 0.90 × 0.74 × 42 =	1,680
S. Glass (Doors) 35 sq ft × 1.0 × 42 =	1,470
E. Glass (Doors) 18 sq ft × 15 =	270
N. Glass 30 sq ft × 15 =	450
Internal Load	
Infiltration 67 cfm × 1.08 × 15° =	1,085
Lights (12,000 × 1.20 + 4000) 3.41 =	62,700
People 85 × 200 =	17,000
Motor, Fan 7.5 hp × 2544 =	19,100
TOTAL SENSIBLE SPACE LOAD	184,765

Latent Load		
Infiltration 67 cfm × 4840 × 0.0071 =	2,300	
People 85 × 250 =	21,250	
Total Latent Space Load	——	23,550
Ventilation Air		
Sensible 1275 cfm × 1.08 × 15° =		20,700
Latent 1275 cfm × 4840 × 0.0071 =		43,800
Grand Total Load		272,815

LETTER SYMBOLS USED IN CHAPTER 15

β = solar altitude, degrees.

γ = difference between the azimuth angle of an outdoor wall and the azimuth angle of the horizontal projection of the sun's rays, degrees.

μ = Permeability of material to moisture transmission, grains per (sq ft) (hr) (inch Hg).

λ = amplitude decrement factor, a variable depending on thickness, material, and orientation of the wall or roof dimensionless.

θ = angle of incidence for sun's rays striking a surface, degrees.

τ = fraction of incident radiant energy transmitted through a glass section, dimensionless.

A = area across which heat is being transferred, square feet.

a = fraction of incident radiant energy absorbed within a glass section, dimensionless.

b = fraction of incident radiant energy absorbed by a non-transparent surface, dimensionless.

e = ratio of direct solar radiation to sky radiation falling on a horizontal surface, dimensionless.

f_i = unit convective conductance for indoor surface = film coefficient of heat transfer of indoor air, Btu per (square foot) (hour) (Fahrenheit degree).

f_o = unit convective conductance for outdoor surface = film coefficient of heat transfer of outdoor air, Btu per (square foot) (hour) (Fahrenheit degree).

G_t = fraction of total window area receiving direct solar radiation when shaded by window reveal, dimensionless.

h_i = enthalpy of indoor air per pound of dry air, Btu per pound.

h_o = enthalpy of outdoor air per pound of dry air, Btu per pound.

I_d = direct solar radiation incident upon a surface at any angle of incidence, Btu per (square foot) (hour).

I_n = direct solar radiation incident on a surface at normal incidence, Btu per (square foot) (hour).

I_s = sky radiation incident upon a surface, Btu per (square foot) (hour).

I_t = $I_s + I_d$, Btu per (square foot) (hour).

K = cosine of angle of incidence for direct solar radiation striking a surface, dimensionless.

k = thermal conductivity of building material, Btu per (square foot) (hour) (Fahrenheit degree per foot).

L = thickness of building material, feet.

l = height of window, feet.

Q = rate of entry of outdoor air, cubic feet per minute.

Q_{ra} = required air quantity through conditioning equipment, cubic feet per minute.

q = instantaneous rate of heat transfer, Btu per hour.

q_e = instantaneous latent heat load, Btu per hour.

q_m = Latent head load due to moisture transmission through materials, Btu per (hr) (sq ft).

q_s = instantaneous sensible heat load, Btu per hour.

q_t = $q_e + q_s$, Btu per hour.

q_w = the heat-moisture ratio of the room load based upon the ratio of the total heat added divided by the required rate of water-vapor removed.

r = fraction of incident radiation reflected from glass surface, dimensionless.

t_e = sol-air temperature, Fahrenheit degrees.

t_e^* = sol-air temperature at a time earlier than the time for which heat gain is being found by an amount that is equal to the time lag of the wall or roof, Fahrenheit degrees.

t_i = indoor air temperature, Fahrenheit degrees.

t_L = temperature of outer surface of building, Fahrenheit degrees.

t_m = 24-hr cyclic average sol-air temperature, Fahrenheit degrees.

t_p = $t_m + (t_e^* - t_m)$, *net* equivalent outdoor temperature for combined periodic and mean heat flow, Fahrenheit degrees.

U = over-all coefficient of heat transfer of a structural section, Btu per (square foot) (hour) (Fahrenheit degree).

v_o = volume of outdoor air per pound of dry air, cubic feet.

w = width of window, feet.

W_i = humidity ratio of indoor air, pounds moisture per pound of dry air.

W_o = humidity ratio of outdoor air, pounds moisture per pound of dry air.

REFERENCES

[1] Application Engineering Standards for Air Conditioning for Comfort (*Air Conditioning and Refrigerating Machinery Association, Inc.*, 1947, p. 4–7).

[2] Proposed Standard Solar Radiation Curves for Engineering Use, by P. Moon (*Journal of the Franklin Institute*, November, 1940, Vol. 230, No. 5, p. 583–617).

[3] Summer Cooling for Comfort as Affected by Solar Radiation, by G. A. Hendrickson and J. H. Walker (*Heating and Ventilating*, Vol. 29, No. 11, November, 1932, p. 14–21).

[4] Dynamic Meteorology, by B. Haurwitz (McGraw-Hill Book Co., Inc., New York, 1941).

[5] Summer Weather Data and Sol-Air Temperature—Study of Data for New York City, by C. O. Mackey and E. B. Watson (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 75).

[6] A.S.H.V.E. RESEARCH REPORT No. 1268—Summer Weather Data and Sol-Air Temperature—Study of Data for Lincoln, Nebr., by C. O. Mackey (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 93).

[7] A.S.H.V.E. RESEARCH REPORT No. 923—Heat Transmission as Influenced by eat Capacity and Solar Radiation, by F. C. Houghten, . M. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 231). Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban (A.S.H.-V.E. TRANSACTIONS, Vol. 45, 1939, p. 369). Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 75). Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293). Periodic Heat Flow—Composite Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 52, 1946, No. 1299).

[8] A.S.H.V.E. RESEARCH REPORT No. 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J. Wahl (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 53).

[9] Study of Actual vs. Predicted Cooling Load on An Air Conditioning System, by James N. Livermore (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 287).

[10] A.S.H.V.E. RESEARCH REPORT No. 1195—Heat Gain Through Walls and Roofs

as Affected by Solar Radiation, by F. C. Houghten, E. C. Hach, S. I. Taimuty and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 91).

[11] Solar Heat Gain Through Walls and Roofs for Cooling Load Calculations, by J. P. Stewart (A.S.H.V.E. JOURNAL SECTION *Heating, Piping and Air Conditioning*, August 1948, p. 121).

[12] A.S.H.V.E. RESEARCH REPORT No. 1281—The Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 317).

[13] A.S.H.V.E. RESEARCH REPORT No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 83).

[14] A.S.H.V.E. RESEARCH REPORT No. 975—Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 101). A.S.H.V.E. RESEARCH REPORT No. 1180—Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten, and David Shore (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 251).

[15] The Mechanism of Heat Transfer Panel Cooling Heat Storage, by C. S. Leopold, (*Refrigerating Engineering*, July 1947, p. 33). Hydraulic Analogue for the Solution of Problems of Thermal Storage, Radiation, Convection and Conduction, by C. S. Leopold, (*Refrigerating Engineering*, July, 1947, p. 33).

[16] See Reference 1, p. 8.

[17] Cooler Footcandles for Air Conditioning, by W. G. Darley (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 367). Lighting and Air Conditioning Design Factors, Report of *I.E.S.*—A.S.H.V.E. Joint Committee on Lighting in Air Conditioning (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1941. p. 605). Lighting and Air Conditioning, by Howard M. Sharp (*Heating and Ventilating*, November, 1942, p. 35).

[18] Compiled by J. P. Stewart from various sources.

CHAPTER 16

FUELS AND COMBUSTION

Classification of Coals, Cokes, Fuel Oils, and Gas, Dustless Treatment of Coal, Fundamental Principles of Combustion, Heat of Combustion, Air Required for Combustion, Excess Air, Heat Balance, Firing Methods, Secondary Air, Draft Requirements, Draft Regulation, Furnace Volume, Combustion of Gas, Soot, Condensation and Corrosion

FUELS may be classified according to their physical state as solid, liquid, or gaseous. The principal fuels used for domestic heating are coal, oil, and gas. However, coke, wood, kerosene, sawdust, briquettes, and other substances are used for heating in special applications or in localities where an adequate supply is available. Experiments are in progress in the use of a colloidal suspension of coal particles in fuel oil, but this fuel has not attained wide-spread usage as yet. The choice of fuel is usually based on dependability, cleanliness, availability, economy, operating requirements, and control.

CLASSIFICATION OF COALS

Coal has a complex composition that makes classification into clear-cut types difficult. Chemically it consists of carbon, hydrogen, oxygen, nitrogen, sulfur, and a mineral residue called ash. A chemical analysis provides some indication of the quality of a coal, but does not define its burning characteristics sufficiently. The coal user is interested principally in the available heat per pound of coal, in the handling and storing properties, the amount of ash and dust produced and the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in a Bureau of Mines Bulletin[1].

There are two forms of coal analyses; namely, the *proximate analysis* and the *ultimate analysis*. In the *proximate analysis* the proportions of moisture, volatile matter, fixed carbon, sulfur, and ash are determined. This analysis is more easily made and is satisfactory for indicating most of the characteristics which are of interest to the user. For the proximate analysis the moisture is determined by observing the loss of weight of a sample of coal when dried at about 220 F. To determine the volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The remaining sample is then burned in an open crucible, and the accompanying loss of weight represents the fixed carbon. The unburned residue is ash. Although determined separately, the sulfur content is frequently reported with the proximate analysis because the usefulness of a coal for certain purposes depends on its sulfur content.

In the *ultimate analysis*, which is difficult to make, the percentages of carbon, hydrogen, oxygen, nitrogen, sulfur, and ash in the coal sample are determined. It is used for detailed studies of fuels and in computing a heat balance when required in testing of heating devices. Typical ultimate analyses of the various kinds of coal are shown in Table 1[2].

Other important qualities of coals are the screen sizes, ash fusion temperature, friability, caking tendency, and the qualities of the volatile

matter. In considering these factors the following points are of interest. The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases and is therefore low in the sub-bituminous coals and in lignite. The percentage of ash and its fusion temperature do not indicate the composition or distribution of its constituents.

A classification of coals is given in Table 2, and a brief description of the kinds of fuel is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

TABLE 1. TYPICAL ULTIMATE ANALYSES FOR COALS

Rank	Btu per Lb		Constituents, Per Cent						
	Moist, Mineral-matter-free[a]	Moist, as Received	Oxygen	Hydrogen	Carbon	Nitrogen	Sulfur	Ash	O_2+H_2+C
Anthracite	14,600	12,910	5.0	2.9	80.0	0.9	0.7	10.5	87.9
Semi-Anthracite	15,200	13,770	5.0	3.9	80.4	1.1	1.1	8.5	89.3
Low-Volatile Bituminous	15,350	14,340	5.0	4.7	81.7	1.4	1.2	6.0	91.4
Medium-Volatile Bituminous	15,200	13,840	5.0	5.0	79.0	1.4	1.5	8.1	89.0
High-Volatile Bituminous *A*	14,500	13,090	9.2	5.3	73.2	1.5	2.0	8.8	87.7
High-Volatile Bituminous *B*	13,500	12,130	13.8	5.5	68.0	1.4	2.1	9.2	87.3
High-Volatile Bituminous *C*	12,000	10,750	21.0	5.8	60.6	1.1	2.1	9.4	87.4
Sub Bituminous *A*	......	......	......	......	......	......	......	......	......
Sub Bituminous *B*	10,250	9,150	29.5	6.2	52.5	1.0	1.0	9.8	88.2
Sub Bituminous *C*	9,000	8,940	35.8	6.5	46.7	0.8	0.6	9.6	89.0
Lignite	7,500	6,900	44.0	6.9	40.1	0.7	1.0	7.3	91.0

[a] $\left(\frac{100}{100 - 1.1 \text{ Ash}}\right)$ (Btu as received).

Anthracite is a clean, dense, hard coal which creates little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires no attention to the fuel bed between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in a Bureau of Mines Report[3]. Standard anthracite sizing specifications are shown in Table 3.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bitumi-

nous coals, it can be burned with less production of smoke, and it is sometimes called *smokeless coal.*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics.

TABLE 2. CLASSIFICATION OF COALS BY RANK[a]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHY[S] PROPERTIE[S]
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerat
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II. Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomera or non-weather
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5. High volatile *C* bituminous Coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering non-agglomerat
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[e]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[e]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a] This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks. All of these coals either contain less than 48 per cent dry, mineral-matter-free fixed carbon or have more than 15,500 moist, mineral-matter-free Btu.

[b] If agglomerating, classify in low-volatile group of the bituminous class.

[c] Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d] It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e] Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f] There are three varieties of coal in the high-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials*, Philadelphia.

The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different

coals, but it is long. Much smoke and soot are possible, if improperly fired, especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like

TABLE 3. STANDARD ANTHRACITE SIZING SPECIFICATIONS[a]

SIZE	TEST MESH, IN.		ROUND MESH			MAXIMUM IMPURITIES, PER CENT	
			Oversize	Undersize			
	Through	Over	Max. Per Cent	Max. Per Cent	Min. Per Cent	Slate[b]	Bone
Broken	4⅜	3¼	....	15	7½	1½	2
Egg	3¼ to 3	2⁷⁄₁₆	5	15	7½	1½	2
Stove	2⁷⁄₁₆	1⅝	7½	12½	7½	2	3
Nut	1⅝	¹³⁄₁₆	7½	10	5	3	4
Pea	¹³⁄₁₆	⁹⁄₁₆	10	15	7½	4	5
Buckwheat	⁹⁄₁₆	⁵⁄₁₆	10	15	7½	12 Ash	
Rice	⁵⁄₁₆	³⁄₁₆	10	15	7½	13 Ash	
Barley	³⁄₁₆	³⁄₃₂	10	20	10	....	

[a] Approved and adopted by *Anthracite Committee*, State Street Building, Harrisburg, Pa.
[b] When slate content on Broken to Pea inclusive is less than above standards, bone content may be correspondingly increased, but slate content specified above shall not be exceeded in any event and the total maximum impurities shall not exceed those above specified.

charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with various petroleum products, a solution of calcium chloride or a mixture of calcium and magnesium chlorides.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

Dustless treatments which are of such a corrosive nature that they may damage coal handling or burning equipment should not be used.

CLASSIFICATION OF COKES

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distil-

lation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke.* The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

CLASSIFICATION OF FUEL OILS

Fuel oils are produced by distillation from crude petroleum after gasoline, naphtha, and other lighter products have been removed. Fuel oil is composed chemically of about 85 per cent carbon and 12½ per cent hydrogen with small amounts of oxygen, nitrogen, and sulfur. Oils are classified according to their specific gravity, but specific gravity alone is not a sufficient index of the properties that are important for heating purposes. Other characteristics that must be considered in the choice of a fuel oil are the flash point, pour point, water and sediment content, carbon residue, ash, sulfur content, distillation temperatures, and viscosity.

The flash point and distillation characteristics are important relative to easy ignition and complete gasification of the oil in a burner. A low pour point and low water content are of interest in connection with the storage of the fuel in outdoor tanks, while a low viscosity permits easy passage through a small orifice. The sediment, carbon residue, and ash content should be low to prevent clogging of strainers and accumulation of unburned material in the burner. The sulfur content may be of importance because of the corrosive effect of sulfur compounds in the burner and heating appliance or in special commercial processes.

The Commercial Standard Specifications for Fuel Oils (CS 12-40) of the *U. S. Department of Commerce* are given in Table 4. These specifications conform to *American Society for Testing Materials* Tentative Specifications for Fuel Oils D 396–38T.

The relationship between the *A.P.I.* gravity of fuel oils and their calorific value is given in Table 5. Fuel oil grades No. 1, No. 2 and No. 3 only are used in domestic heating equipment. Grades No. 5 and No. 6 are used in commercial and industrial burners and usually require preheating.

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually

Table 4. Detailed Requirements for Fuel Oils[a]

Grade[b]	Flash Point F Deg		Pour Point F Deg	Water and Sediment Per Cent	Carbon Residue Per Cent	Ash Per Cent	Distillation Temperatures F Deg				Viscosity Seconds			
							10 Per Cent Point	90 Per Cent Point		End Point	Saybolt Universal at 100 F		Saybolt Furol at 122 F	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	0[c]	Trace	0.05 on 10% Residuum[d]		410			560[e]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or Legal	190	10[c]	0.05	0.25 on 10% Residuum[f]		440	600						
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[c]	0.10	0.15 Straight			675	600[g]		45			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1.00		0.10						50	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[h]									300	45

[a]Recognizing the necessity for low sulfur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulfur, Max. Per cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No Limit
No. 6	No Limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b]It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c]Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

[d]For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e]The maximum end point may be increased to 590 F when used in burners other than sleeve type blue flame burners

[f]To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[g]The minimum distillation temperature of 600 F for 90 per cent may be waived if *A.P.I.* gravity is 26 or lower.

[h]Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

Table 5. Approximate Gravity and Calorific Value of Standard Grades of Fuel Oil

Commercial Standard No.	Approximate Gravity Range A.P.I.	Calorific Value Btu per Gallon
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	143,500
6	14-16	152,000

a combination of certain proportions of gases produced by two or more processes. Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 6.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 1000 to 1200 Btu per cu ft, the majority of natural gases averaging about 1000 Btu per cu ft. Table 6 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 6 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution

Table 6. Representative Properties of Gaseous Fuels, Based on Gas at 60 F and 30 in. Hg

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (F deg)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1085	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	970	870	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1 17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1130	1025	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	570	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	590	520	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carbureted water gas	540	495	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	300	280	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	135	125	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

TABLE 7. GENERAL DATA OF COMBUSTIBLE ELEMENTS AND COMPOUNDS [a]

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] F Deg	Calorific Value: Btu per Lb		Btu per Cu Ft[b]	Theoretical Oxygen and Air Requirements: Lb per Lb		Cu Ft per Cu Ft	
				Higher	Lower	Higher	O_2	Air	O_2	Air
Carbon (to CO)	—	$2C + O_2 = 2CO$	—	3950	—	—	1.332	5.763	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14093	—	—	2.664	11.527	—	—
Sulfur (to SO_2)	—	$S + O_2 = SO_2$	—	3983	—	—	0.998	4.285	—	—
Sulfur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	1.497	6.428	—	—
Carbon Monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	4347	—	321.8	0.571	2.471	0.5	2.382
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1260-1380	23879	21520	1013.2	3.990	17.265	2.0	9.528
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21500	20776	1499	3.073	13.297	2.5	11.911
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21644	20295	1613.8	3.422	14.807	3.0	14.293
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	990-1120	22320	20432	1792	3.725	16.119	3.5	16.675
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	60958[c]	51571[c]	325	7.937	34.344	0.5	2.382
Hydrogen Sulfide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	7100	6545	647	1.409	6.097	1.5	7.146
Propane	C_3H_8	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	950-1080	21661	19944	2590	3.629	15.703	5.0	23.821
n-Butane	C_4H_{10}	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	890-1020	21308	19680	3370	3.579	15.487	6.5	30.967
Commercial Propane	—	—	—	21650	—	2500	—	—	—	—
Commercial Butane	—	—	—	21400	—	3200	—	—	—	—

[a]Values in table taken chiefly from page 51 of Fuel Flue Gases published by *American Gas Association*.
[b]Gas measured at 60 F and 30 in. Hg.
[c]Value from *National Bureau of Standards*.

of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cu ft with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. However, in any community the variations in gas composition are held within suitable limits so that the performance of approved gas appliances will not be adversely affected.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen, partly upon the rate at which oxygen is supplied, and upon the temperature obtained due to surrounding conditions.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combusion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume, Per Cent	By Weight, Per Cent
Oxygen, O_2	20.9	23.15
Nitrogen, N_2	79.1	76.85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 7.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 7.

HEAT OF COMBUSTION

As previously stated, the process of combustion results in the evolution of heat. The heat generated by the complete combustion of a unit of fuel is constant for a given combination of combustible elements and compounds, and is known as the *heat of combustion*, *calorific value*, or *heating value* of the fuel. The heat of combustion of the several substances found in the more common fuels is given in Table 7.

The calorific value of a fuel may be determined either by direct measurement of the heat evolved during combustion in a calorimeter, or it may be computed from the ultimate analysis and the heat of combustion of the several chemical elements in the fuel. When the heating value of a fuel

is determined in a calorimeter the water vapor is condensed and the latent heat of vaporization is included in the heating value of the fuel. The heating value so determined is termed the *gross* or *higher* heating value and this is what is ordinarily meant when the heating value of a fuel is specified. In burning the fuel, however, the products of combustion are not cooled to the dew-point and the higher heating value cannot be utilized.

When combustion is complete, the carbon in the fuel unites with oxygen to form carbon dioxide, CO_2, the hydrogen unites with oxygen to form water vapor, H_2O, and the nitrogen, being inert, passes through the reaction without change. When combustion is incomplete, some of the carbon may unite with oxygen to form carbon monoxide, CO, and some of the hydrogen and hydrocarbon gases may not be burned at all. When carbon monoxide or other combustible gases are present in the flue gases, considerably less heat is produced per unit of fuel consumed, and a lower combustion efficiency is obtained. Incomplete combustion may result from any or all of the following three conditions:

1. Inadequate air supply.
2. Insufficient mixing of air and gases.
3. A temperature too low to produce ignition or maintain combustion.

AIR REQUIRED FOR COMBUSTION

The weight of air required for perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to Equations 1 and 2. The various elements are expressed in percentages by weight.

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56\left[\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right] \quad (1)$$

Gaseous Fuels:

$$\text{Pounds air required per pound fuel} = 2.46\,CO + 34.56\,H_2 + 17.28\,CH_4 + 13.29\,C_2H_2 + 14.81\,C_2H_4 + 16.13\,C_2H_6 + 6.10\,H_2S - 4.32\,O_2 \quad (2)$$

When the analysis is given on a volumetric basis the equation is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\,(CO + H_2) + 9.56\,CH_4 + 11.98\,C_2H_2 + 14.35\,C_2H_4 + 16.74\,C_2H_6 - 4.78\,O_2 \quad (3)$$

Equations 4 and 5 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Heating value (Btu per pound)}}{1000} \quad (4)$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Heating value (Btu per unit)}}{100} \quad (5)$$

Approximate values for the theoretical air required for different fuels are given in Table 8.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual amount of air supplied for combustion. The analysis of flue gases has been well

TABLE 8. APPROXIMATE THEORETICAL AIR REQUIREMENTS

SOLID FUEL	POUNDS AIR PER POUND FUEL
Anthracite	9.6
Semi-bituminous coal	11.2
Bituminous coal	10.3
Lignite	6.2
Coke	11.2
FUEL OIL	**POUNDS AIR PER GALLON FUEL**
Commercial Standard No. 1	102.6
Commercial Standard No. 2	104.5
Commercial Standard No. 3	106.5
Commercial Standard No. 5	112.0
Commercial Standard No. 6	114.2
GASEOUS FUELS	**CUBIC FEET AIR PER CUBIC FOOT GAS**
Natural gas	10.0
Mixed, natural and water gas	4.4
Carbureted water gas	4.4
Water gas, coke	2.1
Coke oven gas	5.2

described in various publications of the *U. S. Bureau of Mines* and in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 11.)

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by Equation 6.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\,CO_2 + 8\,O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)} \times C \qquad (6)$$

Values for CO_2, O_2, CO, and N_2 are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

EXCESS AIR

Since one measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied, a method of determining the latter factor is of value. Equation 7 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} \quad \frac{3.04\,N_2}{(CO_2 + CO)} \times C \qquad (7)$$

Values for CO_2, CO, and N are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

The difference between the air actually supplied for combustion and the theoretical air required is known as excess air.

$$\text{Per cent excess air} \quad \frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}} \times 100 \qquad (8)$$

Since the calculation is usually made from Orsat analysis, Equation 9 will be found to be a convenient statement of this relationship.

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \qquad (9)$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for complete combustion of several fuels are given in Table 9.

To produce heat efficiently with any of the common fuels the following requirements must be observed:

1. Adequate heat absorbing surface is necessary.
2. The heat transfer surfaces must be clean.
3. A minimum of excess air should be used.
4. The combustion air and the combustible gases produced by the fuel must be well mixed.
5. The quantity of combustible gases escaping to the stack must be kept small.

If insufficient heating surface is provided in a heating appliance, or if the heat transfer surfaces are covered with soot, ash or scale, the flue gas temperature will be excessive and the amount of sensible heat passing up the stack will be unnecessarily large. Too much excess air dilutes the flue gases excessively and increases the sensible flue gas loss, while a deficiency of air will cause some combustible gases to pass out of the appliance unburned. The highest combustion efficiency is not always obtained by supplying enough excess air to reduce the incomplete combustion loss to zero, but the incomplete combustion loss should be kept small. If the secondary air is not well mixed with the combustible gases, some incomplete combustion may still occur. Unnecessary secondary air also dilutes the flue gases and increases the sensible heat escaping up the chimney. Some excess air is always required in the practical operation of heating plants. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, depending upon the fuel used.

HEAT BALANCE

In analyzing the performance of a heating appliance, it is frequently desirable to make an accounting, insofar as possible, of the disposition

Table 9. Representative Maximum CO_2 Value

Fuel	Theoretical CO_2	CO_2 Usually Attained In Practice
Coke	21.00	12-14
Anthracite	20.20	12-14
Bituminous Coal	18.20	13
No. 2 Fuel Oil	15.00	10.5
No. 6 Fuel Oil	16.50	13.5
Natural Gas	12.00	9.7
Coke Oven Gas	11.00	8.5

of all the heat units in the fuel used. Such an accounting is sometimes called a *heat balance*. The several components of the heat balance may either be expressed in terms of Btu per pound of fuel used or as a percentage of the calorific value of the fuel. The components of the heat balance are listed in items 1 to 7.

1. Useful heat transferred to heating medium and usually evaluated by determining the rate of flow of the heating fluid through the heating device and the change in enthalpy of the fluid (heat added) between the inlet and outlet.

2. Heat loss in the dry chimney gases.

$$h_1 = w_g c_p \, (t_g - t_a) \tag{10}$$

3. Heat loss in water vapor formed by the combustion of hydrogen.

$$h_2 = \frac{9H_2}{100}\,(1091.8 + 0.455\, t_g - t_a) \tag{11}$$

4. Heat loss in water vapor in the air supplied for combustion.

$$h_3 = 0.455\, M\, w_a\, (t_g - t_a) \tag{12}$$

5. Heat loss from incomplete combustion.

$$h_4 = 10143\, C \left(\frac{CO}{CO_2 + CO}\right) \tag{13}$$

6. Heat loss from unburned carbon in the ash or refuse.

$$h_5 = 14093 \left(\frac{C_u}{100} - C\right) \tag{14}$$

7. Radiation and all other unaccounted for losses.

Since the radiation and convection losses from a heating appliance are not usually determined by direct measurement, they, together with any other losses not measured, are determined by subtracting the total of items 1 to 6 inclusive from the heat of combustion of the fuel. Frequently, when there is CO in the flue gases there also will be small amounts of unburned hydrogen and hydrocarbon gases in the products of combustion. The loss represented by these unburned gases may easily be as large as that resulting from the presence of carbon monoxide. In this event item 7 of the heat balance would also include this unmeasured loss.

Symbols used in Equations 10 to 14 inclusive are:

h_1 = heat loss in the dry chimney gases, Btu per pound of fuel.

h_2 = heat loss in water vapor from combustion of hydrogen, Btu per pound of fuel.

h_3 = heat loss in water vapor in combustion air, Btu per pound of fuel.

h_4 = heat loss from incomplete combustion of carbon, Btu per pound of fuel.

h_5 = heat loss from unburned carbon in the ash, Btu per pound of fuel.

w_g = weight of dry flue gas per pound of fuel (from Equation 6), pounds.

c_p = mean specific heat of flue gases at constant pressure (c_p ranges from 0.242 to 0.254 for flue gas temperatures from 300 F to 1000 F)[2], Btu per pound.

t_g = temperature of flue gases at exit of heating device, Fahrenheit degrees.

t_a = temperature of combustion air, Fahrenheit degrees.

H_2 = percentage of hydrogen in fuel by weight from ultimate analysis of fuel as fired.

1091.8 = enthalpy of saturated water vapor at a temperature of 70 F, Btu per pound.

M = humidity ratio of combustion air, pounds of water vapor per pound of dry air.

w_a = weight of combustion air per pound of fuel used, pounds, from Equations 1, 2, 8, and 9.

C = weight of carbon burned per pound of fuel corrected for carbon in ash, pounds.

$$C = \frac{WC_u - W_aC_a}{100W} \tag{15}$$

where

C_u = percentage of carbon in the fuel by weight from the ultimate analysis.

CO, CO_2 = percentages of CO, CO_2 in flue gases by volume.

W_a = weight of ash and refuse, pounds.

C_a = per cent of combustible in ash by weight (combustible in ash is usually considered to be carbon).

W = weight of fuel used, pounds.

The flue gas losses listed as items 2, 3, and 4 of the heat balance may be determined with considerable accuracy from the curves shown in Fig. 1[2] in many cases. The values of the losses plotted for fuel oil were computed from the ultimate analysis of a typical fuel oil used in domestic burners, while those plotted for the several ranks of coal were computed from the typical ultimate analyses shown in Table 1. The curves for medium volatile bituminous coal may be used for high volatile bituminous coal with negligible error. No curves are shown for gaseous fuel because various natural gases and manufactured gases vary considerably in their composition.

FIRING METHODS FOR ANTHRACITE

An anthracite fire should never be poked or disturbed, as this serves to bring ash to the surface of the fuel bed where it may melt into clinker.

Egg size is suitable for large fire-pots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. For best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. When fired carefully pea coal can be burned on standard grates. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. A satisfactory method of firing pea coal consists of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open and regulating solely by means of the cold air check and the air inlet damper.

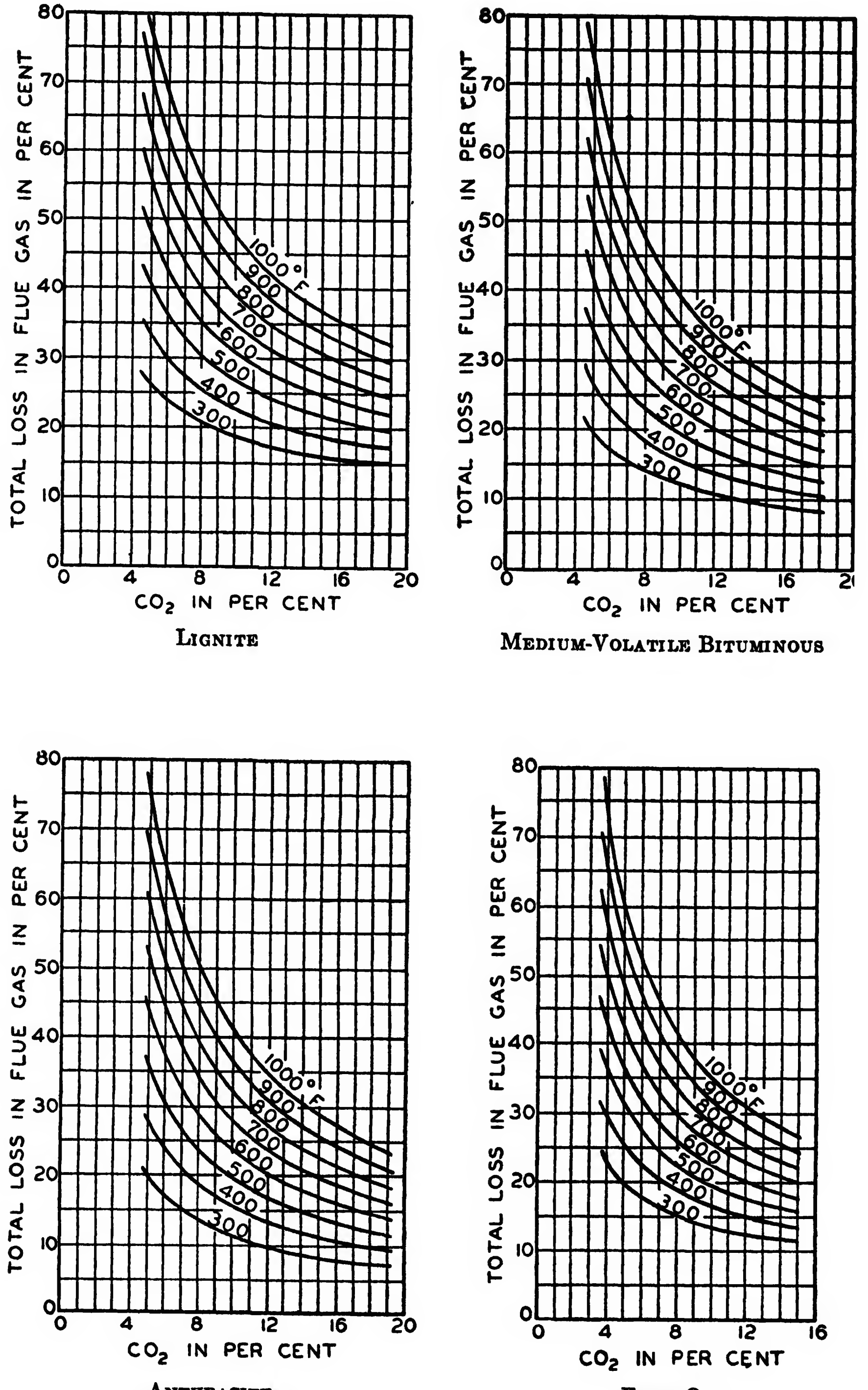


FIG. 1. FLUE GAS LOSSES WITH VARIOUS FUELS[3]

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with the pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a straight poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent delayed ignition within the fire pot, which in some cases, depending upon the thickness of the bed of fresh coal, is severe enough to blow open the doors and dampers of the furnace. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate. Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For greater convenience, domestic stokers are used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

FIRING METHODS FOR BITUMINOUS COAL

A commonly recommended procedure for firing domestic heating units, called the side-bank method, requires the movement of live coals to one side or the back of the grate, and placing the fresh fuel charge on the opposite side. The results are a more uniform release of volatile gases, and the subjection of these gases to the high temperature of the red coals. If the fresh charge is covered with a layer of fine coal, still better results may be obtained because of slower release of volatile matter.

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

The importance of firing bituminous coal in small quantities at short intervals is discussed in a U. S. Bureau of Mines technical paper[1]. Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the fire-box. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent

attendance is impracticable. The adherence to these methods insofar as practicable, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtained with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will usually require frequent attention to the fuel bed.

Preventing Smoke

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals.

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher *A.P.I.* gravity oil always produces less smoke than a high volatile coal or low *A.P.I.* gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel-burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is often necessary.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large

quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

FIRING METHODS FOR SEMI-BITUMINOUS COAL

The *Pocahontas Operators' Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone, the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

FIRING METHODS FOR COKE

Coke ignites less readily than bituminous coal and more readily than anthracite and burns rapidly with little draft. In order to control the air

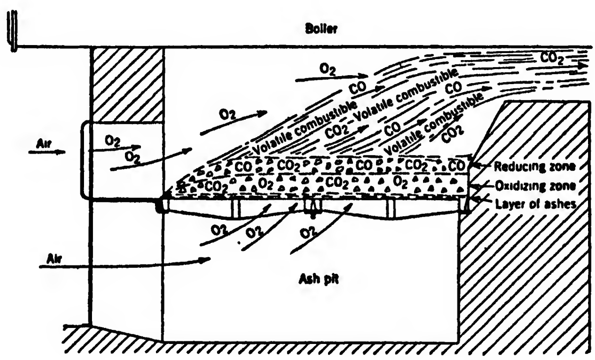


FIG. 2. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small fire-pots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a $1\frac{1}{2}$ in. screen. For large fire-pots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of the coke is CO_2 and under certain conditions some CO may arise from

the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 per cent of the heat in the fuel in the combustion space over the fuel bed.

The air that passes through the fuel bed is called *primary air* and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called *secondary air*.

This process of combustion is illustrated in Fig. 2[4]. The free oxygen of the air passes through the grate and the ash above it and burns the carbon in the lower 3 or 4 in. of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel bed are mainly carbon monoxide, carbon dioxide, nitrogen, and a small amount of free oxygen. Free oxygen is admitted through the firing door in an attempt to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size and type of fuel, depth of fuel bed, and size of fire-pot.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings, but, nevertheless, the smallest size of fuel will require the largest secondary air openings. For certain sizes of fuel no secondary air openings are required, and for large sizes, too much excess air may pass through the fuel bed.

In general, the efficiency of domestic hand-fired furnaces and boilers burning either anthracite or bituminous coal can be increased for an hour or two after firing, if some secondary air is admitted through the slots of the fire door. However, unless the slots are closed when secondary air is no longer beneficial, the decrease in efficiency during the remainder of the firing cycle because of excess air may more than offset the gain resulting from the secondary air at the beginning of the firing period. Unless the secondary air can be readjusted between firings, it is probable that a greater average efficiency will be obtained for domestic hand-fired devices by leaving the secondary air slots closed at all times. There is usually an appreciable amount of air leakage around the firing door and secondary air slots of domestic furnaces and boilers.

When attention is given between firings the efficiency of combustion can be appreciably raised by admitting secondary air over a bituminous coal fire to burn the gases and reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial.

Secondary air that enters the combustion chamber too far removed from the zone of combustion will also be harmful, for the oxygen in the secondary air will not react with any unburned gases unless the mixture is subjected to high temperatures.

The air requirements of oil and gas burners are discussed in Chapter 17, Automatic Fuel Burning Equipment.

DRAFT REQUIREMENTS

The draft required to effect a given rate of burning the fuel is dependent on the following factors:

1. Kind and size of fuel.
2. Grate area.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient

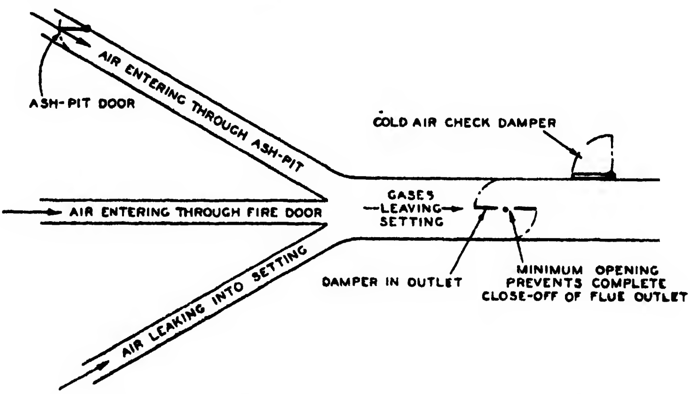


Fig. 3. Correct and Incorrect Methods of Draft Regulation in a Hand-Fired Furnace

draft also restricts the control that can be accomplished by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed, or an extremely thin fuel bed, it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

For amount of draft required see Chapter 19, Chimneys and Draft Calculations.

DRAFT REGULATION

Because of the varying heating load demands present in most installations it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Correct and incorrect methods of draft regu-

lation are shown in Fig. 3. The air enters through the ashpit draft door, firing door, and by leaks in the setting, whereas the gases leave only through the outlet. By throttling the gases with the damper in the outlet all the air entering by each of the three intakes is reduced in the same proportion, thus maintaining about the same per cent of excess air. If inlet air is controlled by the ashpit draft door, the air admitted through the ashpit is reduced, while it is increased through the other two intake openings, resulting in an increase of excess air. A considerable increase in the efficiency of hand-fired furnaces and boilers can be realized by regulating the air supply with the damper in the outlet instead of the ashpit damper. Use of the ashpit damper is required, of course, for low rates of combustion. The cold air check damper is to be used only when chimney draft is excessive. It is normally closed unless closing of the outlet damper and ashpit damper is unable to control the rate of combustion.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 17, Automatic Fuel Burning Equipment.

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires comparatively little combustion space.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports. Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. This secondary air should be brought into intimate contact with the gas.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat. The radiants also serve as baffles in directing the flow of the products of combustion.

The quantity of air given in Table 6 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. In order to insure freedom from carbon monoxide under conditions which may obtain in installations, it is customary to design gas burning appliances for a supply of 30 to 35 per cent of excess air. In individual installations in which flue gas analyses are made, the excess air is sometimes reduced to approximately 20 per cent. The division of the air into primary and secondary, is a matter of burner design, the pressure of gas available, and the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become farther apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface and reduces the heat transmission to the water or air. The *Bureau of Mines Report of Investigations* No. 3272[5] shows that the loss of seasonal efficiency is not so great as has been believed, and usually is not over 6 per cent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54[6] points out that, although the decrease in efficiency of an oil fired boiler due to soot deposits is relatively small, the attendant increase in stack temperature may be considerable.

Table 10. Average Flue Gas Dew-Point for Various Fuels[8]

Type of Fuel	Average Dew-Point Temperature, F
Anthracite	68
Semi-Bituminous Coal	84
Bituminous Coal	93
Oil	111
Natural Gas	127
Manufactured Gas	137

The soot accumulation clogs the flues, reduces the draft, and may prevent proper combustion. Soot can probably be most effectively removed by a jet of compressed air or by means of a brush. However, it has been found that copper chloride, lead chloride, tin chloride, zinc chloride, common salt and some other salts are partially effective in removing soot from furnaces and boilers when properly used.[7]

CONDENSATION AND CORROSION

Sulfur dioxide or sulfur trioxide formed by the combustion of sulfur in fuels is the principal corroding element in flue gases, and becomes active whenever moisture is present for the formation of sulfurous or sulfuric acid. It is necessary, therefore, to maintain a flue gas temperature in excess of the dew-point temperature of the flue gases in all parts of appliances unless they are made of materials that will resist these corrosive influences. It is usually desirable to maintain a flue gas temperature above the dew-point temperature throughout the heating appliance and the chimney or smokestack because of these same corrosive effects. The average dew-point temperatures of the flue gases from the several

fuels, when burned with the amount of excess air usually supplied to insure complete combustion, are shown in Table 10.

LETTER SYMBOLS USED

h_1 = heat loss in the dry chimney gases, Btu per pound of fuel.

h_2 = heat loss in water vapor from combustion of hydrogen, Btu per pound of fuel.

h_3 = heat loss in water vapor in combustion air, Btu per pound of fuel.

h_4 = heat loss from incomplete combustion of carbon, Btu per pound of fuel.

h_5 = heat loss from unburned carbon in the ash, Btu per pound of fuel.

w_g = weight of dry flue gas per pound of fuel (from Equation 6), pounds.

c_p = mean specific heat of flue gases at constant pressure.

t_g = temperature of flue gases at exit of heating device, Fahrenheit degrees.

t_a = temperature of combustion air, Fahrenheit degrees.

H_2 = percentage of hydrogen in the fuel by weight from ultimate analysis of fuel as fired.

M = humidity ratio of combustion air, pounds of water vapor per pound of dry air.

w_a = weight of combustion air per pound of fuel used, pounds.

C = weight of carbon burned per pound of fuel corrected for carbon in ash, pounds.

C_u = percentage of carbon in the fuel by weight from the ultimate analysis.

CO, CO_2 = percentages of CO, CO_2 in the flue gases by volume.

W_a = weight of ash and refuse, pounds.

C_a = per cent of combustibles in ash and refuse by weight.

W = weight of fuel used, pounds.

REFERENCES

[1] Five Hundred Tests of Various Coals in Househeating Boilers (*U. S. Bureau of Mines Bulletin* No. 276).

[2] Combustion Efficiencies as Related to Performance of Domestic Heating Plants, by A. P. Kratz, S. Konzo and D. W. Thompson (*Illinois Engineering Experiment Station Circular* No. 44).

[3] Quality of Anthracite as Prepared at Breakers, 1935 (*U. S. Bureau of Mines Report of Investigation*, R. I. 3283).

[4] Hand Firing Soft Coal Under Power Plant Boilers (*U. S. Bureau of Mines Technical Paper* No. 80).

[5] Effect of Soot on Heat Transmission in Boilers (*U. S. Bureau of Mines Report of Investigation* No. 3272).

[6] Effect of Soot on the Rating of an Oil-Fired Heating Boiler (*National Bureau of Standards Report* BMS 54).

[7] Removal of Soot from Furnaces and Flues by the Use of Salts and Compounds, by P. Nicholls and C. W. Staples (*U. S. Bureau of Mines Bulletin* No. 360).

[8] Condensation of Moisture in Flues, by William R. Morgan (*Illinois Engineering Experiment Station Circular* No. 22).

BIBLIOGRAPHY

Fuels and Their Combustion, by Haslam and Russell (McGraw-Hill Co., 1926).

Principles of Combustion in the Steam Boiler Furnace, by Arthur D. Pratt (Babcock and Wilcox Co.).

Smoke-Producing Tendencies in Coals of Various Ranks, by H. J. Rose and F. P. Lasseter (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 329).

Fundamentals of Combustion in Small Stokers, by C. A. Barnes (*Bituminous Coal Research, Inc., Technical Report* No. IV).

Hand-Firing of Bituminous Coal in the Home, by A. P. Kratz, J. R. Fellows, and J. C. Miles (*Illinois Engineering Experiment Station Circular* No. 46).

Classification and Selection of Illinois Coals, by G. H. Cady (*Illinois State Geological Survey Bulletin* No. 62).

Bituminous Coal Research, Inc.:

Technical Report VI, The Treatment of Coal with Oil and Other Petroleum Products, by J. M. Pilcher and R. A. Sherman.

Information Bulletin No. 4, Dustless Treatment of Coals with Materials Other than Oil, by R. A. Sherman and G. W. Land.

Information Bulletin No. 6, Questions and Answers on the Use of Fuel Oil for Dustless Treatment.

Technical Report VII—Application of Overfire Air Jets, by R. B. Engdahl.

Bureau of Mines Publications:

Bulletin No. 97, Sampling and Analyzing Flue Gases, by Henry Kreisinger and F. K. Ovitz.

Report of Investigations (R. I. 2980), Coke as a Domestic Heating Fuel, by P. Nicholls and B. A. Landry.

Technical Paper No. 303, Value of Coke, Anthracite, and Bituminous Coal for Generating Steam in a Low-pressure Cast-Iron Boiler, by John Blizard, James Neil, and F. C. Houghten.

Bulletin 378, Effect of Preheat, and Distribution of Ash in Fuel Beds, by P. Nicholls.

Handbook, Questions and Answers for the Home Fireman (Revised), by J. F. Barkley.

Anthracite Industries Laboratories Publications:

Report 2015, Comparison of Sizes, Egg, Stove and Chestnut Anthracite.

Report 2018, Domestic Survey.

Report 2062, Utilization of Anthracite for Domestic Heating.

Report 2204, The Crater Method of Firing.

Report 2403, Anthracite Industries Manual.

Oil and Gas Publications:

Oil Fuels and Burners, by James A. Moyer (McGraw-Hill Co., 1937).

Handbook of Oil Burning, by Harry F. Tapp (*American Oil Burner Association*).

Industrial Gas Series, Combustion (*American Gas Association*).

Comfort Heating (*American Gas Association*).

CHAPTER 17

AUTOMATIC FUEL BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Rating; Classification of Oil Burners, Combustion Process, Combustion Chamber Design; Classification of Gas-Fired Heating Equipment, Combustion Process, Ratings; Sizing of Gas Piping, Fuel Burning Rates

AUTOMATIC mechanical equipment for the combustion of solid, liquid, and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

CLASSIFICATION OF STOKERS ACCORDING TO CAPACITY

Stokers may be classified according to their coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association*.

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

Class 1 Stokers

These stokers are used primarily for home heating and are designed for quiet, automatic operation. Simple, trouble-free construction and attractive appearance are desirable characteristics of these small units.

A common stoker in this class (Fig. 1) consists essentially of a coal hopper, a screw for conveying the coal from the hopper to the retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor for supplying power for coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort which may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals, and coke. The *U. S. Department of Commerce* has issued commercial standards for household anthracite stokers[1].

Units are available in either the hopper type, as shown in Fig. 1, or in the bin-feed type as shown in Figs. 2 and 3. Some stokers, particularly those designed for use with anthracite, automatically remove ash from the ash pit and deposit it in an ash receptacle as shown in Fig. 3. Most

of the bituminous models, however, require removal of the ash from the fuel bed after it is fused into a clinker.

Stokers in this class feed coal to the furnace intermittently in accordance with temperature or pressure demands. A special control is used to insure sufficient stoker operation to maintain a fire during periods when no heat is required. Where year-round domestic hot water is supplied by a boiler and indirect water heater connected to a storage tank, the stoker will usually be called on to operate often enough to maintain the fire.

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are available having design features closely coordinating the heat absorber and the stoker. Although efficient and satisfactory performance can be obtained from the application of stokers to existing boilers and

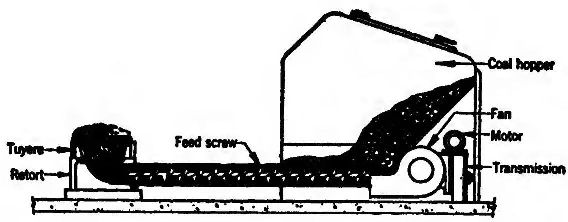


Fig. 1. Underfeed Stoker, Hopper Type, Class 1

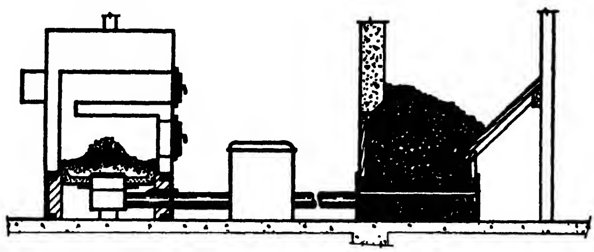


Fig. 2. Underfeed Stoker, Bin Feed Type, Class 1

furnaces, some of the combination stoker-fired units (Fig. 4) are more compact and attractive in appearance.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial plants. They are of the underfeed type and are available in both the hopper type, as illustrated in Fig. 5, and the bin feed type, shown in Fig. 6. These units also are built in plunger feed type with an electric motor or a steam or hydraulic cylinder coal feed drive.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies according to the fuel and load conditions Stationary type grates are used on bituminous models and the clinkers formed from the ash accumulate on the grates surrounding the retort.

Anthracite stokers in this class are equipped with moving grates which

discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and on some installations is of sufficient capacity to hold the ash for several weeks' operation.

Class 4 Stokers

Stokers in this group vary widely in details of design and several methods of feeding coal are employed. The underfeed stoker is widely used, although a number of the overfeed types are used in the larger sizes. Bin-feed, as well as hopper models, are available in both underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (1) underfeed side cleaning, (2) underfeed rear cleaning, (3) overfeed flat grate, and (4) overfeed inclined grate.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. They are not so varied in design as those in the smaller classes, although the principle of operation is similar. A stoker of this type is illustrated in Fig. 7.

The rear cleaning underfeed stoker is usually of the multiple retort design and is used in some of the largest industrial plants and central power stations. Zoned air control has been applied to these stokers, both longitudinally and transversely of the grate surface.

The overfeed flat grate stoker is represented by the various chain—or traveling-grate stokers. A typical traveling-grate stoker is illustrated in Fig. 8.

Another distinct type of overfeed flat-grate stoker is the spreader (Figs. 9 and 10) type in which coal is distributed either by rotating paddles or by air over the entire grate surface. This type of stoker is adapted to a wide range of fuels and has a wide application on small sized fuels, and on fuels such as lignites, high-ash coals, and coke breeze.

The overfeed inclined-grate stoker operates on the same general combustion principle as the flat-grate stoker, the main difference being that rocking grates, set on an incline, are provided in the former to advance the fuel during combustion.

Combustion Process

In anthracite stokers of the Class 1 underfeed type, burning takes place entirely within the stoker retort. The refuse of combustion spills over the edge of the retort into an ash pit or receptacle from which it may be removed either manually or automatically.

Larger underfeed anthracite stokers operate on the same principle, except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually the No. 1 buckwheat or No. 2 buckwheat size.

Because the majority of the smaller bituminous coal stokers operate on the underfeed principle, a general description of their operation is given. When the coal is fed into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

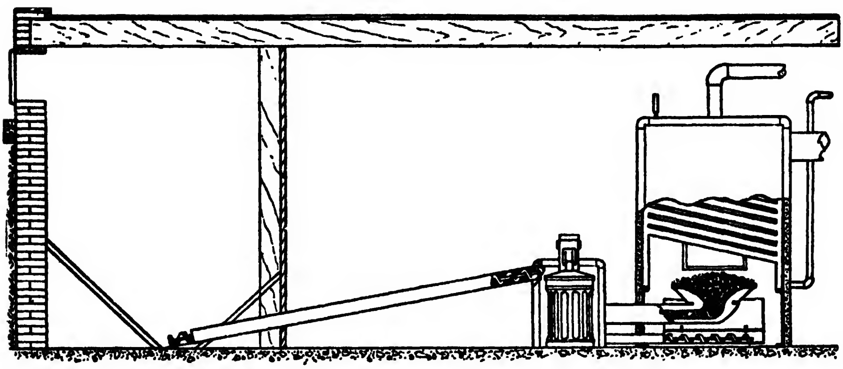


FIG. 3. UNDERFEED ANTHRACITE STOKER WITH AUTOMATIC ASH REMVOVAL, BIN TYPE

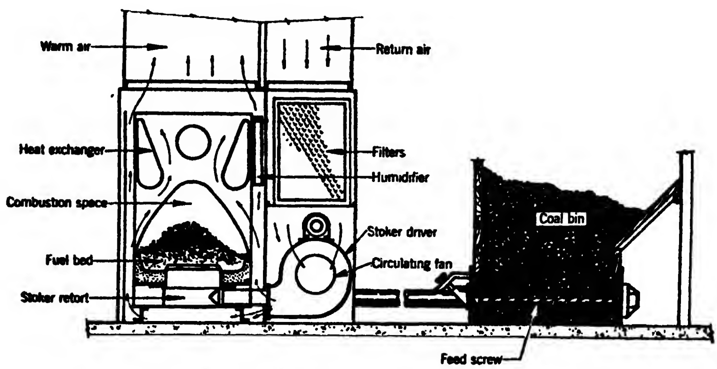


FIG. 4. STOKER-FIRED WINTER AIR CONDITIONING UNIT

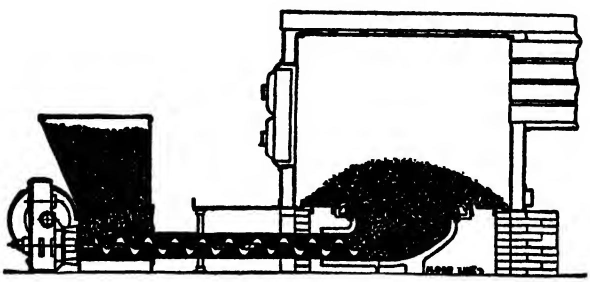


FIG. 5. UNDERFEED SCREW STOKER, HOPPER TYPE, CLASS 2, 3 OR 4

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal. While some of the ash fuses into particles on the surface of the coke as it is released, most of it remains on the hearth or grates and, as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort fuses into a clinker. The

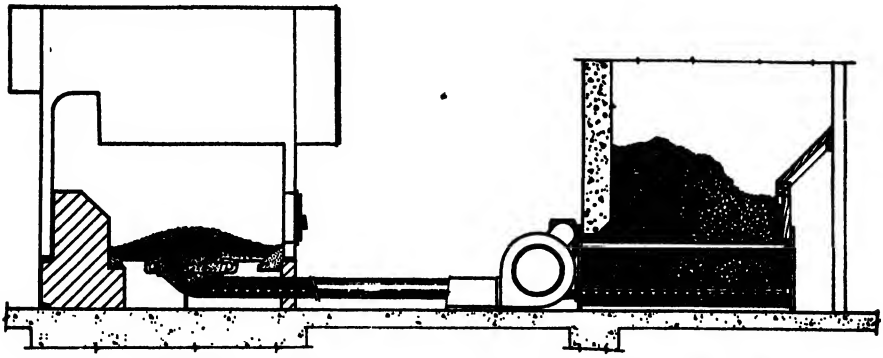


FIG. 6. UNDERFEED SCREW STOKER, BIN TYPE, CLASS 2, 3 OR 4

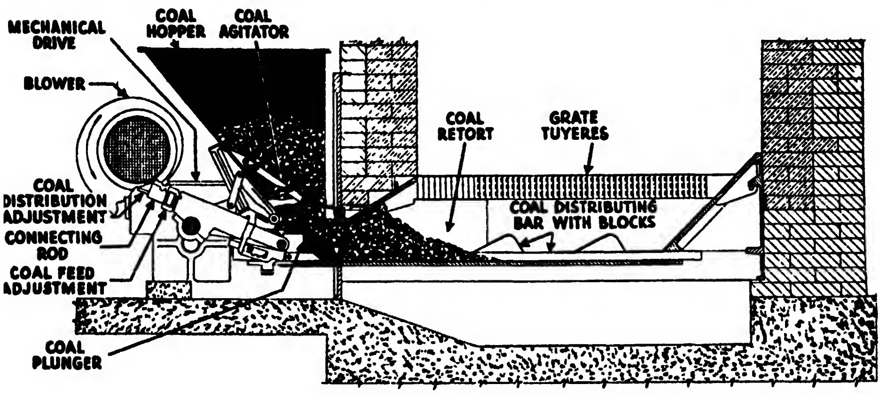


FIG. 7. UNDERFEED SIDE CLEANING STOKER

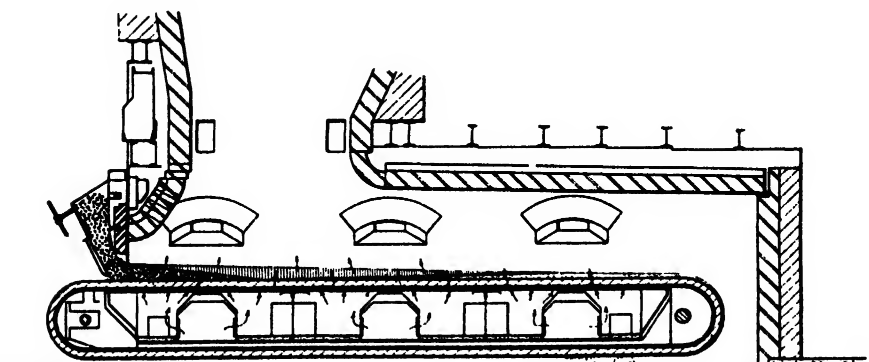


FIG. 8. OVERFEED TRAVELING-GRATE STOKER

temperature in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2, 3 and 4 require manual removal of the ash in clinker form.

In the underfeed side-cleaning stokers the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all bituminous coals while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously or periodically discharged at the sides.

The underfeed rear-cleaning stoker accomplishes combustion in much the same manner as the side-cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed type.

Overfeed flat-grate stokers receive fuel at the front of the grate in a

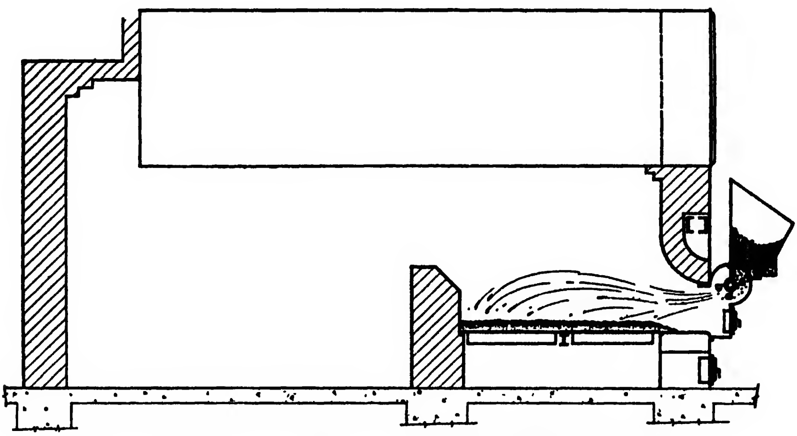


FIG. 9. OVERFEED SPREADER STOKER (ROTOR TYPE)

layer of uniform thickness and move it horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ash pit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze, and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the fuel bed to maintain ignition of the incoming fuel and, frequently a rear combustion arch.

In addition to the use of rocking grates, the overfeed inclined-grate stoker is provided with an ash plate on which ash is accumulated and dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action keeps the fuel bed broken up thereby allowing free passage of air. Because of its agitating effect on the fuel it is not desirable for badly clinkering coals. It usually should be provided with a front arch to ignite the volatile gases.

Combustion Adjustments

The coal feeding rate and air supply to the stoker should be regulated so

as to maintain a balance between the load demand and the heat liberated by the fuel. Under such conditions no manual attention to the fuel bed should be required, other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the maintenance of the correct proportions of air and fuel is essential. It is desirable to supply the minimum amount of air required to properly burn the fuel at the rate of feed.

While there may be only slight variations in the rate at which the coal is being fed due to variations in the size or density of the coal, there may be wide variations in the rate of air flow as the result of changes in fuel bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically com-

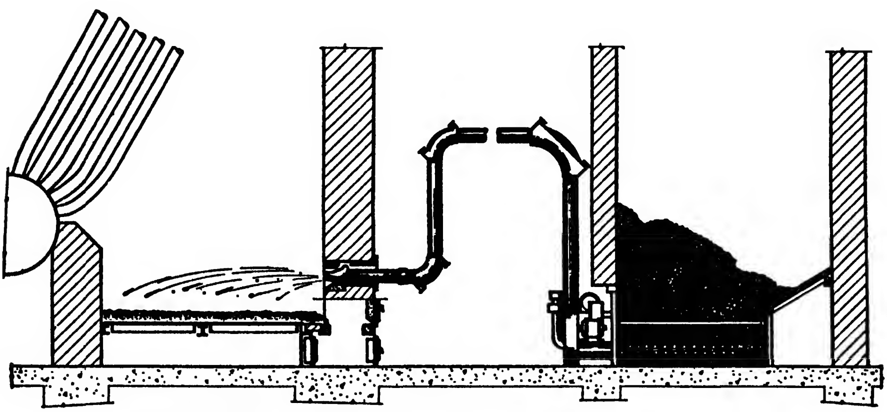


FIG. 10. OVERFEED SPREADER STOKER (PNEUMATIC TYPE)

pensate for these changes in resistance and maintain a constant air fuel ratio. The efficiency of combustion may be determined by analyzing the flue gases as explained in Chapters 11 and 16.

It is desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking, or off, periods of the stoker.

Furnace Design

Although there is considerable variation in stoker, boiler, and furnace design, the stoker industry, from long-time experience, has established certain rules for the proportioning of furnaces for domestic, and commercial stokers. The stoker installer and designer of stoker-fired equipment should give careful consideration to these factors.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour[2].

The empirical formulas for determining these setting heights are:

For burning rates up to 100 lb coal per hour

$$H = 0.1125\,B + 15.75 \qquad (1)$$

For burning rates from 100 to 1200 lb coal per hour

$$H = 0.03\,B + 24$$

where

H = minimum setting height, inches, measured from dead plates to crown sheet for steel boilers. For cast-iron boilers height may be $\frac{3}{4}\,H$.

B = burning rate coal per hour, pounds.

Standards for minimum firebox dimensions and base heights have been formulated by the *Stoker Manufacturers Association* as shown in Fig. 11[2].

In considering these recommendations, it should be understood that they show the average recommended minimum. There are many factors affecting the proper application of stokers to various types of boilers and furnaces, and, in certain instances, setting height or firebox dimensions shown in the standards may be modified without impairing performance. Such modification rests with the experience of the installer, or designer,

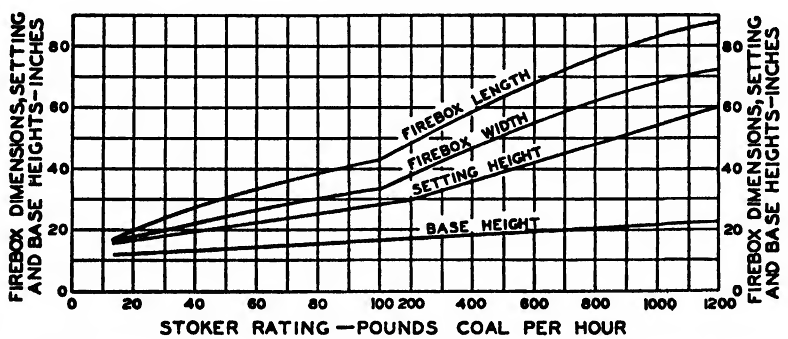


FIG. 11. SUGGESTED MINIMUM FIREBOX DIMENSIONS AND BASE HEIGHTS[a]

[a] For reference in selecting or designing boilers and furnaces for stoker firing. Dimensions shown are for net inside clearance at grate level using coal with heating value of not less than 12,000 Btu per pound. Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums.

with a particular stoker, the type of fuel used, and the construction of the boiler or furnace.

Installation of stokers (particularly smaller sizes) from the side of the boiler or furnace will sometimes facilitate clinker removal.

Rating and Sizing Stokers

The capacity or rating of small underfeed stokers is usually stated as the burning rate in pounds of coal per hour. Codes for establishing uniform methods of rating anthracite and bituminous coal stokers have been adopted by the *Stoker Manufacturers Association*[3].

The Association also has adopted a uniform method of selecting stokers that is published in convenient tables and charts[2]. The required capacity of the stoker is calculated as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{over-all efficiency of stoker and boiler or furnace}} = \text{Stoker burning rate required (pounds of coal per hour)}$$

In determining the total load placed on a stoker-fired boiler by a steam

or hot water heating system, a piping and pick-up factor of 1.33 is commonly used in sizing the stoker, but this factor should be increased at times due to unusual conditions.

Controls

The heat delivery from the stoker of the smallest household type to the largest industrial unit can be regulated accurately with fully automatic controls. The smaller heating applications are controlled normally by a thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure being developed in the furnace or boiler and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically-fired steam boilers. (See Chapter 34.)

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically from liquid fuels. Two methods are employed for the preparation of the

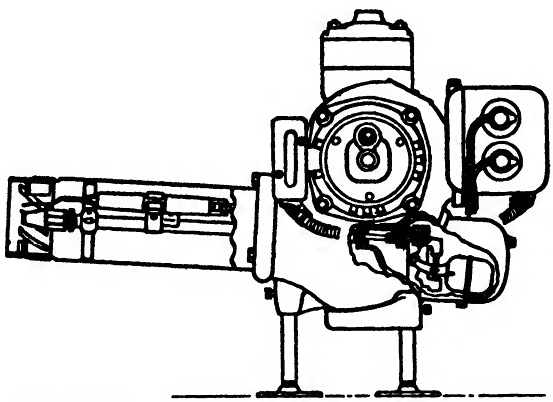


Fig. 12. Low Pressure Atomizing Oil Burner

oil for the combustion process; atomization, and vaporization. The simpler types of burners depend upon the natural chimney draft for supplying the air for combustion. Other burners provide mechanical air supply or a combination of atmospheric, and mechanical. Ignition is accomplished by an electrical spark or hot wire, or by an oil or gas pilot. Some burners utilize a combination of these methods. Continuously operating burners may use manual ignition. Burners of different types operate with luminous or non-luminous flame. Operation may be intermittent, continuous with high-low flame, or continuous with graduated flame.

CLASSIFICATION OF BURNERS

Domestic oil burners may be classified by type of design or operation into the following groups: pressure atomizing or gun, rotary, and vaporizing or pot. These are further classified as mechanical draft, and natural draft.

Pressure Atomizing (Gun Type)

Gun type burners may be divided into two classes, low-pressure, and high-pressure atomization. In the first group, a mixture of oil and primary

air is pumped as a spray through the nozzle at a pressure of 2 to 7 psi. Secondary air is supplied by a fan. Ignition is obtained by means of a high-voltage electric spark used alone, or as primary ignition for a gas pilot. Various features of a low pressure atomizing burner are shown in Fig. 12.

The high-pressure atomizing type, illustrated in Fig. 13, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced, at about 100 psi, and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes is employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

This type of burner utilizes a fan to supply the air for combustion, and ignition is established by a high-voltage electric spark that may be operative continuously while the burner is running, or just at the beginning of the running period. Gun type burners operate on the intermittent *on-off* principle, and with a luminous flame.

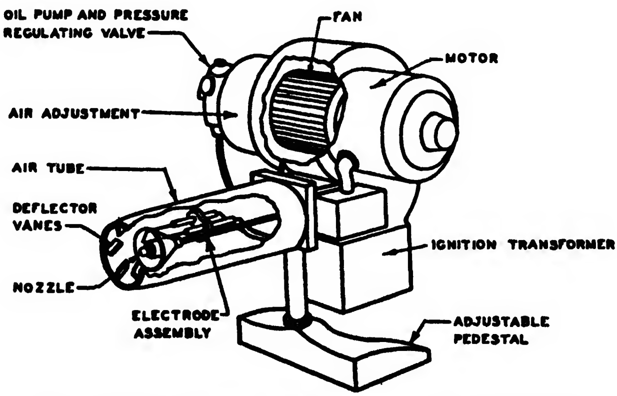


FIG. 13. HIGH-PRESSURE ATOMIZING OIL BURNER

The combustion process is completed in a chamber constructed of refractory material, or stainless steel, this being a part of the installation. Pressure-atomizing burners generally use the distillate oils, No. 1, 2 or 3 grade. (See Chapter 16.)

Rotary Type

This class of burners may be divided into two groups: vertical, and horizontal. Most of the smaller rotary burners are of the vertical type, and use the lighter distillate oils, No. 1 or 2 grade.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 14), the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 15) differs in that combustion takes place in a ring of stainless steel or refractory material, which is placed around the hearth. Dependent upon combustion adjustment, these burners may operate with either a semi-luminous or non-luminous flame.

Both types of vertical rotary burners are further characterized by their

installation within the ash pit of the boiler or furnace. Various types of ignition are utilized, gas and electric, either spark or hot wire. The air for combustion is supplied partially by natural draft, and partially by fan effect of the central spinner element.

Horizontal rotary burners are used principally to burn the heavier oils, Nos. 5 and 6 grades, principally in larger commercial and industrial installations, although domestic sizes are available. Such burners are of the mechanical atomizing type, using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 16).

Horizontal rotary burners commonly use a combination electric-gas ignition system, or are lighted manually. Primary air for combustion is supplied by a blower, and secondary air, often introduced through a

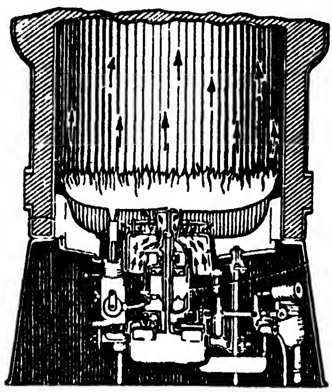


FIG. 14. CENTER FLAME VERTICAL ROTARY BURNER

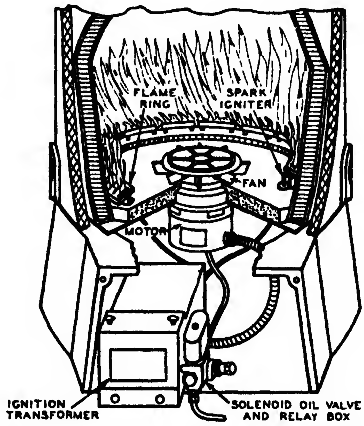


FIG. 15. WALL FLAME VERTICAL ROTARY BURNER

checkerwork in the combustion chamber, is controlled by chimney draft. These burners operate with a luminous flame, usually on high-low or continuous setting.

In larger installations, burners may be installed in multiple in a common combustion chamber. Because of the high viscosity oils used in these burners, it is customary to preheat the oil between the tank and the burner. Preheating when delivering from tank car, or truck, is often required in cold weather.

Vaporizing Burners

In the vaporizing burner, fuel oil is ignited (manually or electrically) and vaporized in a vessel or pot which is open at the top or one side. Heat for vaporization is supplied by the combustion process. Openings in the side walls of the burner admit primary air which forms a rich mixture of air and oil vapors in the burner. Adjacent to the outlet opening sufficient additional or secondary air is admitted to complete combustion. The openings for admitting air are arranged to obtain gradual and intimate mixing of air and oil vapor for combustion with a minimum amount of excess air and resulting high combustion efficiency.

Fuel is fed by gravity from a constant level control valve and the flow is either *on* (at rated capacity) or *off* (at pilot flow) according to the demand of the thermostat. However, the high fire can be reduced and the pilot

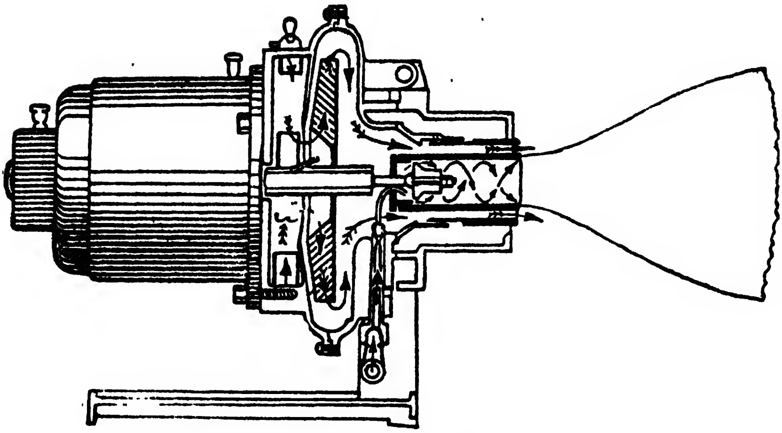


FIG. 16. HORIZONTAL ROTATING CUP OIL BURNER

fire can be increased to give almost any desired control characteristic within the range of the burner. The majority of vaporizing burners are manufactured in sizes up to one gallon per hour input. Most vaporizing burners are limited to use with No. 1 fuel oil having a maximum end point of 625 F and a minimum A.P.I. gravity of 35 deg.

A barometric draft regulator is required to maintain the recommended draft. A draft of not more than 0.06 in. of water column is recommended for most natural draft burners. When burners are equipped with mechanical forced draft, a slightly lower chimney draft can be used. A burner of this type is illustrated in Fig. 17.

Vaporizing burners are adaptable to water heaters, space heaters, and furnaces. Some types have also been applied successfully to conversion installations. The heat output is in the range of requirements for the average or small home.

The modulating flame allows simple manual control by regulation of a metering valve and simplifies the control equipment. Quiet combustion and the absence of moving parts contribute to quiet operation when the heating device is located in the living quarters.

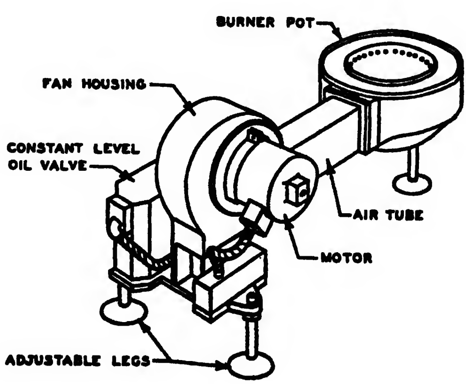


FIG. 17. VAPORIZING POT-TYPE BURNER

The ability to operate on natural draft and gravity feed of the fuel, makes possible the use of these burners where electric current is not available or is unreliable. However, most furnaces are thermostatically controlled and many are provided with mechanical draft.

Oil-Fired Boiler and Furnace Units

A number of types of specially designed oil-fired boiler-burner and furnace-burner units are available. Various locations of burners will be noted in such units; some having the combustion chamber and burner at the top, some at the bottom, and some at the center of the appliance. One type of boiler-burner unit is shown in Fig. 18. The coordinated design of boiler (or furnace) and burner elements insures the optimum in operating characteristics, and the maintenance of balanced performance. This type of equipment usually has more heating surface, better flue proportions and gas travel than conventional boilers or furnaces. Some

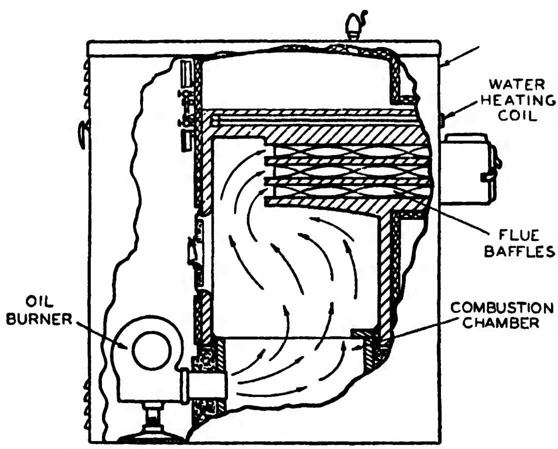


FIG. 18. TYPICAL BOILER-BURNER UNIT

of the better conversion installations, however, may equal the unit type in performance.

Operating Requirements for Oil Burners

The U. S. Department of Commerce in conjunction with the oil burner and heating appliance industries has established commercial standards for conversion burners and burner-appliance units which cover installation, construction and performance tests[4].

Combustion Process

Efficient combustion must produce a clean flame and use a relatively small excess of air, *i.e.*, between 25 and 50 per cent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the liquid fuel to the gaseous state by the application of heat before the oil vapor mixes with air to any extent and, if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced.

In an atomizing burner the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result is the ability to burn more and heavier oil within a given combustion space. Because the air enters the combustion chamber with the liquid fuel particles, mixing, vaporization and burning occur all at once in the same space. This produces a luminous flame. A deficient amount of air is indicated by a dull red or dark orange flame with smoky tips.

An excessive supply of air may produce a brilliant white flame or a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be detected, it is not possible to distinguish, by eye, the effect of the finer adjustment which competent installation requires.

Combustion Adjustments

The present-day oil burner with mechanical oil and air supply, properly installed and equipped with an automatic draft regulator, is capable of maintaining efficient combustion for a considerable period following the initial adjustments of oil and air. Eventually certain changes may occur, however, that will cause the per cent of excess air to decrease below allowable limits. A decrease in air supply while the oil delivery remains constant, or an increase in oil delivery while the air supply remains constant, will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment the more critical it will be. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (1) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (2) erosion of atomizing nozzle, (3) fluctuations in by-pass relief pressures, and (4) possible variations in methods of atomization. Any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service.

The following factors may influence the air supply: (1) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney, and changes in draft resistance of boiler due to partial stoppage of the flues), and (2) changes in air inlet adjustments at the fan.

Air leakage into the boiler or furnace setting should be reduced to a minimum. The amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02–0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Even though a fan is generally used to supply the air for combustion, in most oil burners, the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure that the draft will be uniform within the limits given if maximum efficiency throughout the heating season is to be maintained.

Measurement of the Efficiency of Combustion

Since efficient combustion is based upon a clean flame and definite proportions of oil and air employed, it is possible to determine the results by analyzing the combustion gases. It is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations show from 8 to 10 per cent CO_2. Taking into account the potential hazard of low excess air (high CO_2), a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners.

Combustion Chamber Design

With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

The atomizing burner is dependent upon the surrounding heated refractory or firebrick surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the firebrick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat, but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber resembles in shape the outline of the flame. In this way as much firebrick as possible is close to the flame so it may be kept hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning, and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. When installing some of the vertical rotary burners the manufacturer's instructions must be followed carefully when installing the hearth, as in this class successful performance depends upon this factor.

Boiler Settings

As the volume of space available for combustion is a determining factor in oil comsumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ash pit volume; in new installations the boiler may be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horse power, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btu per cubic foot per hour. At times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 fps. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best

adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or firebrick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Controls

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 34.

GAS-FIRED HEATING EQUIPMENT

A gas burner is defined by the *American Gas Association* as "a device

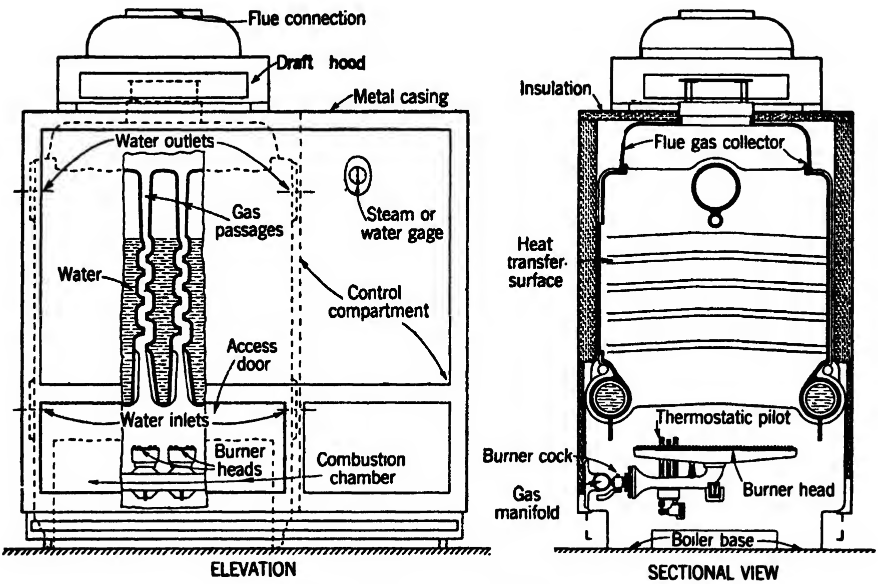


FIG. 19. GAS-FIRED BOILER

for the final conveyance of the gas, or a mixture of gas and air, to the combustion zone." Burners used for domestic heating are of the atmospheric injection, luminous flame, or power burner types.

The use of gas has resulted in the production of a number of types of domestic gas heating appliances, and systems. These may be classified in types designed for central heating plants, and those for unit application. Gas-designed units and conversion burners are available for the several kinds of central systems. Unit heaters, space heaters and circulators may be had for installation in the space being heated.

Central Heating Systems

Boilers and furnaces specially designed for gas-firing incorporate design features for obtaining maximum efficiency and performance. Small flue passes to secure good heat transfer, the use of materials resistant to the corrosive effects of products of combustion, and draft hoods are notable

features. Control equipment includes gas pressure regulators, automatic pilots, and limit controls designed to protect the appliance and to insure safety of operation. A boiler designed for gas-burning is illustrated in Fig. 19.

Conversion burners are usually complete burner and control units designed for installation in existing boilers and furnaces. Burner heads are of circular or rectangular shape in order to fit in the space available. Single port burners, discharging the flame against a ceramic, stainless steel, or cast iron target, have become popular in the past few years. The control equipment is generally the same as for gas boilers and furnaces. Various baffles made of clay radiants or metal are used for the purpose of guiding the products of combustion along the heating surface in the firebox or flues. Automatic air dampers are supplied on many models to prevent flow of air into the firebox when the burner is not operating. A typical gas conversion burner is shown in Fig. 20.

Burners of this type are available in sizes ranging from 50,000 to 400,000 Btu per hour capacity. Burners of even larger capacity, for use with natural

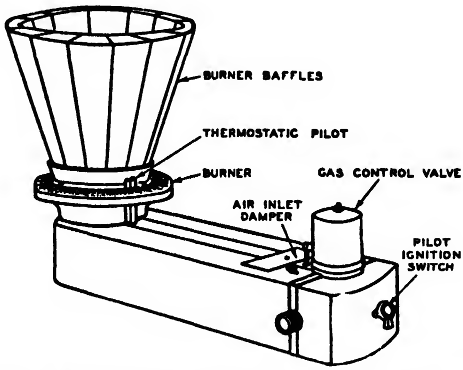


FIG. 20. TYPICAL GAS CONVERSION BURNER

gas in large steel boilers, are usually engineered by the local utility or contractor. They are available in an infinite number of sizes because the burner may be an assembly of multiple burner heads filling the entire firebox.

Domestic sizes of conversion burners should conform to American Standard Listing Requirements for Conversion Burners, A.S.A. Z21.17-1948 and installation should be made in accordance with American Standard Requirements for Installation of Domestic Gas Conversion Burners. A.S.A. Z21.8-1948.

Draft hoods, conforming to American Standard Requirements, should be installed in place of the dampers used with a solid fuel.

One form of central heating system is the warm air floor furnace[5]. The use of these furnaces is adaptable to mild climates or for auxiliary heating or heating of single rooms in colder climates. They are used for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility. With the usual type the register is installed in the floor, the heating element, gas piping, and also the flue gas vent piping being suspended below the floor.

Unit Type Heaters

Space heaters may be used for auxiliary heating, but in many cases are installed for furnishing heat to entire buildings. With the exception of wall heaters, they are semi-portable.

Parlor heaters or *circulators* are usually of the cabinet type. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base, passes up inside the jacket around a heating section, and out of the heater at, or near, the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass around baffles within the heating element, and out the flue at the back near the top. They are well adapted for residence room heating and also for stores and offices.

Unvented type circulators should not be used in residences unless provision is made to remove the excess moisture caused by release of flue gases into the living quarters.

Radiant heaters give off a considerable portion of their heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to seven or more radiants. An atmospheric burner is supported near the center of the base. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are portable; however, there are also types which are encased in a jacket with a grilled front and fit into the wall.

Gas-fired steam and hot water radiators are other types of room heating appliances. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to steam or hot water radiators. They are usually constructed of sheet metal hollow sections. The products of combustion circulate through the sections and are discharged from a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Unit heaters are used extensively for heating large spaces such as stores, garages, and factories. These heaters consist of a burner, heat exchanger, fan for distributing the air, draft hood, automatic pilot, and controls for burners and fan. They are usually mounted in an elevated position from which the heated air is directed downward by louvers. Some unit heaters are suspended from the ceiling, and others are free-standing floor units of the heat tower type.

Unit heaters are available in two types, classified according to their use, with, or without ducts. Only those types of unit heaters tested and approved as warm air furnaces can be connected safely to ducts, as they have sufficient blower capacity to deliver an adequate air supply against duct resistance and are equipped with limit controls.

Combustion Process and Adjustments

Most domestic gas burners are of the atmospheric injection (Bunsen) type in which primary air is introduced, and mixed with the gas in the throat of the mixing tube. A ratio of about 3 parts primary air to 1 part

gas for manufactured gas, and a 5½ to 1 ratio for natural gas, are generally used as theoretical values. For normal operation of most atmospheric type burners 40 to 60 per cent of the theoretical value of primary air will give best operation. The amount of excess air required in practice depends upon several factors, notably: uniformity of air distribution and mixing, direction of gas travel from burner, and the height and temperature of combustion chamber.

Secondary air is drawn into gas appliances by natural draft. As with other fuels, excess secondary air constitutes a loss, and should be reduced to a proper minimum, which usually cannot be less than 25 to 35 per cent if the appliance is to meet *A.S.A.* approval. Yellow flame burners depend upon secondary air, alone, for combustion.

The flame produced by atmospheric injection burners is non-luminous. Air shutter adjustments for manufactured gas should be made by closing the air shutter until yellow flame tips appear and then by opening the air shutter to a final position at which the yellow tips just disappear. This type of flame obtains ready ignition from port to port and also favors quiet flame extinction. When burning natural gas the air adjustment is generally made to secure as blue a flame as obtainable.

Little difficulty should be had in maintaining efficient combustion when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and the rate of heat supply is nominally constant. Because the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft hood insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft hood is also helpful in controlling the amount of excess air and preventing back drafts that might extinguish the flame. (See Chapter 16.)

Due to the use of draft hoods and gas pressure regulators both the input and combustion conditions of gas appliances are maintained quite uniform until deposits of dirt, corrosion, or scale accumulate in the air inlet openings, burner ports, or on the heating surface. Periodic cleaning is necessary to keep any gas appliance in proper operating condition.

Measurement of the Efficency of Combustion

The efficiency of combustion may be judged from the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. The CO_2 and O_2 may be obtained by means of an Orsat apparatus but the CO must be determined by more accurate equipment. It is customary to use simple indicators to determine whether CO is present and to make adjustments of the appliances to reduce the CO below 4/100 of one per cent before continuing tests in which the CO_2 and O_2 can then be found by use of the Orsat apparatus. Since the ultimate CO_2 for any gas depends on the carbon-hydrogen ratio the quality of the combustion should not be judged from the value of the CO_2 in the flue gas without reference to the ultimate CO_2 obtainable. Practical values of CO_2 will usually be from 8 to 14 per cent depending on the gas used.

Ratings for Gas Appliances

Input rating for a gas appliance is established by demonstrating that the appliance can meet the Approval Requirements of the *A.S.A.* The tests are conducted at the *A.G.A. Testing Laboratories.* Output rating is

TABLE 1. CAPACITY OF GAS PIPING

Length of Pipe in Feet	Nominal Diameter of Pipe in Inches				
	¾	1	1¼	1½	2
	Capacity—Cu Ft Per Hr with a 0.6 Sp Gr Gas and Pressure Drop of 0.3″ Water Column				
15	172	345	750		
30	120	241	535	850	
45	99	199	435	700	
60	86	173	380	610	
75	77	155	345	545	
90	70	141	310	490	
105	65	131	285	450	920
120		120	270	420	860
150		109	242	380	780
180		100	225	350	720

determined from the approved input and an average efficiency stated in the Approval Requirements and is the heat available at the outlet.

Sizing Gas-Fired Heating Plants

Although gas-burning equipment usually is completely automatic, maintaining the temperature of rooms at a predetermined figure, there are some manually controlled installations. In order to overcome effectively the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up loads and consequently, it is possible to use a lower selection factor. For a gas-fired boiler or furnace under thermostatic control a factor of 20 to 25 per cent is usually sufficient for pick-up allowance.

In those installations, in mild climates where 100 per cent outside air is used, furnaces should be of larger size in order to provide adequate capacity and quick pick-up under intermittent heating conditions.

The factor to be allowed for loss of heat from piping will vary somewhat, the proportionate amount of piping installed being greater for small installations than for large ones. For selection factors to be added to installed radiation under thermostatic control see Chapter 18.

TABLE 2. MULTIPLIERS FOR VARIOUS SPECIFIC GRAVITIES
For Use With Table 1

Specific Gravity	Multiplier	Specific Gravity	Multiplier
.35	1.31	1.00	.775
.40	1.23	1.10	.740
.45	1.16	1.20	.707
.50	1.10	1.30	.680
.55	1.04	1.40	.655
.60	1.00	1.50	.633
.65	.962	1.60	.612
.70	.926	1.70	.594
.75	.895	1.80	.577
.80	.867	1.90	.565
.85	.841	2.00	.547
.90	.817	2.10	.535

Appliances used for heating with gas should bear the approval seal of the *A.G.A. Testing Laboratories* on the manufacturer's nameplate, together with the official input and output ratings. It is not permissible to operate a gas heating unit above its stated rating. It may be necessary to operate below this rating at elevations above 2000 ft unless the appliance has been tested and approved for operation at altitudes up to 5200 ft as shown on name plate.

Installations should be made in accordance with recommendations shown in the publications of the *American Gas Association.*

Controls

Temperature controls for gas burners are described in Chapter 34. Some central heating plants are equipped with push-button or other manual control. The main gas valve may be of either the snap action or throttling type. Automatic electric ignition is available.

SIZING OF GAS PIPING

Piping for gas appliances should be of adequate size and so installed as to provide a supply of gas sufficient to meet the maximum demand without undue loss of pressure between the point of supply (the meter) and the burner.

The size of gas pipe necessary to install depends upon the following factors:

1. Maximum gas consumption to be provided.
2. Length of pipe and number of fittings.
3. Allowable loss in pressure from the outlet of the meter to the burner.
4. Specific gravity of the gas.

To obtain the cubic feet per hour of gas required by the burner divide the Btu input at which the burner will be adjusted by the average Btu heating value per cubic foot of the gas.

Capacities of different sizes and lengths of pipe, in cubic feet per hour, with a pressure drop of 0.3 in. of water column for a gas of 0.60 sp. gr. are shown in Table 1. In adopting a 0.3 in. pressure drop, due allowance for an ordinary number of fittings was made.

To convert the figures given in Table 1 to capacities for another gas of different specific gravity, multiply the tabular values by the multipliers shown in Table 2.

FUEL BURNING RATES

The burning rate for automatic fuel burning devices is determined by the gross heat output required of the boiler, or furnace, to carry the net heating load plus allowances for system losses, and pick-up. General values for these allowances previously have been noted. Detailed information for piping and pick-up allowances for steam, and hot water systems is given in Chapter 18 and for warm air systems in Chapters 21 and 22.

When the gross output, operating efficiency, and heat value of the fuel are known, the required rate of burning can be determined by means of Figs. 21, 22 and 23 for the several fuels.

As the rate of fuel burning is directly proportional to the load for a given efficiency, these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales.

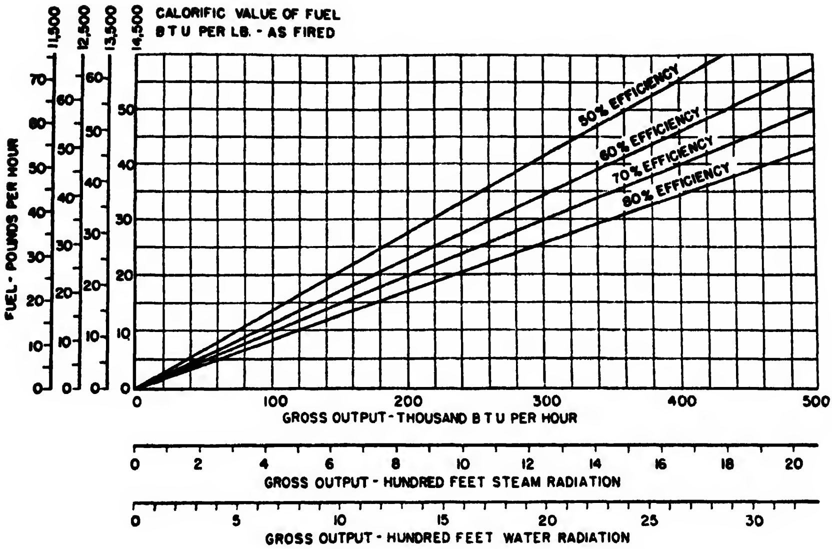


FIG. 21. COAL FUEL BURNING RATE CHART

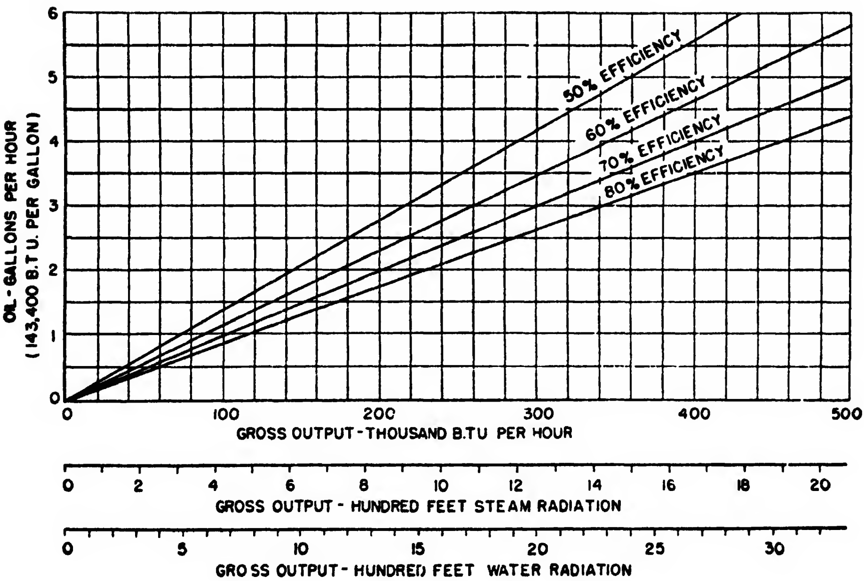


FIG. 22. OIL FUEL BURNING RATE CHART[a]

[a] This chart is based upon No. 3 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 2 oil (141,000 Btu per gallon) 1.017; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

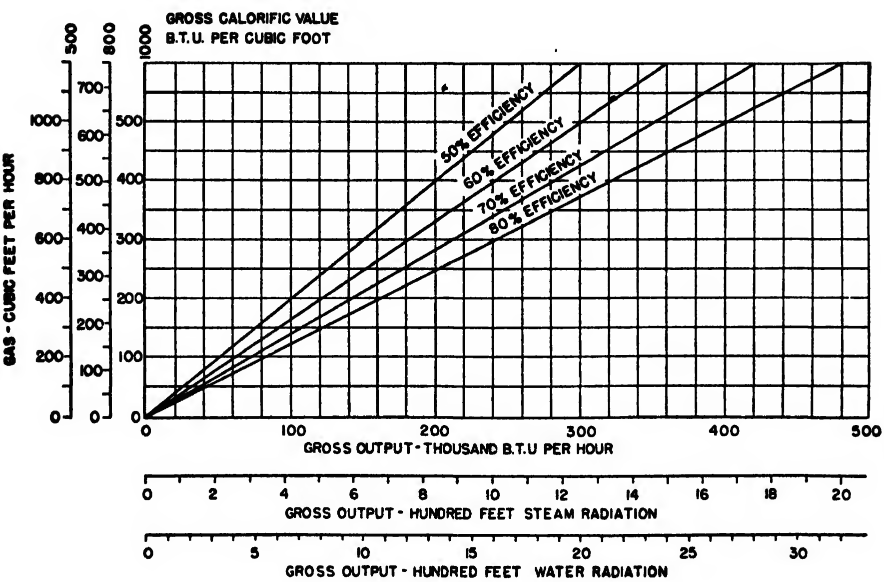


FIG. 23. GAS FUEL BURNING RATE CHART

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fue' at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stroker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

REFERENCES

[1] Domestic Burners for Pennsylvania Anthracite (Underfeed Type), (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS48-40).

[2] *Stoker Manufacturers Association Manual:* Industry Standards, Recommended Practices, Technical Information. Published by *Stoker Manufacturers Association*, 307 N. Michigan Ave., Chicago 1, Ill.

[3] Code for Determination of Rated Capacities of Anthracite Underfeed Stokers, adopted June 1, 1944, and a Code for Determination of Rated Capacities of Bituminous Underfeed Strokers, adopted May 3, 1944. See *Stroker Manufacturers Association Manual.*

[4] Automatic Mechanical Draft Oil Burners Designed for Domestic Installations (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS75-42). Flue Connected Oil Burning Space Heaters Equipped with Vaporizing Pot Type Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS101-43). Warm-Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS(E)104-43). Oil-Burning Floor Furnaces equipped with Vaporizing Pot-Type Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard C. S. 113-44).

[5] Gas Floor Furnaces, Gravity Circulating Type (*U. S. Department of Commerce, National Bureau of Standards*, (Commercial Standard No. CS99-42).

[6] American Standard Requirements for Installation of Domestic Gas Conversion Burners (A.S.A. Z21.8-1948 American Standard Association).

BIBLIOGRAPHY

Performance Expectancy of Domestic Underfeed Stokers for Anthracite, by Allen J. Johnson (*Transactions, A.I.M.E.*, Coal Division, Vol. 119, 1936).

The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—Burning Tests on Four Typical Coals, by R. A. Sherman, E. R. Kaiser and H. R. Limbacher, Technical Report No. 1, Bituminous Coal Research, Inc. (July, 1937) Part II.

Oil Fuels and Burners, by James A. Moyer (McGraw-Hill).

Handbook of Oil Burning, by Harry F. Tapp.

A Study of the Oil Burner as Applied to Domestic Heating, by Arthur H. Senner (Technical Bulletin 109, *U. S. Department of Agriculture*).

Progress in Domestic Oil Heating, by Rene J. Bender (*Mechanical Engineering*, October, 1942).

A.S.H.V.E. RESEARCH REPORT No. 907—Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 517).

A.S.H.V.E. RESEARCH REPORT No. 925—A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, VOL. 38, 1932, p. 317).

Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, by L. E. Seeley, J. H. Powers and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 75).

Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, VOL. 40, 1934, p. 319).

Oil Burning in Residences, by D. W. Nelson (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 355).

A Study of Oil-Fired Heating Boilers, by R. C. Cross and W. R. Lyman (*Heating and Ventilating*, October, 1931).

Gaseous Fuels, *American Gas Association.*

Comfort Heating, *American Gas Association.*

Approval Requirements of Central House Heating Gas Appliances, *American Standards Association.*

A Method for Determining Fuel Burning Rates in Heating Boilers Fired by Automatic Devices, by R. C. Cross (*Heating and Ventilating*, January, 1932).

Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 185).

CHAPTER 18

HEATING BOILERS, FURNACES, SPACE HEATERS

BOILERS: Construction, Types, Design Considerations, Testing and Rating Codes, Efficiency, Rating, Selection, Space Limitations, Connections and Fittings, Erection, Operation and Maintenance. FURNACES: Types, Materials and Construction, Ratings, Testing and Rating Codes, Efficiency, Design Considerations, Humidification Equipment. SPACE HEATERS: Types: Solid Fuel, Oil, Gas; Materials and Construction, Testing and Rating, Design Considerations; Special Features; Installation

IN presenting the subject of Boilers, Furnaces and Space Heaters this chapter is divided into three parts; the first dealing with boilers, the second treating warm air furnaces, and the third covering space heaters.

HEATING BOILERS

Steam and hot water boilers for low pressure heating are built of steel or cast-iron in a wide variety of types and sizes, many of which are illustrated in the *Catalog Data Section*.

CONSTRUCTION

The nationally recognized code governing the construction of low-pressure steel and cast-iron heating boilers is the *ASME* Boiler Construction Code for Low Pressure Heating Boilers. Some states and municipalities have their own codes which apply locally but these are usually patterned after the *ASME* Code.

The maximum allowable working pressures are limited by the *ASME* Code to 15 psi for steam and 30 psi for hot water heating boilers. Hot water boilers may be used for higher working pressures, for heating purposes or for hot water supply, when designed and tested for the higher pressure.

TYPES OF HEATING BOILERS

Heating boilers are classified in a number of different ways, such as:

1. According to materials of construction. These are steel and cast-iron. Very few non-ferrous boilers are made.

2. According to the fuels for which the boilers are designed. These are coal, hand fired or stoker fired; oil; gas; or wood. Some boilers are designed specifically for one fuel but many boilers are designed for more than one fuel.

3. According to the specific purpose or application for which the boiler is used, such as space heating or domestic hot water supply.

4. According to the design or construction of the boiler such as, sectional, round, fire-tube, water-tube, magazine feed, Scotch, etc.

Cast-Iron Boilers

Cast-iron boilers are generally classified as:

1. Square or rectangular boilers with vertical sections and rectangular grates, commonly known as sectional boilers.

2. Round boilers with horizontal pancake sections and circular grates.

Cast-iron boilers are usually shipped in sections and assembled at the place of installation. In the majority of boilers the sections are assembled with push nipples and tie rods. Many sectional boilers are provided with large push nipples at top to permit the circulation of water between adjacent sections at both the water line and bottom of the boiler, which is necessary to enable the use of an indirect water heater with the boiler for summer-winter hot water supply. Round and sectional boilers may be increased in size by the addition of sections and corresponding plate work.

Small sectional type boilers are available with *wet-base* construction, wherein the ashpit or combustion chamber sides and bottom are surrounded by extensions of the water legs of the boiler sections and thus no separate base is required. This type of construction permits the boiler to be set directly on a wood or composition floor without danger of fire. The *wet-base* also provides some additional heating surface.

Capacities of cast-iron boilers range generally from capacities required for small residences up to about 12,000 sq ft of steam radiation. There are a few boilers made with capacities up to 18,000 sq ft of steam radiation. For larger loads, boilers must be installed in multiple.

Steel Boilers

Steel boilers may be of the fire-tube type, in which the gases of combustion pass through the tubes and the boiler water circulates around them, or of the water-tube type, in which the gases circulate around the tubes and the water passes through them.

Either the fire-tube or water-tube type may be designed with integral water jacketed furnaces or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water jacketed furnaces are called portable firebox boilers and are the most commonly used type. They are usually shipped in one piece, ready for piping. Refractory furnaces are usually installed in refractory lined furnace boilers after they are set in place.

Capacities of steel boilers range from those required for small residences up to about 35,000 sq ft of steam radiation.

Boilers for Special Applications

One of these is known as the *magazine feed boiler* developed for the burning of small sizes of anthracite and coke and has a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types burning coal of buckwheat size. Special attention must be given to proper chimney sizes and connections in order to insure adequate draft.

Boilers for *hot water supply* are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-forming properties of the water supplied and the temperatures maintained. If low water temperatures are maintained the life of the heater will be much longer due to decreased scale formation and minimized corrosion. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the

steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam or a forced circulation hot water heating system is installed, the domestic hot water may be heated by an indirect heater attached to the boiler. For most satisfactory performance in the steam system, this heater is placed just below the water line of the boiler. In a forced circulation hot water system, it should be located as high as possible with respect to the boiler.

BOILER DESIGN CONSIDERATIONS

Furnace Design

Good efficiency and proper boiler performance are dependent on correct furnace design. There must be sufficient volume for burning the particular fuel which is used, and means to obtain a thorough mixing of air and gases at a high temperature and at a velocity low enough to permit complete combustion of all the volatiles. For hand fired boilers, the furnace volume should be large enough to hold sufficient fuel for reasonably long firing periods. (See Chapters 16 and 17.)

Heating Surface

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only as *indirect* or convection surface. The amount of heating surface, its distribution, and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow increases boiler efficiency and reduces stack temperature, but increases the draft loss through the boiler.

Heat Transfer Rate

Practical average over-all heat transfer rates expressed in Btu absorbed per square foot of surface per hour will average about 3300 for hand fired boilers and 4000 for mechanically fired boilers when operating at *design load*. When mechanically fired boilers are operating at *maximum load* as defined in this chapter under heading *Selection of Boilers*, these values will run between 5000 and 6000. Boilers operating under favorable conditions at these heat transfer rates will give exit gas temperatures that are

considered consistent with good practice, although there are boilers which have high efficiencies and also operate at higher transmission rates.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code, and an oil fuel testing code.

ASHVE Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June, 1929)[1], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics.

ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers —Code No. 3—(Edition of 1929)[1] is intended for use with ASHVE Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[2]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

The ASHVE Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[3], (Adopted June, 1932), is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

The ASHVE Standard Code for Testing Stoker-Fired Steam Heating Boilers[4], (Adopted June, 1938), is intended to provide a test method for determining the efficiency and performance characteristics of any stoker and boiler combination burning any type of solid fuel such as anthracite or bituminous coal.

The *Steel Boiler Institute, Inc.* has adopted a Rating Code for Commercial Steel Boilers and Residential Steel Boilers and for Testing Oil-Fired Residential Steel Boilers (Fifth Edition as Revised Jan. 1 1948). The commerical boilers (defined as those having 129 to 2500 sq ft of heating

TABLE 1. *SBI* NET RATING DATA FOR RESIDENTIAL STEEL BOILERS—OIL FIRED[a]

SBI Net Rating			Minimum Furnace Volume Cu Ft	Heating Surface Sq Ft
Sq Ft Steam	Sq Ft Water	Btu		
275	440	66000	2.5	16
320	510	77000	2.9	19
400	640	96000	3.6	24
550	880	132000	5.0	32
700	1120	168000	6.4	41
900	1440	216000	8.2	53
1100	1760	264000	10.0	65
1300	2080	312000	11.8	77
1500	2400	360000	13.6	88
1800	2880	432000	16.4	106
2200	3520	528000	20.0	129
2600	4160	624000	23.6	153
3000	4800	720000	27.3	177

[a] Stoker-fired and Gas-fired *SBI* Net Rating not greater than Oil-fired. Hand-fired, *SBI* Net Rating (Steam) not greater than 14 times the square feet of heating surface.

TABLE 2. *SBI* RATINGS FOR COMMERCIAL STEEL BOILERS

Mechanically Fired								Heating Surface Sq Ft	Hand Fired						Minimum Grate Area Sq Ft	Flow Tappings	
SBI Rating			*SBI* Net Rating			Minimum Furnace Volume Cu Ft[a]	Minimum Furnace Height[b] In.		*SBI* Rating			*SBI* Net Rating				Steam Outlet I.P.S. In.	Steam Return I.P.S.[c] In.
Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu				Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu			
2,190	3,500	526,000	1,800	2,880	432,000	15.7	26	129	1,800	2,880	432,000	1,500	2,400	360,000	7.9	6	3
2,680	4,280	643,000	2,200	3,520	528,000	19.2	28	158	2,200	3,520	528,000	1,830	2,930	439,000	8.9	6	3
3,160	5,050	758,000	2,600	4,160	624,000	22.6	29¼	186	2,600	4,160	624,000	2,170	3,470	521,000	9.7	6	3
3,650	5,840	876,000	3,000	4,800	720,000	26.1	29½	215	3,000	4,800	720,000	2,500	4,000	600,000	10.5	6	3
4,250	6,800	1,020,000	3,500	5,600	840,000	30.4	30	250	3,500	5,600	840,000	2,920	4,670	700,000	11.4	6	3
4,860	7,770	1,166,000	4,000	6,400	960,000	34.8	30½	286	4,000	6,400	960,000	3,330	5,330	800,000	12.2	6	3
5,470	8,750	1,313,000	4,500	7,200	1,080,000	39.1	31¼	322	4,500	7,200	1,080,000	3,750	6,000	900,000	13.4	6	3
6,080	9,720	1,459,000	5,000	8,000	1,200,000	43.5	31¾	358	5,000	8,000	1,200,000	4,170	6,670	1,000,000	14.5	6	3
7,290	11,660	1,750,000	6,000	9,600	1,440,000	52.1	32¾	429	6,000	9,600	1,440,000	5,000	8,000	1,200,000	16.4	8	4
8,500	13,600	2,040,000	7,000	11,200	1,680,000	60.8	34	500	7,000	11,200	1,680,000	5,830	9,330	1,400,000	18.1	8	4
10,330	16,520	2,479,000	8,500	13,600	2,040,000	73.8	35½	608	8,500	13,600	2,040,000	7,080	11,330	1,700,000	20.5	8	4
12,150	19,440	2,916,000	10,000	16.000	2,400,000	86.8	37½	715	10,000	16,000	2,400,000	8,330	13,330	2,000,000	22.5	8	4
15,180	24,280	3,643,000	12,500	20,000	3,000,000	108.5	40¼	893	12,500	20,000	3,000,000	10,420	16,700	2,500,000	25.6	8	4
18,220	29,150	4,373,000	15,000	24,000	3,600,000	130.2	43	1,072	15,000	24,000	3,600,000	12,500	20,000	3,000.000	28.4	8	4
21,250	34,000	5,100,000	17,500	28,000	4,200,000	151.8	46	1,250	17,500	28,000	4,200,000	14,580	23,330	3,500,000	30.9	8	4
24,290	38,860	5,830,000	20,000	32,000	4,800,000	173.5	48¾	1,429	20,000	32,000	4,800,000	16,670	26,670	4,000,000	33.2	8	4
30,360	48,570	7,286,000	25,000	40,000	6,000,000	216.9	54½	1,786	25,000	40,000	6,000,000	20,830	33,330	5,000,000	37.4	10	6
36,430	58,280	8,743,000	30,000	48,000	7,200,000	260.3	60½	2,143	30,000	48,000	7,200,000	25,000	40,000	6,000,000	41.2	10	6
42,500	68,000	10,200,000	35,000	56,000	8,400,000	303 6	66	2,500	35,000	56,000	8,400,000	29,170	46,670	7,000,000	44.7	10	6

[a] Oil, gas or bituminous stoker-fired coal. Minimum furnace volumes for anthracite, stoker-fired, are not specified in this code.
[b] Bituminous, stoker-fired.
[c] The tapping sizes shown for boilers having 129 to 500 sq ft of heating surface, inclusive are adequate for forced hot water.

surface) are rated in square feet (steam) on the basis of heating surface with limitations set for grate area, furnace volume, and furnace height. The residential boilers (defined as those having not more than 177 sq ft of heating surface) are rated on tests for oil-fired boilers, with limitations in relation to heating surface and testing conditions. Stoker-fired and gas-fired *residential* boilers are rated (*SBI* Net Rating) not in excess of the oil-fired rating. Hand-fired residential boilers are rated (*SBI* Net Rating) not greater than 14 times the heating surface.

Tables 1 and 2 show the *SBI* ratings of residential and commercial steel boilers respectively.

The *Institute of Boiler and Radiator Manufacturers* has adopted a Code[5] for rating cast-iron heating boilers based upon performance obtained under controlled test conditions. This Code applies to all sectional cast-iron heating boilers except those of magazine feed type.

The *Gross I = B = R* Output is obtained by test and is subject to certain limiting factors. For hand fired boilers, the number of boilers of a series to be tested, the minimum over-all efficiency, the minimum time limit (the time an *Available Fuel Charge* will last when burned at a rate which will produce the *Gross I = B = R Output*), the chimney area and height, and the draft in the stack are all subject to the limits established in the Code. Tests are run using anthracite coal of standard specification. Bituminous coal and coke ratings are the same as for anthracite coal.

For automatically fired boilers, the number of boilers of a series to be tested, the flue gas temperature and analysis, the minimum over-all efficiency, the draft loss through the boiler, and the heat release in the combustion chamber are subjected to limitation by the Code[5]. Automatically fired boiler ratings are established by oil fired tests using gun type oil burners and commercial grade No. 2 fuel oil. Stoker fired and gas fired ratings (where no *A.G.A.* Rating is published) are based on the *Gross I = B = R Output* obtained by oil fired tests.

The *Net I = B = R Rating* is determined from the *Gross I = B = R Output* by applying specified *Piping and Pickup Factors* which range from 2.36 to 1.40 for hand fired boilers and from 1.56 to 1.288 for automatically fired boilers. In both cases, the factor decreases as the boiler size increases. Table 3 is abstracted from the *I = B = R Rating Tables* in the Code and illustrates the relationship between *Net I = B = R Rating* and *Gross I = B = R Output.*

The *American Gas Association* has adopted a method of rating gas designed boilers based upon performance under tests. This is described in Approval Requirements for Central Heating Gas Appliances.

The *Heating, Piping and Air Conditioning Contractors National Association* has adopted a method of rating boilers based on their physical characteristics for those boilers that are not rated in accordance with the *SBI* or *I = B = R* Codes. Ratings are expressed on a Net Load basis in square feet of steam radiation.

BOILER EFFICIENCY

The term efficiency as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

TABLE 3. $I=B=R$ RATING TABLE

Net $I=B=R$ Rating		Piping Factor	Hand-Fired					Automatic-Fired			
Sq Ft Steam	1000 Btu		Piping and Pickup[a] Factor	Gross $I=B=R$ Output 1000 Btu	Time Available Fuel Will Last, Hr	Maximum Stack Height[b] Ft	Minimum Stack Area[b] Sq In.	Piping and Pickup[a] Factor	Gross $I=B=R$ Output 1000 Btu	Minimum Stack Area[b] Sq In.	Maximum Allowable Draft Loss
1	2	3	4	5	6	7	8	9	10	11	12
100	24.0	1.300	2.360	56.6	7.50	29.0	50.0	1.560	37.4	50.0	0.044
250	60.0	1.293	2.341	140.5	6.77	33.0	50.0	1.548	92.9	50.0	0.051
400	96.0	1.275	2.268	217.7	6.32	36.5	50.0	1.525	146.4	50.0	0.058
550	132.0	1.260	2.201	290.5	5.99	39.5	50.0	1.507	198.9	50.0	0.065
700	168.0	1.248	2.139	359.4	5.73	42.0	54.0	1.492	250.7	50.0	0.072
850	204.0	1.236	2.084	425.1	5.50	44.0	68.5	1.478	301.5	50.0	0.078
1000	240.0	1.225	2.039	489.4	5.32	46.0	82.0	1.466	351.8	52.0	0.084
1150	276	1.215	2.001	552	5.15	47.5	95.5	1.454	401	63.0	0.090
1300	312	1.205	1.967	614	4.99	49.0	110.0	1.444	451	73.5	0.096
1450	348	1.198	1.939	675	4.85	50.5	122.5	1.434	499	84.0	0.102
1600	384	1.190	1.914	735	4.73	52.0	135.0	1.424	547	95.0	0.108
1750	420	1.182	1.891	794	4.60	53.0	147.0	1.416	595	105.0	0.113
1900	456	1.177	1.872	854	4.49	54.5	160.0	1.408	642	115.0	0.118
2050	492	1.169	1.855	913	4.41	55.5	172.0	1.401	689	125.0	0.124
2200	528	1.162	1.837	970	4.34	56.5	185.0	1.394	736	135.0	0.130
2350	564	1.158	1.821	1027	4.29	58.0	196.0	1.388	783	145.0	0.136
2500	600	1.152	1.805	1083	4.24	59.0	207.5	1.382	829	155.0	0.141
2650	636	1.148	1.789	1138	4.19	60.0	219.0	1.376	875	165.0	0.146
2800	672	1.142	1.774	1192	4.15	61.0	231.0	1.369	920	173.5	0.152
2950	708	1.139	1.759	1245	4.11	62.0	242.0	1.364	966	183.5	0.157
3100	744	1.136	1.744	1298	4.07	63.0	253.0	1.359	1011	192.5	0.162
3250	780	1.132	1.730	1349	4.03	64.0	264.0	1.354	1056	202.5	0.167
3400	816	1.129	1.717	1401	4.00	64.5	275.0	1.349	1101	211.5	0.172
3550	852	1.128	1.704	1452	4.00	65.5	286.0	1.344	1145	221.0	0.178
3700	888	1.122	1.691	1502	4.00	66.5	296.0	1.339	1189	230.0	0.183
3850	924	1.121	1.678	1550	4.00	67.5	306.0	1.335	1234	239.5	0.188
4000	960	1.120	1.666	1599	4.00	68.0	315.0	1.331	1278	249.0	0.192
4200	1008	1.120	1.650	1663	4.00	69.0	326.0	1.325	1336	261.5	0.200
4400	1056	1.120	1.634	1726	4.00	70.0	337.0	1.320	1394	274.0	0.206
4600	1104	1.120	1.620	1788	4.00	70.5	349.0	1.314	1451	285.5	....
4800	1152	1.120	1.605	1849	4.00	71.5	358.0	1.310	1509	297.0	....
5000	1200	1.120	1.590	1908	4.00	72.5	367.0	1.305	1566	308.0	....
5200	1248	1.120	1.577	1968	4.00	73.0	376.0	1.301	1624	318.5	....
5400	1296	1.120	1.564	2027	4.00	74.0	385.0	1.297	1681	329.5	....
5600	1344	1.120	1.552	2086	4.00	74.5	393.0	1.294	1739	339.0	....
5800	1392	1.120	1.539	2142	4.00	75.5	402.0	1.291	1797	350.0	....
6000	1440	1.120	1.526	2197	4.00	76.0	409.0	1.290	1858	359.0	....
6200	1488	1.120	1.514	2253	4.00	77.0	417	1.289	1918	369	....
6400	1536	1.120	1.502	2307	4.00	77.5	425	1.288	1978	377	....
6600	1584	1.120	1.491	2362	4.00	78.0	431	1.288	2040	386	....
6800	1632	1.120	1.480	2415	4.00	79.0	438	1.288	2102	395	....
7000	1680	1.120	1.470	2470	4.00	79.5	446	1.288	2164	405	....
7500	1800	1.120	1.447	2605	4.00	81.5	464	1.288	2318	426	....
8000	1920	1.120	1.426	2738	4.00	83.0	481	1.288	2473	446	....
8500	2040	1.120	1.409	2874	4.00	85.0	497	1.288	2628	467	....
9000	2160	1.120	1.400	3024	4.00	87.0	515	1.288	2782	486	....
9500	2280	1.120	1.400	3192	4.00	89.0	535	1.288	2937	504	....
10000	2400	1.120	1.400	3360	4.00	91.5	555	1.288	3091	522	....
11000	2640	1.120	1.400	3696	4.00	95.0	595	1.288	3400	560	....
12000	2880	1.120	1.400	4032	4.00	99.0	634	1.288	3709	596	....
13000	3120	1.120	1.400	4368	4.00	103.0	673	1.288	4019	633	....
14000	3360	1.120	1.400	4704	4.00	106.5	712	1.288	4328	668	....
15000	3600	1.120	1.400	5040	4.00	109.5	751	1.288	4637	704	....
16000	3840	1.120	1.400	5376	4.00	112.5	789	1.288	4946	740	....
17000	4080	1.120	1.400	5712	4.00	115.5	827	1.288	5255	776	....
18000	4320	1.120	1.400	6048	4.00	118.0	863	1.288	5564	810	....
19000	4560	1.120	1.400	6384	4.00	120.0	899	1.288	5873	844	....
20000	4800	1.120	1.400	6720	4.00	120.0	900	1.288	6182	877	....

[a] Includes pickup allowance and correction for difference between test and operating conditions.
[b] To be specified in catalog.

2. *Liquid and Gaseous Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel to the calorific value of 1 lb or cubic foot of fuel respectively.

The following efficiencies apply to current designs of boilers operated under favorable conditions at their gross output ratings. Some older boilers designed primarily for hand firing may have lower efficiencies when automatically fired.

Anthracite, hand fired	60 to 75 per cent
Bituminous Coal, hand fired	50 to 65 per cent
Stoker fired	60 to 75 per cent
Oil and Gas fired	70 to 80 per cent

Higher efficiencies for hand fired bituminous coal may be obtained by careful firing of either a regular or a smokeless boiler.

RATING OF BOILERS

In referring to boiler rating it is necessary to know the basis on which the rating has been established in order to understand the exact meaning of the term. The following example will illustrate the meaning of three ratings which might be established for the same boiler.

Assume that an installation has the following loads determined in accordance with the section *Selection of Boilers:*

Net Load	1000 sq ft of steam radiation
Piping Tax	200 sq ft of steam radiation
Design Load	1200 sq ft of steam radiation
Pickup Allowance	240 sq ft of steam radiation
Maximum or Gross Load	1440 sq ft of steam radiation

A boiler that is just large enough to carry this system might be said to have a *net load rating* of 1000 sq ft, a *design load* rating of 1200 sq ft, or a *gross load* rating of 1440 sq ft, depending on the basis on which the boiler is rated.

On a *net load* basis the boiler would be rated 1000 sq ft of steam radiation and would have sufficient excess capacity to supply the *normal piping* and *pickup load.* Net $I = B = R$ Ratings, *SBI* Net Ratings, and Net Load Ratings of the *Heating, Piping and Air Conditioning Contractors National Association* are established on this basis.

On a *design load* basis the boiler would be rated 1200 sq ft of steam radiation and would have sufficient excess capacity to supply the *pickup load.* It would be of adequate size for a system in which the *sum* of the *net load* and the *piping heat loss* did not exceed 1200 sq ft of steam radiation. The *SBI* Ratings shown in columns 1, 2, 3, 10, 11 and 12 of Table 2 (not to be confused with *SBI* Net Rating) are established on a *design load* basis.

On a *gross output* basis of rating the boiler would be rated 1440 sq ft of steam radiation and would be of adequate size for a system in which the sum of the *net load, piping load,* and *pickup load* did not exceed 1440 sq ft of steam radiation. Gross $I = B = R$ Output and *A.G.A.* Ratings are established on a *gross output* basis.

In the determination of boiler ratings the *Gross Output* is the quantity of heat available at the boiler nozzle with the boiler normally insulated and when operating under limitations stipulated in the code or method by which the boiler is rated. The boiler may be capable of producing a greater nozzle output but in doing so would exceed some of these limitations.

SELECTION OF BOILERS

General Factors

The *Maximum Load* or *Gross Load* on the boiler is the sum of the four following items.

The *Design Load* is the sum of items 1, 2, and 3.

The *Net Load* is the sum of items 1 and 2.

1. *Radiation Load.* The estimated heat emission in Btu per hour of the connected radiation (direct, indirect, or forced convection coils) to be installed.

The connected radiation is determined by calculating the heat losses for each room in accordance with data given in Chapter 6, 8, and 14. The sum of the calculated heat losses for all the rooms represents the total required heat emission of the connected radiation expressed in Btu per hour. As practically all boilers are now rated on a Btu basis, it is unnecessary to convert the radiation load to equivalent square feet of equivalent radiation.

2. *Hot Water Supply Load.* The estimated maximum heat in Btu per hour required to heat water for domestic use.

When the hot water supply is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. A common practice is to add 240 Btu per hour to the radiation load for each gallon of storage tank capacity. For more specific information see Chapter 50.

3. *Piping Tax.* The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

As the heating industry as a whole is not entirely agreed upon piping tax allowances for different sizes of installations it is better to compute the heat emission from both bare and covered pipe surface in accordance with data in Chapter 28. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 per cent of the Net Load. In determining Net $I=B=R$ Ratings from Gross $I=B=R$ Output, the piping factor allowed varies from 30 per cent for small boilers to 12 per cent for larger boilers.

4. *Warming-Up or Pick-Up Allowance.* The estimated increase in the normal load in Btu per hour caused by the heating up of the cold system.

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature, and heating up cold radiation and piping. The factors to be used for determining the allowance to be made should be selected from Table 4.

TABLE 4. WARMING-UP ALLOWANCES FOR HAND-FIRED LOW-PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3)		PERCENTAGE CAPACITY TO ADD FOR WARMING-UP[c]
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a] This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

Vo
ating S
Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c] This table refers to *hand-fired, solid fuel boilers*. A factor of 20 per cent over design load is adequate when *automatically-fired fuels* are used.

[d] 240 Btu per square foot.

Other items to be considered in boiler selection are:

a. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.
b. Crate area with hand fired coal, or fuel burning rate with stokers, oil, or gas.
c. Combustion space in the furnace.
d. Type of heat liberation, whether continuous or intermittent, or a combination of both.
e. Convenience in firing and cleaning.
f. Adaptability to changes in fuel and kind of attention.
g. Height of water line.
h. Miscellaneous items such as draft available, possibility of future extension, possibility of break-down, and head room in the boiler room.
i. The most economical size of boiler is usually one that is just the right size for the load. Either larger or smaller boilers may be less economical.

Cast-Iron Boilers

Net load ratings of cast-iron boilers are usually available from manufacturers' catalogs. They may also be obtained conveniently from published tables of $I = B = R$ ratings[6], or from recommendations of the *Heating, Piping and Air Conditioning Contractors National Association*[7] and can be used in selection of boilers, unless the heating system contains an unusual amount of bare pipe, or the nature of the connected load is such that the normal allowances for pipe loss and pickup do not apply. In such a case, the selection must be based on the gross output.

Steel Heating Boilers

SBI catalog ratings in accordance with the previously mentioned *Steel Boiler Institute, Inc.* code are intended to correspond with the estimated

Table 5. Practical Combustion Rates for Coal-Fired Heating Boilers Operating at Maximum Load on Natural Draft of from ¼ in. to ½ in. Water[a]

Kind of Coal	Sq Ft Grate	Lb of Coal per Sq Ft Grate per Hour
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a] Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

design load. When the heat emission of the piping is not known, the net load to be considered for the boiler may be determined from Tables 1 and 2. The difference between design load and net load represents an amount which is considered normal for piping loss of the ordinary heating system.

Boilers with less than 177 sq ft of heating surface and having *SBI* net ratings (steam) of not more than 3,000 sq ft if mechanically fired and 2,480 sq ft if hand fired, are classified as residence size. An insulated residence boiler for oil, gas, or stoker firing may carry a *net load* expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler has been tested in accordance with the *SBI* Code for Testing Oil-Fired Steel Boilers at output rates of 125, 150, and 175 per cent of the *SBI* Net Rating. The *SBI* Net Rating (square feet steam) for hand-fired residence boilers is not greater than 14 times the heating surface. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the *net load.*

Heating Surface and Grate Area Basis

Where neither the *net load* nor *gross output* ratings based upon performance tests are available, a good general rule for conventionally designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load. This is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the *maximum load* including the warming-up allowance will be provided for by operating the boiler in excess of the *design load*, that is, in excess of the 100 per cent rating on a boiler-horsepower basis. *SBI* ratings for hand firing are based on 10 sq ft of heating surface per boiler horsepower.

Due to the wide variation which may be encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.

H = required *gross output* of the boiler, Btu per hour (see Selection of Boilers).

C = desirable combustion rate for fuel selected, pounds of dry coal per square foot of grate per hour (see Table 5).

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required *gross output* of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Equation 1. With small boilers, where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per

cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Gas-Fired Boilers

After determining the *net load* for the installation, gas designed boilers can usually be selected from manufacturers' tables of *net load ratings* which are based on piping and pickup allowances varying from 56 per cent for boilers of 200 sq ft and less to 35 per cent for boilers of 4000 sq ft and larger. If the piping and pickup load or other factors create an unusual load, a boiler should be selected which has an *A.G.A. output rating* equal to the maximum output required. Detailed recommendations for selection of gas designed boilers are given in the A.G.A. publication, Comfort Heating[8].

SPACE LIMITATIONS

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flue tubes, and should be at least 3 ft greater than the length of the tubes.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts, and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

CONNECTIONS AND FITTINGS

Steam outlet connections should be the full size of the manufacturers' tappings in order to keep the velocity of flow through the outlet reasonably low and to avoid fluctuation of the water line and undue entrainment of moisture, and should extend vertically to the maximum height available above the boiler. A steam velocity in boiler outlets not exceeding 25 to 30 fps at maximum load is recommended unless data are available to show that a higher velocity is satisfactory.

Particular attention should be given to *fitting connections* to secure conformity with the *ASME* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections, and safety valve capacity.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent cleanout plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water

boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapter 23 and 24 and the *ASME* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer should always be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

Five precautions that should be taken in all installations to prevent damage to the boiler are:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.
2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.
3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.
4. Condensation in steam systems must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.
5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for mechanically fired boilers.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; (*h*) insufficient radiation installed; and (*i*) with mechanical firing, fuel burning equipment too small.
2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive rate of output.
3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return

piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities tend to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least $1\frac{1}{4}$ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure, and while fire is burning briskly open valve in blow-off line. When pressure recedes, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use, as given by the boiler manufacturer, should be carefully followed.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulfur in the

soot with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard that some one may inadvertently build a fire in a dry boiler, however, it is safer to keep the boiler filled with water, particularly in residential installations. Air can be excluded from a steam boiler by raising the water level into the steam outlets. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings, and regulator parts.

WARM AIR FURNACES

Warm air heating furnaces of a number of types and a wide range of sizes are listed and illustrated in the *Catalog Data Section*.

Warm air furnaces may be classified in several different ways:

1. According to method of heat distribution—these are either gravity or mechanical (blower) furnaces.

2. According to fuels for which the furnaces are designed—these are coal hand-fired or stoker-fired, oil, gas, or wood.

3. According to materials of construction—they are cast-iron, low carbon steel, and occasionally high temperature steel alloys.

4. According to design or construction, such as drum and radiator, tubular, horizontal, etc.

Gravity Warm Air Furnaces

A gravity furnace is one in which the motive head producing air flow depends upon the difference in density between the heated air leaving the top of the casing and cooled air entering the bottom of the casing. Since this gravity head is relatively low, the furnace must have low internal resistance to the flow of air and relatively large areas must be available for free circulation within the furnace casing. It is common practice to provide approximately 50 per cent free air area through gravity type furnaces.

Furnaces for gravity type systems are available in designs suitable for central heating, pipeless furnace, or unit floor furnace installations. Booster fans are sometimes used in conjunction with gravity design systems, to increase air circulation. Where a fan is to be used with a furnace casing sized for gravity air flow, some form of baffling must be employed to restrict the free area within the casing and to force impingement of the air against the heating surfaces. Where square casings are used, the corners must be baffled.

Mechanical Warm Air Furnaces

Mechanical or forced warm air furnaces include fans or blowers as integral parts for the purpose of circulating the air and usually include air filters.

Centrifugal fans with either backward or forward curved blades are the type most commonly used. Motors may be mounted on the fan shaft or connected to the fan by a belt drive. Adjustable pulleys are desirable to provide means of regulating the quantity of air distributed to the heated spaces. Either the motor load or the noise considerations may limit the maximum operating fan speed. Two-speed motors have given successful operating results and are recommended. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with the fan housing.

Filters

Several types of filters are available for mechanical warm air furnace applications and are discussed in Chapter 33. For maximum efficiency and life under operating conditions, filters should not be subjected to a temperature in excess of 150 F. Filters should have at least 80 per cent average efficiency on an 8-hr test at a maximum resistance of 0.25 in. of water. Filter resistance rises rapidly with the accumulation of dirt, and may reduce the air circulation over heating surfaces. In domestic furnaces, the maximum velocity, based on nominal filter area, should not exceed 300 fpm.

Fuel Utilization

A combustion rate of from 5 to 8 lb of coal per (square foot of grate) (hour) is recommended for residential furnaces. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes the ratio may be as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of a number of furnaces, using one or more fans.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. Furnaces for burning oil fuel are usually designed for blow-through installations so that the pressure in the air space is higher than that in the combustion chamber or flues. The *National Warm Air Heating and Air Conditioning Association* has prepared a Tentative Code for Testing and Rating of Oil-Fired Furnaces. Compact fan-furnace-burner units are available, suitable for basement, closet, or attic installations.

Gas-fired forced air furnaces should conform in construction and performance to *A.G.A.* Approval Requirements.

Heavy Duty Fan Furnaces

Fan furnaces for large commercial and industrial buildings, churches, schools, etc., are available in sizes ranging from 300,000 to 6,000,000 Btu per (hour) (unit). Heavy duty furnace heaters may be arranged in battery combinations of one or more units.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu

per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area of furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 43. Ducts are designed by the method outlined in Chapter 41.

MATERIALS AND CONSTRUCTION

Cast-Iron Furnaces

Cast-iron furnaces are made in a multiplicity of designs or shapes. For solid fuels they are generally of round sectional construction, the sections being cemented or bolted together. Various types of radiators for secondary convection heat transfer are employed. Such radiators are of the circular, doughnut type, or tubular type.

Cast-iron is frequently used in the construction of gas or oil-fired furnaces, designs varying considerably with two general types in common use: multi-sectional type and those with single combustion chambers having auxiliary secondary surface.

Cast-iron furnaces are made in capacities ranging from those for small insulated residence application with inputs of 40,000 Btu per hour or less, to capacities as large as 600,000 Btu per hour.

Cast-iron furnaces are usually constructed with a minimum sectional thickness of $\frac{1}{4}$ in. and effectively resist high temperatures and corrosion. They usually have a fairly large heat capacity because of their mass, which provides a distinct *fly wheel* or carry-over heating effect.

Steel Furnaces

Formed sheet steel construction is frequently used in furnace design. Welding, riveting, or both are used to join the formed metal. The use of steel castings, however, is rare, because of the cost, and because high stresses are not encountered in normal furnace construction. Types of design employed vary greatly, although perhaps the most common type consists of a drum and circumferential or rear radiator. Steel gas furnaces may also be sectional in design or may be combinations of common combustion chambers and sectional or tubular radiation surfaces connected to a flue gas collector.

Steel furnaces are made in capacities ranging from 40,000 Btu per hour to capacities as large as 600,000 Btu. Steel furnaces have low heat capacities as a result of their relatively low mass and, therefore, deliver heat rapidly on demand.

FURNACE RATING

Warm air furnaces are generally rated in Btu per hour output at the bonnet (point of heat generation) or at the register (point of heat delivery).

Rating Equations for Gravity Warm Air Furnaces[9]

Until a method of testing and rating gravity warm air furnaces has been developed, the following empirical rating equations are recommended by the *National Warm Air Heating and Air Conditioning Association.*

Gravity warm-air furnaces of conventional design, having ratios (of heating surface to grate area) of 15 to 1 or greater, and having a ratio of casing area to face area not less than 0.4, are rated by the following equations:

a. *Hand-fired furnaces Converted to Stoker, Gas, or Oil Firing.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (2)$$

b. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (3)$$

c. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times 25 \times G \times 1.333 \qquad (4)$$

where

S = heating surface, in square feet.

G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.75 x (*Bonnet Capacity*). The *Leader Pipe Rating* in square inches, formerly used as a rating unit, may be found by dividing the *Register Delivery Rating* by 136.

Heating Surface of Furnace

Prime heating surface is defined[9] as surface above the top of the grate having *hot gases or live fuel on one side and circulating air over the other*, and in all cases is measured on the exterior or air side. The areas of the outer casing, the inner liner, and any radiation shields shall not be considered as heating surface.

In determining the amount of heating surface, extended surfaces are considered to be prime heating surface subject to the following limitations:

1. Extended heating surface may consist of fins, ribs, webs, lugs, or other projections from the prime heating surface. Projections less than ½ in. thick at the base and extending more than 1 in. from the prime surface are classified as fins.

2. Integral fins are continuously welded to, or cast as a part of, the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from Prime Surface	1st inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area................	0.40	0.30	0.20	None

3. Non-integral fins are spot welded to, or otherwise held in line contact with the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from Prime Surface.	1st inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area................	0.30	0.20	0.15	None

4. In the case of ribs, webs, or lugs more than ½ in. thick at the base and extending less than 1 in. from the prime surface, the entire surface in contact with circulating air is included as heating surface.

5. In the case of ribs, webs, or lugs more than ½ in. thick at the base and extending more than 1 in. from the prime heating surface the areas of both sides of the first inch are included as prime heating surface. The portions projecting beyond 1 in. are treated as integral fins.

Grate Area

Grate area is defined[9] and treated for purpose of rating as follows:

1. The *nominal grate area* is defined as the total cross-sectional area of the bottom of the firepot. In steel furnaces the nominal grate area is the cross-sectional area inside the firebrick lining.

2. The *actual grate area*, used for calculating the ratios of heating surface to grate area, is the nominal grate area minus certain areas that cannot be considered as part of the grate itself. The following rules govern these deductions: (1) If a solid, continuous ledge extends around the grate and inside the firepot, any area of this ledge extending inside of a circle, the diameter of which is 1 in. less than the diameter of the bottom of the firepot, shall be deducted. (2) If separate, solid projections extend from the firepot towards the grate, the areas of any portions of these projections extending inside of a circle, the diameter of which is 3 in. less than the diameter of the bottom of the firepot, shall be deducted. (3) In the case of grates which are inclined, or are conical, the projected area is the same as the nominal grate area. The latter should, therefore, be used after making any necessary deductions.

Ratings for Forced Air Furnaces

For solid fuel burning, forced air furnaces having bonnet capacities between 80,000 and 250,000 Btu per hour, no standard method of test has been accepted, although eventually such codes will be developed. The *National Warm Air Heating and Air Conditioning Association* recommends empirical equations similar to Equations 2, 3, and 4 for gravity furnaces, except that a constant of 2265 is used in place of the 1785.

The following testing and rating codes have been generally accepted in the industry:

Commercial Standards CS-109-44 for rating solid fuel-burning, forced-air furnaces having bonnet outputs of 80,000 Btu per hour or less. This provides a method of rating small coal-fired forced-air furnaces by test.

A Tentative Code for Testing Oil-Fired Furnaces. This code has been adopted by the *National Warm Air Heating and Air Conditioning Association* for rating oil-fired furnaces by test.

The American Gas Association method of rating gas-fired furnaces based upon performance under tests. This is described in the Approval Requirements for Central Heating Gas Appliances.

Commercial Standards 113-44 is a method of rating oil-burning floor furnaces by test.

Commercial Standard CS 104-43 is a method of rating warm air furnaces equipped with pot-type oil burners by test.

Various codes covering the construction and performance of appliances as related to fire hazards have been developed by Underwriter Laboratories, Inc. In addition, there are many municipal codes[10] which regulate construction and installation of furnace equipment.

The yardstick of the *National Warm Air Heating and Air Conditioning Association* provides criteria for evaluating a furnace design and installation against industry accepted standards.

FURNACE EFFICIENCY

Rating formulas of the *National Warm Air Heating and Air Conditioning Association* are based on 55 per cent efficiency for gravity coal furnaces and 65 per cent efficiency for forced air coal furnaces. In the tentative Oil Testing Code the contemplated minimum efficiency is 70 per cent for

oil fired forced air furnaces. Gravity gas furnaces approved by the *American Gas Association* are assigned a rating based on 75 per cent efficiency. All *forced air gas-fired furnaces* approved by *American Gas Association* are assigned a rating based on 80 per cent efficiency.

DESIGN CONSIDERATIONS

Considerations of prime importance in the design of warm air furnaces and some general suggestions to be observed in connection with each are as follows:

1. *Adequate heat transfer surface.*
   a. Heat transfer rates of 2,000 to 4,500 Btu per (hour) (square foot) of heating surface may be obtained without unduly high metal temperatures.
   b. Fins, pins and bosses are frequently used to add surface and to break down superficial gas films, both on gas-to-metal and metal-to-air surfaces.
   c. Surface and stack (flue gas) temperatures are good indications of the amount and effectiveness of the heating surfaces.
2. *Safe and efficient combustion of fuel.*
   a. Proper mixture of fuel and air is necessary for efficient combustion. This necessitates careful attention to the design of grates, nozzles, burners, air inlet areas and location, and combustion chamber baffling.
   b. Regulation of the quantity and the distribution of the air for combustion should be provided by use of check dampers, draft regulators, draft hoods, air shutters and air orifices.
   c. Total draft loss through appliances should not exceed that available from chimneys which would normally be obtainable in the size of building which the appliance will supply with heat.
   d. The use of ignition safety devices such as safety pilots, hold-fire controls, and the like is recommended.
3. *Fuel Capacity of Appliance.*
   a. With solid fuels adequate coal capacity should be provided for at least 5 hr of operation at the maximum rated combustion rate.
4. *Adequate circulation of air over heating surface.*
   a. In gravity furnaces, free air space between casing and heat exchanger should be great enough to permit free flow over all surfaces.
   b. Forced air furnace design must include fans having proper capacity and suitable performance characteristics. Internal static pressures must be minimized without losing the advantages of high velocity circulation over the heat exchanger surfaces.
   c. The air flow over the heating surface must be directed to obtain maximum efficiency and to eliminate hot spots and air noises.
   d. Air velocities at bonnet should not be much in excess of 1,000 fpm and air temperature distribution at the furnace outlet should be uniform within approximately $\pm$ 30 deg.
5. *Durability.*
   a. A minimum metal weight for gas-fired heat exchangers is established as No. 20 U.S. Gage for plain carbon steel by the *A.G.A.* Approval Requirements for Central Heating Gas Appliances with some municipal codes specifying 18 gage. Cast-iron sectional thicknesses of ¼ in. to ⅜ in. are recommended.
   b. Added strength and reinforced designs may be required to preclude damage in shipment, burning out from overfiring, or corrosion from condensation.
   c. Maximum heat exchanger surface temperatures which may be used vary with the metal. The *American Gas Association* Approval Requirements for Central Heating Gas Appliances specify a maximum of 875 F for cast-iron or steel gas furnaces, and the *National Bureau of Standards* CS 109-44 Code for Forced Air Solid Fuel-Burning Furnaces specifies 1000 F as a maximum surface temperature. These temperatures define the range in which oxidation of non-alloy ferrous metal begins. The use of proper alloy additions increases the temperature resistance properties of metals.
   d. Casing temperatures should be controlled so that they do not become hazards to burn those who touch them, or to create fires.

6. *Serviceability.*
   a. Those parts of the furnace which may be subject to soot, fly-ash, or condensation deposits should be accessible for cleaning.
   b. Parts which may require adjustments or replacements, such as grates, baffles, liners, controls, should be removable.
   c. Furnaces should be so designed that they can be installed with a minimum of difficulty.
7. *Control.*
   a. Thermostatic controls of various types should be used to correlate space temperatures with unit operation.
   b. Controls should be provided wherever possible, to prevent the occurrence of excessive temperatures or other conditions in any part of the unit which might cause unsafe operation.
8. *General design considerations.*
   a. Furnace casings are normally constructed of formed and painted sheet steel or of galvanized iron. The casing should be protection from excessive radiation losses and temperatures by use of insulation or sheet steel air space liners. Liners should extend from the grate level to the top of the furnace and should be spaced from 1 in. to 1½ in. from the outer casing.
   b. The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet and thus provides a larger plenum chamber.
   c. Warm air outlet and return air connections should be designed so that the ductwork may be easily attached. A ¾ in. flange is normally used for this purpose.
   d. Suitable provision shall be made in appliances so that the controls and humidifiers may be installed in the proper location. When these auxiliary units are installed in the ductwork, detailed instructions should be provided to insure their proper location.
   e. The flue connection should be of integral flue pipe size and provision should be made to attach the flue pipe to the flue outlet of the furnace.

HUMIDIFICATION EQUIPMENT

Water evaporating pans are usually located in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. There is a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. All humidifiers require provision for removal of dirt and lime.

SPACE HEATERS

Space heaters may be classified in several ways, such as:

1. By the type of fuel used as coal, wood, gas, and fuel oil.

2. According to the method of heat distribution as circulators or radiant types. A radiant heater is one in which the heat exchanger surface is exposed directly to the room atmosphere, and the generated heat is dissipated primarily by radiation. A circulating heater is essentially a jacketed radiant heater from which circulation of room air is promoted by the chimney effect caused by the movement of air passing upward between the jacket and heat exchanger surface.

3. According to method of design for particular fuel types, such as: (a) surface-fired and magazine-feed for solid fuels, (b) vaporizing pot type and blue flame heater

for oil, and (c) vented and unvented heaters for gas. (The type of gas burner design, such as, injection, yellow flame, power, and pressure, may also be mentioned.)

SOLID FUEL-FIRED HEATERS

Surface fired heaters normally have a front firing door and are operated with relatively shallow fuel beds. A magazine feed heater includes a deep reservoir of fuel to lenghten the attention intervals. In a true magazine feed heater the rate of fuel ignition would be equal to the rate of burning; and self-feeding should operate to move the unburned fuel by gravity flow from the magazine section into the hearth area. However, the ideal balance between rate of ignition and rate of burning is virtually impossible to attain for any solid fuel under normal usage, although self-feeding may be obtained with wood and some free burning coals. Thus, a magazine-type space heater is essentially a deep surface-fired heater, its principal difference being increased fuel capacity.

Materials and Construction

There is no accepted code governing the construction of solid fuel burning space heaters. In past years cast iron was used predominantly in the construction of coal and wood heaters, with the exception of the so called *air tight* heaters designed as low cost wood burning units with little consideration for long life, and stoves were priced on a poundage basis. The present trend is toward fabricated steel parts and welded assemblies, although cast iron is still used for grates, firebox liners, and parts subject to high temperatures. Refractory firebox liners are also used quite extensively.

Formed sheet steel is used predominantly for the outer jacket of circulator heaters, although heaters with outer casings formed from cast iron are still readily available. Circulator cabinets normally have surfaces finished with a porcelain enamel, while the casing of a radiant heater is finished with an air dried japan or a baked enamel.

Both welding and stove bolts are used in unit assembly, and stove cement is used on section joints to prevent air leakage. This latter is extremely important to obtain a low rate of combustion when desired.

Testing and Rating

There is no accepted code governing the method of testing and rating solid fuel space heaters. A tentative procedure, TS-3443, has been issued by the *Division of Trade Standards, National Bureau of Standards* but is based on use of anthracite as a rating fuel although bituminous coals are used predominantly as heating fuels. This procedure which has been the basis of published ratings consists of determining the heater output, expressed in Btu per hour, by the indirect method in which the measurable heat losses: (1) loss due to moisture in the fuel, (2) loss due to heat in the dry flue gases, (3) loss due to unburned carbon monoxide, (4) loss due to unburned combustible in the ash and refuse, and (5) unaccounted for losses, are measured by test and subtracted from the heat input. The difference multiplied by the burning rate in pounds per hour is the heater output.

When using anthracite as the rating fuel, the unaccounted for loss has been assumed to be zero. It has been accepted practice to use a 20 per cent allowance for the unaccounted for losses when burning a bituminous coal. The value of this factor is under study at the present time but no published data are available.

No allowance is made for a radiation loss, as this is useful heat.

Design Considerations

Some important considerations in the design of solid fuel space heaters are:

1. Suitable protection by baffling or insulation against overheating of floors and walls.
2. Tight heater construction to prevent air leakage and to enable maintenance of a suitably low minimum burning rate. This includes a ground, paper tight joint between the ashpit door and door frame.
3. Sufficient free air space between the casing and heat exchanger of circulating heaters to permit free air flow over all surfaces and maintain a suitably low casing temperature.
4. Protection of all metal parts from deterioration due to high temperature. This may be accomplished by: (a) fabrication of certain parts from cast iron or an alloy iron, (b) protection by a refractory liner, (c) use of high temperature enamel coatings, (d) directing air against hot spots.
5. Proper admission of secondary air to complete the combustion process. Care should be taken to prevent this air from also functioning as primary air.
6. Strength in assembly to prevent transportation and use damage.

Considerable work has been directed in the past few years toward improvement of the performance of bituminous coal fired space heaters, with particular reference to smokeless operation under conditions of normal operation such as obtained in homes.

OIL HEATERS

Vaporizing pot type oil heaters consist of: (1) a metal pot in the bottom of which the oil is vaporized, the vapors burning at or near the top of the pot, (2) a secondary combusion chamber, or heat exchanger, in which combustion is completed. The flue connection is made to this chamber or through a second heat exhanger which may be of the diving flue type installed to increase efficiency. The burner may be designed for operation both with and without mechanical draft.

Blue flame oil burners differ from the pot type variety in that removable perforated sleeves are provided above an oil pan instead of a metal pot, and lighting rings or kindlers are used for easy lighting.

Both types of oil burners operate by the burning of the oil vapor rather than the oil itself, the oil being first fed to a chamber in which the oil is entirely vaporized, then mixed with air introduced through suitably located ports and burning at the top of the pot or perforated sleeves. Such heaters are designed to burn No. 1 oil (See Table 4, Chapter 16) or kerosene (coal oil). At no time should oil heavier than that for which the burner is designed be used, as heavier oils may cause excessive carbonization in the burner or fuel feed line.

Materials and Construction

Formed sheet steel and welded assemblies are used primarily in oil heater construction. Standards governing construction which have been generally accepted are:

1. Commercial Standard for Flue-Connected Oil Burning Space Heaters equipped with Vaporizing Pot Type Burners, CS101-43, (*National Bureau of Standards*).
2. Standard for Oil-Burning Stoves, Subject 896, (*Underwriters' Laboratories, Inc.*).
3. Standard for Construction and Performance of Oil Burners for Installation in Stoves and Ranges, Subject 865, (*Underwriters' Laboratories, Inc.*).

Some States or municipalities have codes which apply locally, but these

usually apply primarily to installation and the Underwriters' Laboratories label of approval is sufficient to cover acceptance of the unit.

Testing and Rating

Commercial Standard CS101-43 is intended to provide a uniform standard method for ascertaining the maximum practical heat output in Btu per hour of flue-connected oil-burning space heaters under approximately normal service conditions. This method is based upon the following equations:

$$H_r = A - B \tag{5}$$

and

$$E = H_r/A \tag{6}$$

where

A = total heat of fuel used

B = heat lost in flue gases

H_r = net heat delivered to the room

E = unit efficiency

The following minimum performance requirements are stipulated:

1. Adequate provision for ease of lighting and insurance against loss of ignition prior to heating of burner.
2. Ease of operation of controls.
3. Proper operation of burner without excessive carbonization with grades of oil recommended by the manufacturer.
4. The heater shall be capable of passing the 6 per cent ICHAM smoke test.
5. The heater shall be capable of operating with an overall efficiency of not less than 70 per cent under conditions of test, or at a lower stack draft recommended by the manufacturer.

Design Considerations

Some factors important in the design of oil-burning heaters are:

(1) Proper pitch of oil lines from the sump to the burner, thus preventing vapor and air lock.

(2) Proper positioning of the oil sump to maintain the proper oil level in the burners if factory assembled.

(3) Tight construction, not only of oil lines, but of oil tank, sump, and burner to prevent a hazardous condition due to oil leakage.

(4) Provision for leveling and aligning the entire stove for maintenance of proper operation.

(5) Provision of a draft regulator to prevent abnormal draft fluctuations.

(6) Proper shielding of an attached fuel oil supply tank to prevent excessive oil temperatures.

(7) All metal parts subjected to the corrosive action of the oil shall be made of non-corrodible metal or of metal suitably coated to resist corrosion.

(8) The heater should have suitable baffling or insulation to prevent overheating of floors and walls.

(9) Strength in assembly to prevent transportation and use damage.

GAS HEATERS

Vented gas heaters are defined as those capable of removing 90 per cent of the flue gases through a single flue outlet. All heaters having a gas input rating in excess of 50,000 Btu per hour must be of the vented type in order to meet ASA Approval Requirements for Gas Space Heaters[11].

Space heaters may be classified by burner type as follows:

1. *Injection Burner* type which employs the energy of a jet of gas to inject air for combustion into the burner and mix it with the gas.

2. *Yellow Flame Burner* type in which secondary air only is depended on for the combustion of the gas.

3. *Power Burner* type in which either gas or air or both are supplied at pressures exceeding, for gas, the line pressure; and for air, atmospheric pressure, this added pressure being applied at the burner.

4. *Pressure Burner* type in which an air-gas mixture is supplied under pressure.

Materials and Construction

Standards covering materials and accessories used in the construction of gas heaters are described in ASA Approval Requirements for Gas Space Heaters[11] and in applicable Listing Requirements[12].

Efficiency Requirement

Vented space heaters having input ratings in excess of 20,000 Btu per hour are required to have a heating efficiency of not less than 70 per cent[11], based on the total heating value of the gas. Vented space heaters having input ratings of 20,000 Btu per hour or less are required to have a heating efficiency of not less than 65 per cent[11]. These efficiencies are based upon the following equation:

$$e_t = 100 - \frac{H_t}{q} \times 100$$

where

q = hourly gas heat input, Btu per hour

H_t = heat above room temperature carried away by the flue products, Btu per hour.

e_t = heating efficiency, per cent

Radiant heaters are required to have a radiant efficiency of not less than 28 per cent.

Design Considerations

Some factors important in the design of gas heaters are:

1. Proper design of the burner head, port sizes and locations so that the flame will not lift, float, or flash back, and be excessively noisy in operation.

2. Proper venting of combustion chamber for relief of forces resulting from ignition of an explosive mixture of gas and air.

3. Protection of valve handles to prevent excessive temperature rise during operation.

4. Insulation and baffling of heater to prevent over-heating of walls and floor.

SPECIAL FEATURES

Some special design features found on all types of space heaters are:

(1) Circulating heaters can be equipped with small electric fans for increased room air circulation.

(2) Water evaporation pans are provided on some heaters, located in air which has been heated by contact with the heating surfaces.

(3) Thermostatic control of primary air admission, and oil or gas flow, dependent upon type of fuel used.

INSTALLATION

The two most important considerations involved in the installation of a space heater are safety and chimney draft.

A thorough examination should be made of the chimney prior to connecting the heater, as too often chimney faults, such as loose brick, broken tile, cracks, and open cleanouts, are the cause of poor performance rather than the heater itself. Some installation procedures for obtaining the most available chimney draft are:

1. Repair the chimney before installing the heater by replacing all broken brick or tile, sealing cracks and open cleanouts.
2. Place the heater as close as possible to the chimney outlet to avoid long horizontal runs of stovepipe.
3. Seal the joints between each length of stovepipe, and slope the horizontal sections approximately one inch per foot upward toward the chimney.
4. Do not have any section of stovepipe higher than the point of entry into the chimney.
5. If external factors such as trees or buildings overhanging the top of the chimney might create a downdraft condition, use a downdraft header of suitable design if necessary.
6. A barometric type check damper should be used where excessive or fluctuating drafts are encountered.

The following are some necessary installation safety precautions that should be observed:

1. If the stovepipe must pass through a combustible partition a suitable insulating thimble should be provided.
2. The heater should be placed at distances from combustible partitions as stipulated by the manufacturer, and a noncombustible floor board should be provided.
3. Safe provision for storage of fuel oil should be assured, and all oil lines should be tested for tightness.
4. All gas lines and connections should be tested for tightness.

REFERENCES

[1] See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, pp. 322 and 332.

[2] See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42.

[3] See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 23.

[4] See A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 366.

[5] *I=B=R* Testing and Rating Code for Low Pressure Heating Boilers (*Institute of Boiler and Radiator Manufacturers*).

[6] *I=B=R* Ratings for Cast-Iron Boilers (*Institute of Boiler and Radiator Manufacturers*).

[7] Engineering Standards, Part II, Net Square Feet Radiation Loads in 70 Deg Fahr., Recommended for Low Pressure Heating Boilers, 1943 (*Heating, Piping and Air Conditioning Contractors National Association*).

[8] Comfort Heating, 1938, pp. 35 to 39 (*American Gas Association*).

[9] Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems, Section No. 5, Second Edition, Jan. 1945, (*National Warm Air Heating and Air Conditioning Association*).

[10] Recommended forms for municipal installation and fire codes are included in Manual 7—Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems, Second Edition, 1947, (*National Warm Air Heating and Air Conditioning Association*).

[11] American Standard Approval Requirements for Gas Space Heaters, A.S.A. Z21.11, 1942 (*American Standards Association*).

[12] American Standard Listing Requirements for: Automatic Pilots, Z21.20, 1940, Gas Appliance Thermostats Z21.23, 1940, Domestic Gas Appliance Regulators, Z21.18, 1934, Automatic Main Gas-Control Valves, Z21.22, 1935, (*American Standards Association*).

CHAPTER 19

CHIMNEYS AND DRAFT CALCULATIONS

Theoretical and Available Draft, Determining Chimney Sizes, Factors Affecting Required Draft, Domestic Chimneys, Observed Test Performance, Draft Requirements of Domestic Appliances, Chimneys for Gas Heating, Construction Details, General Considerations

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A DRAFT, in the older sense, is a current of air and the draft of a furnace or boiler is that air current which flows through the fire-box and furnishes the oxygen for combustion. In engineering, however, the word *draft* has come to mean that pressure difference which causes this air current to flow and the word will be used in this sense in this chapter.

Draft is usually measured in inches of water and it is proper to speak of the draft in the fire-box or in the smoke breeching, etc., meaning the difference in pressure between the gases within and the air without those parts of a system. Draft is called positive when the pressure within such a part is less than that outside.

Draft is classified as natural and mechanical, depending on whether it is produced by a chimney or by a blower, and mechanical draft is further classified as induced or forced, depending on whether the air is drawn through or forced through the combustion chamber.

Chimneys can serve both to create a draft and to dispose of combustion products at a desirable height. For the latter purpose, chimneys, stacks, or, in the case of ships, funnels, are used in conjunction with mechanical-draft systems.

THEORETICAL DRAFT

If the air in one of two equal chimneys is heated while that in the other is not, the air in the heated chimney will be less heavy than that in the other chimney and a manometer or other pressure gage connecting the two at the bottom will indicate a pressure difference, called natural draft. The pressure of the air at the tops of the two chimneys will be equal, so that the pressure difference between them at the bottom will depend only on their height and the difference in density of the air they contain. The density of the air in either chimney is inversely proportional to its absolute temperature, so that the difference in pressure between them at the bottom will be proportional to their height and to the difference between the reciprocals of the absolute temperatures within them.

The pressure at the bottom of an unheated (and uncooled) chimney will be the same as that of the air outside, so that the unheated chimney can be dropped from the foregoing illustration. The manometer reading will be the same if its free connection is left open to the atmosphere.

These considerations in conjunction with those of barometric pressure and the difference in density of flue gases from that of air lead to the following formula:

$$D_t = 2.96\ HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) \tag{1}$$

where

H = height of chimney, feet.

B_o = existing barometric pressure, inches of murcury.

W_o = density of air at 0 F and 1 atmosphere pressure, pounds per cubic foot.

W_e = density of flue gas at 0 F and 1 atmosphere pressure, pounds per cubic foot.

T_o = temperature of air surrounding the chimney, Fahrenheit degrees absolute.

T_c = average or effective temperature of the gases in the chimney, Fahrenheit degrees absolute.

The quantity D_t, yielded by the formula, is the pressure difference between the gas inside and air outside of the chimney, in inches of water, when no flow occurs in the chimney. The quantity is variously known as the theoretical draft, the static draft or the computed draft. It is very useful in predicting and analyzing chimney performance, but it is seldom if ever attained in an actual chimney because of the friction incident to gas flow, wind effects, etc.

AVAILABLE DRAFT

The available draft, D_a, *for large chimneys and stacks* has been estimated with apparent satisfaction in the past by means of formulas which in effect deduct an estimated friction loss from a theoretical draft determined as in Equation 1. The friction loss can be estimated by means of one of the formulas available for ducts, such as the Fanning equation. This procedure results in formulas for the available draft as follows:

For a *cylindrical* stack:

$$D_a = 2.96\, HB_o\left(\frac{W_o}{T_o} - \frac{W_e}{T_c}\right) - \frac{0.00126 W^2 T_c f L}{D^5 B_o W_e} \qquad (2)$$

and for a *rectangular* stack:

$$D_a = 2.96\, HB_o\left(\frac{W_o}{T_o} - \frac{W_e}{T_c}\right) - \frac{0.000388 W^2 T_c f L(x + y)}{\overline{xy}^3 B_o W_e} \qquad (3)$$

where

D_a = available draft, inches water gage.

H = height of chimney above grate, feet.

B_o = existing barometric pressure, inches of mercury.

W_o = density of air at 0 F, 1 atmosphere pressure.

W_e = density of flue gas at 0 F, 1 atmosphere pressure.

T_o = temperature of atmosphere, Fahrenheit degrees absolute.

T_c = temperature of flue gas, Fahrenheit degrees absolute.

W = flue gas flow rate, pounds per second.

f = coefficient of friction.

L = length of friction duct (approximately equal to H), feet.

D = minimum diameter of round chimney, feet.

x and y = length and width of cross-section of rectangular chimney, feet.

The following notes facilitate the use of Equations 2 and 3.

1. The *barometric pressure*, represented by B_o, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. Hg per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_e = 0.131CO_2 + 0.095O_2 + 0.083N_2 \tag{4}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_e may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltration in the chimney proper is disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft and in height from 100 to 250 ft Equation 5 was deduced[1]:

$$T_e = \frac{3.13T_1\left[\left(\frac{H_b}{2}\right)^{0.86} - 1\right]}{H_b - 3} \tag{5}$$

where

T_1 = temperature at the center of the connection from the breeching, Fahrenheit degrees absolute.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of the Reynolds number[2] in determining the friction factor, f.

The following problem illustrates the use of Equation 2:

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, $B_o = 29.92$ in. Hg; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 2 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^6 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 1 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1.

When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available draft decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 2 and then plotting the results in the manner shown in Fig. 1.

Fig. 2 is a typical chimney performance chart giving the available draft for various gas flow rates and sizes of chimney. This chart is based on an

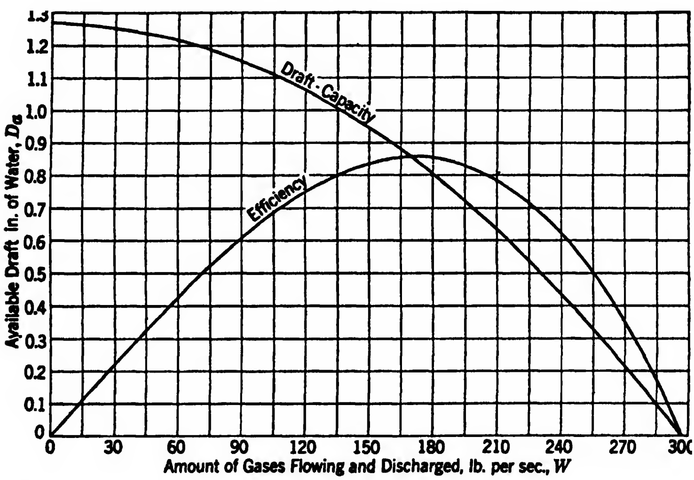


FIG. 1. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific conditions, a new chart may be prepared from Equation 2 or 3.

DETERMINING CHIMNEY SIZES

If the required performance for a proposed chimney is known and *if a chimney-gas velocity is assumed*, Equation 2 can be transposed to yield the necessary height and an equation can be developed for the required diameter. These operations result in the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184 f W_c B_o V^2}{T_c D}} \tag{6}$$

The weight of gas per second, $W = 12.075 \dfrac{D^2 V B_o W_e}{T_e}$ from which

$$D = 0.288 \sqrt{\frac{WT_e}{B_o W_e V}} \tag{7}$$

where

H = required height of chimney above grate, feet.

D = required minimum diameter of chimney, feet.

V = chimney gas velocity, feet per second.

D_r = total required draft, inches of water.

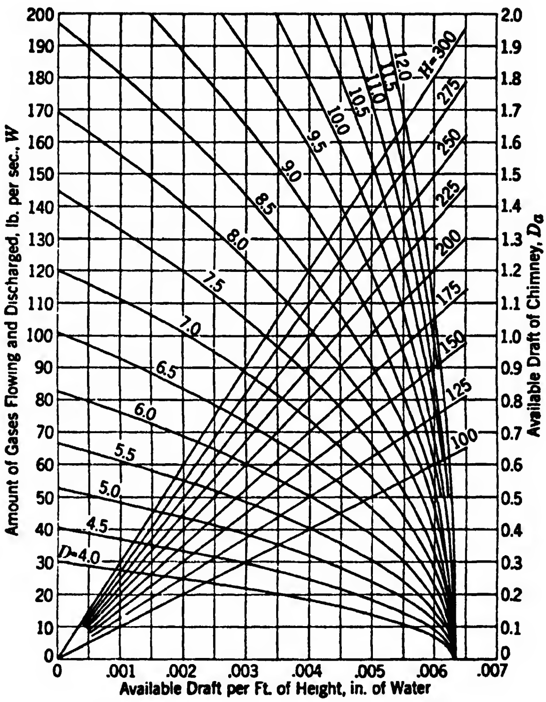


FIG. 2. CHIMNEY PERFORMANCE CHART

To solve a typical example: Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

For large chimneys, it is usual to assume that total construction cost is least when the product HD (height $\times$ diameter) is minimum. On this assumption, the product of Equations 6 and 7 can be differentiated and the differential set equal to zero to find the minimum. Solution for velocity then yields the following equation:

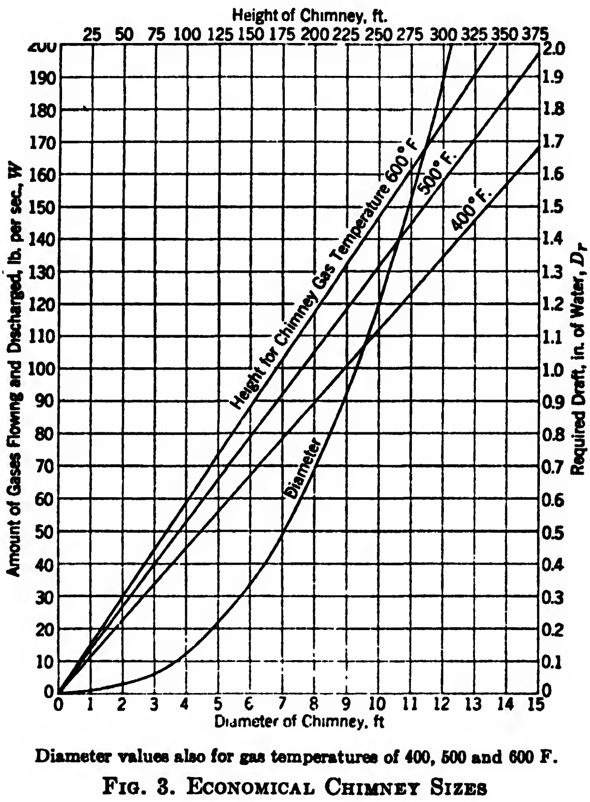
$$V_e = \left(\frac{0.772 T_o \left(\dfrac{W_o}{T_o} - \dfrac{W_e}{T_e} \right) \sqrt{\dfrac{WT_e}{B_o W_e}}}{f W_e} \right)^{2/5} \tag{8}$$

where

V_e = economical chimney gas velocity, feet per second.

Equations 6, 7 and 8 can of course be simplified if values are assumed for some of the factors in it. Some typical figures for boiler plants are:

Average chimney gas temperature 500 F	T_e = 960 F absolute
Average atmospheric temperature 62 F	T_o = 522 F absolute
Average coefficient of friction 0.016	f = 0.016
Average chimney, gas density, 0 F, 1 Atmosphere	W_o = 0.09 lb per cubic foot
Barometer reading, sea level	B_o = 29.92 in. Hg



Diameter values also for gas temperatures of 400, 500 and 600 F.

FIG. 3. ECONOMICAL CHIMNEY SIZES

When these values are substituted in Equations 8, 7 and 6 respectively, the results are:

$$V_e = 13.7W^{1/5} \quad (9) \qquad D = 1.5W^{2/5} \quad (10) \qquad H = 190D_r \quad (11)$$

Fig. 3 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities computed from Equations 9, 10 and 11. They are based on the operating factors used in reducing Equations 6, 7 and 8 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 8.

FACTORS AFFECTING REQUIRED DRAFT

The foregoing considerations deal with chimney size selection when the required draft and flue gas volume and temperature are known. The required draft is, of course, equal to the sum of all the resistances to gas flow from the ash pit door to and including the chimney connection.

Fig. 4 presents information on the fuel-bed draft loss for various kinds of coal burned at different rates and rough generalizations can be given for the losses in the flue passages of boiler or furnace, but, on account of the great differences in such devices, more reliable data on their flue gas volume temperature and flue resistance should be obtained for design purposes from their respective manufacturers.

Flue gases encounter resistance to flow in breechings or smoke pipes and this can probably be treated with sufficient accuracy by means of the

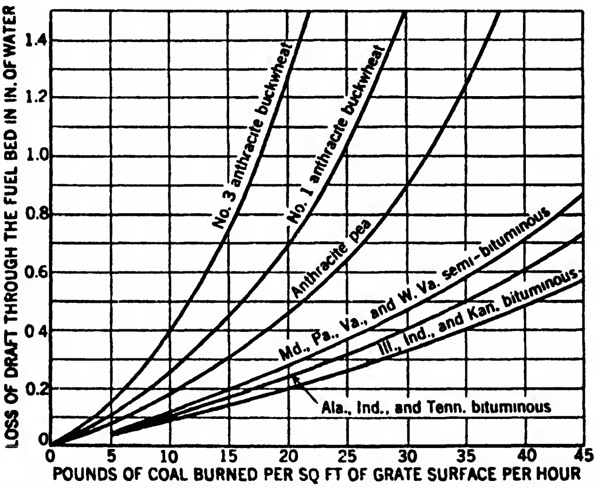


FIG. 4. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

method used for air ducts. (See Chapter 41.) The friction in straight ducts can be estimated by means of the last terms of Equations 2 and 3.

Also, the temperature of flue gases falls during passage through breechings or flue pipes. For uninsulated surfaces this probably can be adequately estimated by assuming a loss of heat from the flue gas of 3 Btu per hour per square foot for each Fahrenheit degree temperature difference between the gases and surrounding air.

DOMESTIC CHIMNEYS

The height of a chimney for a residence or apartment is generally limited by the height of the building since it is desirable to have the chimney architecturally congruous with the building. The height desirable from an architectural viewpoint and the location of the chimney may be disadvantageous to the operation of the boiler or furnace and it is therefore important that the manufacturer of the fuel burning appliance to be installed be consulted in regard to the adequacy of the chimney. A chimney in order to provide satisfactory performance must have adequate *height*

Table 1. Temperature and Draft in 9 in. by 9 in. Masonry Chimney[a]

Chimney Height, Ft		Fuel Oil Equivalent[c] of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch Water	Computed Static Draft Inch Water	Corrected Draft[b]		Indicated Friction Loss Inch Water
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0 5	10	236	142	68	0.040	0.053	0.076	0.089	+0.013
35	31.5	1.5	8	200	158	81	0.043	0.055	0.086	0.098	+0.012
35	31.5	0.5	10	400	224	77	0.081	0.095	0.128	0.142	+0.014
35	31.5	1.5	8	400	278	80	0.092	0.117	0.139	0.164	+0.025
35	31.5	0.5	10	1000	488	77	0.163	0.191	0.206	0.234	+0.028
35	31.5	1.5	8	1000	601	78	0.169	0.214	0.216	0.261	+0.045
30	27.5	0.5	10	250	160	83	0.043	0.049	0.070	0.076	+0.006
30	27.5	1.5	8	250	195	84	0.055	0.064	0.097	0.106	+0.009
30	27.5	0.5	10	400	213	68	0.075	0.084	0.107	0.116	+0.009
30	27.5	1.5	8	400	269	72	0.094	0.108	0.125	0.139	+0.014
30	27.5	0.5	10	1000	460	64	0.146	0.169	0.174	0.197	+0.023
30	27.5	1.5	8	1000	608	72	0.168	0.193	0.204	0.228	+0.025
25	22.3	0.5	10	250	171	92	0.034	0.037	0.072	0.076	+0.003
25	22.3	1.5	8	200	171	94	0.034	0.036	0.074	0.076	+0.002
25	22.3	0.5	10	400	243	78	0.066	0.073	0.097	0.105	+0.006
25	22.3	1.5	8	400	301	79	0.085	0.091	0.116	0.123	+0.006
25	22.3	0.5	10	1000	504	79	0.124	0.137	0.154	0.167	+0.013
25	22.3	1.5	8	1000	642	80	0.146	0.159	0.177	0.190	+0.013
20	17.2	0.5	10	250	179	94	0.029	0.032	0.059	0.062	+0.003
20	17.2	1.5	8	200	188	94	0.031	0.032	0.059	0.060	+0.001
20	17.2	0.5	10	400	270	81	0.058	0.062	0.082	0.086	+0.004
20	17.2	1.5	8	400	297	55	0.075	0.081	0.087	0.093	+0.006
20	17.2	0.5	10	600	330	50	0.086	0.091	0.095	0.100	+0.005
20	17.2	1.5	8	600	425	55	0.104	0.106	0.116	0.118	+0.002
20	17.2	0.5	10	1000	526	37	0.120	0.130	0.123	0.134	+0.010
20	17.2	1.5	8	915	588	44	0.129	0.134	0.135	0.141	+0.005
15	11.3	0.5	10	250	182	87	0.023	0.023	0.041	0.041	+0.000
15	11.3	1 5	8	200	180	84	0.025	0.024	0.042	0.041	−0.001
15	11.3	0.5	10	400	258	32	0.053	0.055	0.054	0.056	+0.002
15	11.3	1.5	8	400	293	43	0.056	0.057	0.060	0.061	+0.001
15	11.3	0.5	10	600	369	34	0.068	0.071	0.069	0.071	+0.003
15	11.3	1.5	8	600	428	40	0.072	0.076	0.075	0.078	+0.004
15	11.3	0.5	10	1000	577	32	0.094	0.092	0.094	0.092	−0.002
15	11.3	1.5	8	1000	654	53	0.092	0.090	0.100	0.098	−0.002

[a]Actual inside dimensions of flue lining 7¼ × 7¼ in.
[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg.
[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

TABLE 2. TEMPERATURE AND DRAFT IN 9 IN. BY 13 IN. MASONRY CHIMNEY[a]

Chimney Height, Ft		Fuel Oil Equivalent[e] of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch Water	Computed Static Draft Inch Water	Corrected Draft[b]		Indicated Friction Loss Inch Water
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	204	125	78	0.035	0.034	0.078	0.077	−0.001
35	31.5	1.5	8	206	155	82	0.047	0.052	0.093	0.098	+0.005
35	31.5	1.5	8	600	355	84	0.140	0.141	0.193	0.194	+0.001
35	31.5	0.5	10	600	272	78	0.104	0.117	0.152	0.165	+0.013
35	31.5	0.5	10	1000	438	77	0.163	0.178	0.205	0.221	+0.015
35	31.5	1.5	8	1000	542	84	0.189	0.201	0.236	0.247	+0.012
30	27.5	0.5	10	260	155	83	0.040	0.045	0.081	0.086	+0.005
30	27.5	1.5	8	202	159	83	0.048	0.047	0.089	0.088	−0.001
30	27.5	0.5	10	605	278	80	0.089	0.103	0.129	0.144	+0.014
30	27.5	1.5	8	595	378	72	0.130	0.144	0.161	0.175	+0.014
30	27.5	0.5	10	1007	438	91	0.135	0.148	0.179	0.191	+0.013
30	27.5	1.5	8	995	609	89	0.166	0.168	0.225	0.227	+0.002
25	22.3	0.5	10	258	155	86	0.034	0.034	0.069	0.069	0.000
25	22.3	1.5	8	200	160	86	0.039	0.037	0.074	0.072	−0.002
25	22.3	0.5	10	600	312	86	0.086	0.090	0.122	0.126	+0.004
25	22.3	1.5	8	592	382	83	0.106	0.110	0.140	0.144	+0.004
25	22.3	0.5	10	1007	402	85	0.120	0.127	0.154	0.161	+0.007
25	22.3	1.5	8	1007	607	84	0.151	0.153	0.184	0.185	+0.002
20	17.2	0.5	10	280	175	79	0.030	0.028	0.062	0.060	−0.002
20	17.2	1.5	8	250	192	81	0.044	0.041	0.069	0.066	−0.003
20	17.2	0.5	10	598	324	74	0.069	0.079	0.089	0.099	+0.010
20	17.2	1.5	8	604	410	74	0.090	0.094	0.111	0.115	+0.004
20	17.2	0.5	10	1015	496	79	0.094	0.107	0.115	0.128	+0.013
20	17.2	1.5	8	1000	645	74	0.122	0.127	0.140	0.147	+0.005
15	11.3	0.5	10	270	172	88	0.022	0.021	0.041	0.040	−0.001
15	11.3	1.5	8	200	166	85	0.022	0.020	0.040	0.038	−0.002
15	11.3	0.5	10	604	356	86	0.055	0.052	0.073	0.070	−0.003
15	11.3	1.5	8	599	431	84	0.066	0.062	0.082	0.078	−0.004
15	11.3	0.5	10	1000	560	82	0.066	0.074	0.082	0.091	+0.008
15	11.3	1.5	8	1000	682	85	0.083	0.082	0.101	0.100	−0.001

[a]Actual inside dimensions of flue lining 6½ × 11 in.

[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg.

[e]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

and *area*, be of permanently tight construction and should be as smooth internally as practicable.

It should be remembered that mechanically fired devices, oil burners and stokers, are equipped with blowers so that, with these devices, the chimney is not required to overcome the resistance of a fuel bed. Nevertheless, a draft in the fire box, of about 0.03 in. of water is considered desirable so that any small openings in the fire box or flue passages will result in leakage of air inward, and not leakage of combustion products outward. This is not to be taken to condone leaks in fire boxes. Such leaks adversely affect plant efficiency.

OBSERVED TEST PERFORMANCE

The observed performances of some brick chimneys[8] are given in Tables 1 and 2.

The tests on which these data are based were made at various outside temperatures as shown and, to make them comparable among themselves, the observed drafts were corrected to 32 F, 1 atmosphere pressure, by the formula:

$$O_2 = O_1 + S_2 - S_1 \tag{12}$$

where

S_1 = computed static (theoretical) draft, experimental conditions.

S_2 = computed static (theoretical) draft, standard conditions.

O_1 = observed draft, experimental conditions.

O_2 = observed draft corrected to standard conditions.

It will be noted that the observed draft exceeded the computed static draft during some observations on the shorter chimneys. This is mainly attributed to the draft producing effect of the hot gases immediately above the chimney. By means of a manometer it was found that a measurable draft existed in this gas column for some distance above the chimney top. However, the temperatures in the chimney were measured with unshielded thermocouples and the actual gas temperatures may have been higher for this reason than the observed temperatures on which the computations of draft were based. The tests were made in calm weather.

Tests were made at the *National Bureau of Standards* to find the draft produced by round, metal smoke pipe set in vertical position to act as chimneys. Curves are presented in Fig. 5 showing the computed static or theoretical draft for short chimneys for various heights and temperatures. For this purpose, the density of chimney gases was assumed to be the same as that of air at the same condition, since the error thus introduced was not considered important in this case. The results of the tests showed that the following procedures would yield the available draft for 6-in. flue pipe used as a chimney within 10 per cent for the range shown and for fuel burning rates from about one-quarter to three-quarters of a gallon of oil per hour.

Using the temperature at the smoke collar of the heater, find the static draft corresponding to the available chimney height. Then:

1. If the chimney is bare, multiply the static draft by 0.76 to find the available draft.

2. If the chimney is insulated with 1 in. of air-cell material with a ½-in. air space, multiply the static draft by 0.85 to find the available draft.

3. If the chimney is insulated with 1 in. of air-cell material with a 1-in. air space, open at top and bottom for ventilation, multiply the static draft by 0.81 to find the available draft.

4. If the chimney is insulated with 1 in. of air-cell material and has a 1-in. air space closed at top and bottom to prevent ventilation, multiply the static draft by 0.85 to find the available draft.

The use of the 1-in. air-cell asbestos insulation in the tests discussed is not to be construed as an approval of such insulation in all cases in regard to fire resistance. Several laboratories are working on the fire resistance

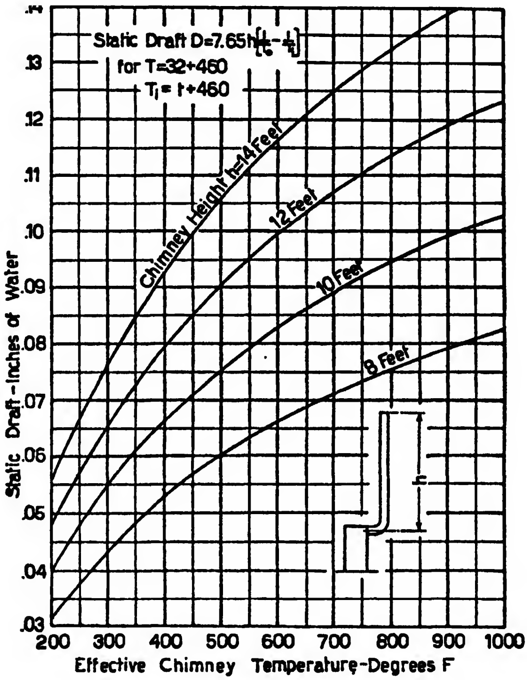


Fig. 5. Computed Static Draft for Short Chimneys

aspects of the problem but definite rules are not yet available. For coal- or oil-burning devices, a bare smoke pipe is probably safe if kept 2 ft or more from any woodwork and the better the pipe is insulated, or the lower its temperature, the nearer it can be placed to combustible materials.

DRAFT REQUIREMENTS OF DOMESTIC APPLIANCES

Typical flue-gas temperatures and drafts required at rated output for several kinds of domestic heating appliances[4] are contained in Table 3.

CHIMNEYS FOR GAS HEATING

Heating appliances designed to burn gas as well as appliances converted to gas burning, except those equipped with power type burners and excepting conversion burner installations in excess of 400,000 Btu per hour input in large steel boilers, are always equipped with a draft hood attached to the flue outlet of the appliance. This draft hood is required if the appliance is to meet the approval requirements of the *American Gas As-*

sociation and the *American Standards Association* and is essential for safe operation. It is designed to prevent excessive chimney draft which would lower appliance efficiency, to prevent a blocked flue or a down draft in the chimney from impairing combustion, to provide a relief opening for the products of combustion during down draft or blocked flue conditions, and to prevent spillage of the products of combustion to the space surrounding the appliance if there is a chimney draft equivalent to that provided by a 3-ft chimney. As the draft hood is designed without moving parts, the relief opening is always open and consequently some air is drawn into the chimney. While the air drawn in lowers the gas temperature in the chimney, it also lowers the dew-point of the gases and tends to prevent condensation.

The installation of conversion burner equipment in large boilers is usually made in accordance with regulations of the local gas company or standards developed under *American Standards Association* procedure[5].

TABLE 3. DRAFTS REQUIRED BY TYPICAL DOMESTIC HEATING DEVICES OR APPLIANCES

DEVICE	DRAFT, INCHES WATER	STACK TEMPERATURE F DEG
Space Heater, Oil Burning, Pot Burner	0.06 to 0.08	1000
Warm Air Furnace, Oil Burning, Pot Burner	0.06	860
Warm Air Furnace, Hand Fired	0.06[b]	900
Floor Furnace, Oil Burning, Pot Burner	0.06	860
Mechanical Oil Burner, Less than 5 gph	0.03[a]	.....
Mechanical Oil Burner, More than 5 gph	0.05[a] or less	.....
Cooking Stove, Solid Fuel	0.04[b]	400
Space Heater, Coal Burning	0.06[b]	900

[a] Draft in fire-box. [b] For chestnut sized anthracite.

In such installations a definite chimney draft may be required for proper combustion and consequently the foregoing reference to the use of draft hoods would not apply.

The products of complete combustion of gas are water vapor (H_2O) and carbon dioxide (CO_2). In the case of manufactured gas, the presence of organic sulfur compounds, generally between 3 and 15 grains per hundred cubic feet, gives rise to minute percentages of sulfur dioxide and sulfur trioxide.

The volume of water vapor in the flue products from natural or coke oven gas is about twice the volume of carbon dioxide. It is extremely important that the chimney be tight and resistant to corrosion not only of moisture, but also of dilute sulfur trioxide.

Vitreous tile linings with joints which prevent retention of moisture and linings made of non-corrosive materials are advantageous. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt chromate will provide excellent protection.

Advice regarding recommended practice and materials for flue connections and chimney linings can usually be obtained from the local gas company and should be given careful consideration.

Since a gas designed appliance must be able to operate at rated input (plus 10 or 15 per cent) without chimney connection, and without producing carbon monoxide, the only function of the chimney is to remove the products of combustion from the room. The chimney provides draft to

overcome the friction in the flue pipe and chimney, but does not draw air into the appliance.

Chimneys for venting appliances designed for burning gas can therefore be low in height, but must have adequate area. The height is usually established by the building height. Chimney sizes are usually selected on the basis of Btu input of the appliance. One chart[6] designed to facilitate selection is shown in Fig. 6. The assumptions made in preparing the chart as well as its limitations should be noted carefully.

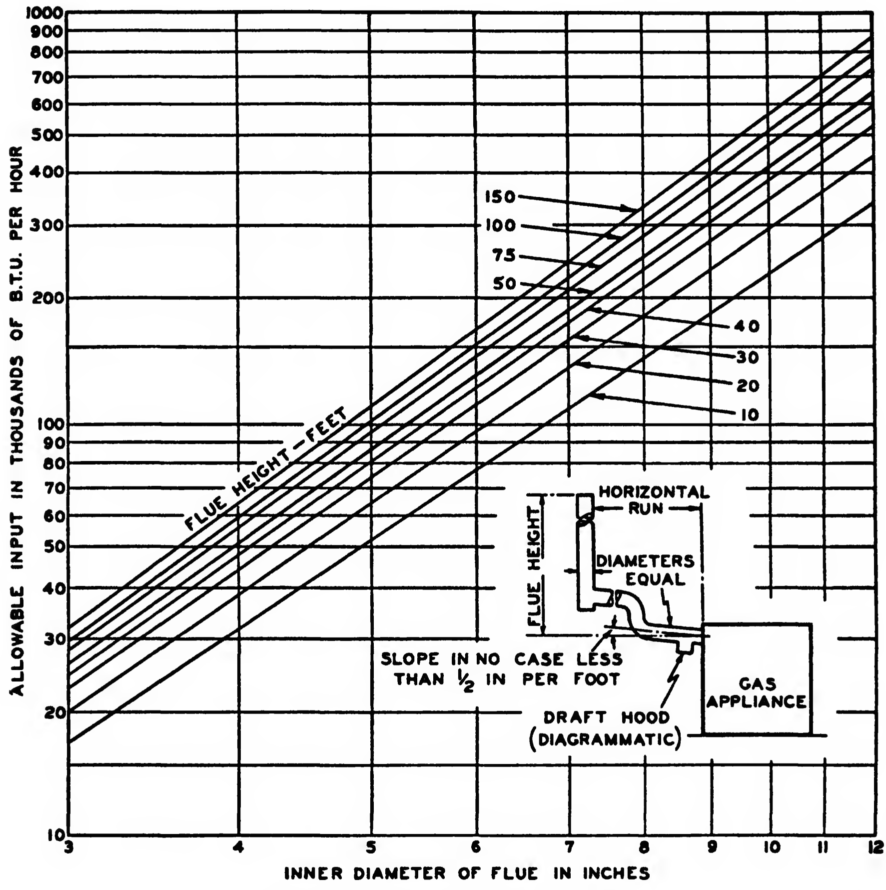


FIG. 6. ALLOWABLE BTU INPUT TO CIRCULAR FLUES FOR DOMESTIC GAS APPLIANCES WITH DRAFT HOODS

NOTES APPLYING TO FIG. 6:

1. Chart is based on: average flue temperature of 150 F, outside temperature of 60 F, barometric pressure of 30 in. Hg, 100 per cent excess air and 100 per cent dilution at draft hood.

2. Based on terra-cotta lined flues. With rough brick flues, capacities are 15 per cent less.

3. Based on condition that horizontal run is not greater than 20 ft except for a flue height less than 20 ft, in which case the horizontal run is not to have greater length than the height of the flue.

4. Two long radius elbows are included in the horizontal run, the diameter of which is equal to that of the flue.

5. Each additional elbow reduces the allowable horizontal run by a length in feet equal to the diameter in inches.

6. When the horizontal run has an effective length in excess of that given (or additional elbows) the next larger size of flue should be chosen. It is desirable that long horizontal runs be insulated to reduce heat loss of flue products and to conserve draft.

7. Capacities should be reduced 3.5 per cent for each 1000 ft above sea level.

Since Fig. 6 has been prepared for circular flues, relative capacities for rectangular and semi-elliptical flues[7] are shown in Fig. 7.

When a flue is connected to several appliances, the number of horizontal runs of various sizes which may be substituted for the single run having a diameter equal to that of the flue may be obtained from Table 4.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Building Code recommended by the *National Board of Fire Underwriters*, Article XI, Sections 1101 to 1105, in which the following are some of the important provisions listed in the 1943 edition:

(*a*) Chimneys erected within or attached to a structure shall be constructed of brick, of solid block masonry, or of reinforced concrete.

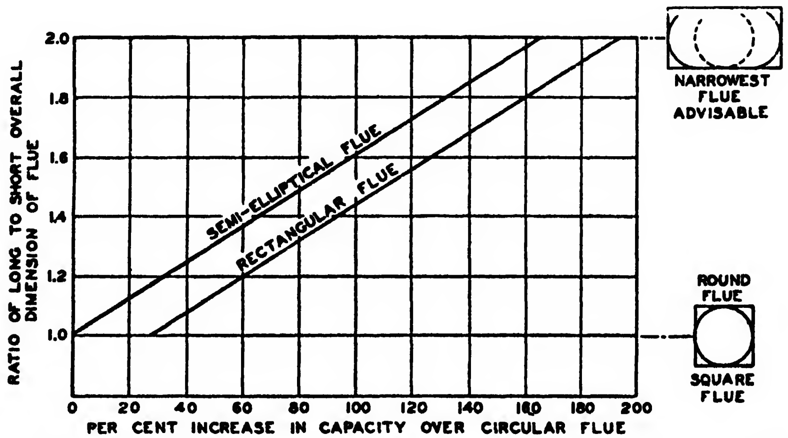


FIG. 7. CAPACITY OF A RECTANGULAR FLUE OR A SEMI-ELLIPTICAL FLUE, WITH SEMI-CIRCULAR ENDS HAVING ITS MINIMUM WIDTH EQUAL TO THE DIAMETER OF A CIRCULAR FLUE, COMPARED WITH THE CAPACITY OF THE CIRCULAR FLUE

(*b*) Chimneys shall extend at least 3 ft above the highest point where they pass through the roof of the building and at least 2 ft higher than any ridge within 10 ft of such chimney.

(*c*) Every such chimney shall be properly capped with brick, terra cotta, stone, cast-iron, concrete or other approved non-combustible, weatherproof material.

(*d*) Chimneys shall be wholly supported on approved masonry or self-supporting fireproof construction.

(*e*) No such chimney shall be corbeled from a wall more than 6 in.; nor shall such chimney be corbeled from a wall which is less than 12 in. in thickness unless it projects equally on each side of the wall; provided that in the second story of two-story dwellings corbeling of chimneys on the exterior of the enclosing walls may equal the wall thickness. In every case the corbeling shall not exceed 1 in. projection for each course of brick projected.

(*f*) No change in the size or shape of a chimney, where the chimney passes through the roof, shall be made within a distance of 6 in. above or below the roof joists or rafters.

(*g*) Smoke flues for warm air, hot water and low pressure steam heating furnaces shall have walls not less than 8 in. thick; the walls may be of solid masonry using brick, stone or concrete, or of solid moulded or solid cast chimney units of concrete, or of burned clay, or of suitably reinforced solid concrete cast in place; provided that for stone masonry, other than sawed or dressed stone in courses, the thickness shall be not less than 12 in. The walls shall be properly bonded, or tied with non-corrosive

metal anchors. In dwellings and buildings of like heating requirements the thickness of the chimney walls may be reduced to not less than 3¾ in. when lined with a flue lining conforming to the Code requirements.

(*h*) Required flue linings shall be made of fire clay or other refractory clay to withstand the action of flue gases and to resist, without softening or cracking, the temperatures to which they will be subjected, but not less than 2,000 F, or of cast-iron of approved quality, form and construction.* Approved corrosion resistant linings may be used in flues for gas appliances.

(*i*) Required clay flue linings shall be not less than ⅝ in. thick for the smaller flues and increasing in thickness for the larger flues.

(*j*) Flue linings shall be built ahead of the construction of the chimney as it is carried up, carefully bedded one on the other in mortar as hereinafter specified with close fitting joints left smooth on the inside.

(*k*) Flue linings shall start from a point not less than 8 in. below the intake. They shall extend, as nearly vertically as possible, for the entire height of the chimney. It is recommended that flue linings be extended 4 in. above the top or cap of the chimney.

(*l*) Only Portland cement mortar, cement lime mortar or fire clay mortar shall be used in setting flue linings.

For *gas appliances* the Building Code specifies lined chimneys and metal smoke stacks for all appliances which may be converted readily to the use

TABLE 4. EQUIVALENT FLUE PIPE SIZES[a]

Diameter of Horizontal Runs	Size of Flue						
	3	4	5	6	8	10	12
3	1	2	3	5	9	12	22
4		1	2	3	5	7	11
5			1	2	3	4	7
6				1	2	3	5
8					1	2	3

[a] Comfort Heating (*American Gas*

of solid or liquid fuel and also for all boilers and furnaces except those having a flue gas temperature not exceeding 550 F at the outlet of the draft hood when burning gas at the manufacturer's rating and which may therefore be connected to *Type B* vent piping. Approved *Type B* vent piping is non-combustible, corrosion resistant piping of adequate strength and heat insulating value, and having bell and spigot or other acceptable joints. Fig. 6 may be used for selection of vent pipe size.

Important points to be considered in the use of *Type B* vent piping are:

1. *Type B* flues must be plainly and permanently marked, at the point where the vent connection enters the flue: *For use of gas appliances only.*

2. *Type B* vent material should not be used for external chimney flues and external runs of it should not exceed 3 ft outside the building roof. When this requirement makes it necessary to cross over through attic space, the piping should be pitched not less than 45 deg.

3. Because of the small size and low temperature, *Type B* vents should be provided with a vent cap with wire screening to prevent building of birds' nests.

4. Each appliance should have the equivalent of a 4 in. diameter *Type B* vent, even though the appliance may have a 3 in. flue collar. A typical minimum vertical flue size for a frame dwelling is 6 in. in diameter or equivalent.

5. When several floor furnaces are to be vented, it is acceptable practice to connect each of these by means of 4 in. vents to a common 6 in. vertical vent. Lateral

piping must have adequate pitch, ¼ in. per foot, and should not exceed 20 ft in horizontal length.

6. An alternate method of connecting several appliances is to run separate *Type B* vents to the attic and then to connect them by means of cross-over piping and Y fittings to a common vertical vent passing through the roof. This reduces the number of holes in the roof.

All flue mortar for flues or vent pipes from gas burning appliances shall be acid resisting.

GENERAL CONSIDERATIONS FOR CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys[7]. Horizontal winds have an aspirating effect as they cross the chimney and are an aid to draft. However, surrounding objects, such as trees or other buildings, may affect the direction of the wind at the chimney top and may even direct it down the chimney, tending to reduce the draft or even to cause it to be negative. Although the chimney should extend well above the highest part of the roof, it is impracticable to carry it much beyond this point.

It is also important to consider the course of the air supply for proper combustion. Usually the boiler or furnace is loacted in the basement. When the furnace room has windows or doors opening to the outside on two or more sides of the house, the leakage of air will be sufficient for combustion, even though the windows and doors may be shut. If, however, the leakage is not sufficient to prevent an appreciable drop of pressure in the furnace room below that of the air outside, the chimney draft will be reduced by the difference between the atmospheric pressure outside and that inside the boiler room. In case the boiler room is fairly tight and is open to the outside on only one side of the house, then the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room the draft will be increased, but if blowing in the opposite direction the draft may be decreased.

It is not to be assumed that increasing the cross-section area of a chimney will always effect a cure for poor draft. The opposite result may be experienced because of the cooling effect of the larger area. This reduces the theoretical draft and the velocity of the gases, and affords a greater opportunity for counter currents in the chimney. Sometimes the only practical remedy for a chimney with bad draft, when the chimney is of the proper size and is affected by conditions beyond control, is to resort to mechanical draft. This can often be done at small expense and the arrangement can be such that the fan or blower need be operated only when conditions are bad.

Two or more chimneys, either large or small, should never be connected together. If connected at the bottom, hot gases in the inverted U-tube thus formed would be in unstable equilibrium. Cold air would descend through one such chimney, from the top, and drive the hot gases out of the other and thus annul the draft.

More than one device can be served by one chimney. Batteries of boilers are commonly connected to a single chimney in power plants. However, if two or more chimneys are used, each chimney should be used separately for part of the boilers, and not connected in manifold with another chimney, in order to avoid the difficulty described previously.

In domestic installations it is sometimes necessary to serve a space heater or cooking stove and a water heater with the same chimney flue. This is not desirable, especially for low chimneys, since doors left open on one

device while it is unfired will tend to annul the draft on another device. Gas burning devices, with their draft hoods and lack of draft dampers, are especially bad in this respect. The traditional method of avoiding this with brick chimneys has been to construct multiple-flue chimneys, so that each fuel-burning device could be served by a separate opening. If two devices must be served by one flue-opening in a chimney, their connections to the chimneys should not be located opposite each other. The connection from the larger device should be reasonably low down and that from the smaller, up near the ceiling, so that each device can be serviced as well as possible, regardless of the treatment of the other.

Excessive height in a chimney does no harm but means for controlling the draft are more than ordinarily essential if the chimney is too large in capacity. Coal-burning devices often have air leaks around the fire-box and the draft doors sometimes fit poorly so that the fire cannot be controlled at a low rate. Perhaps the simplest remedy for such cases is the barometric damper which admits air into the flue pipe and thus reduces draft.

Table 5. Standard Sizes of Flue Linings

Minimum Net Inside Area (Square Inches)	[a] Nominal Dimensions (Inches)	Outside Dimensions (Inches)	Length (Inches)	Minimum Wall Thickness (Inches)	Approximate Maximum Outside Corner Radius (Inches)
15	4×8	3.5×7.5	24 ±0.5	0.5	
20	4×12	3.5×11.5	24 ±0.5	0.625	
27	4×16	3.5×15.5	24 ±0.5	0.75	
35	8×8	7.5×7.5	24 ±0.5	0.625	
57	8×12	7.5×11.5	24 ±0.5	0.75	
74	8×16	7.5×15.5	24 ±0.5	0.875	
87	12×12	11.5×11.5	24 ±0.5	0.875	
120	12×16	11.5×15.5	24 ±0.5	1.	
162	16×16	15.5×15.5	24 ±0.5	1.125	
208	16×20	15.5×19.5	24 ±0.5	1.25	
262	20×20	19.5×19.5	24 ±0.5	1.375	
320	20×24	19.5×23.5	24 ±0.5	1.5	
385	24×24	23.5×23.5	24 ±0.5	1.625	

[a] Cross section of flue lining shall fit within rectangle of dimension corresponding to nominal size.

Clay flue linings used in the construction of chimneys are now generally available in sizes shown in Table 5. These sizes were obtained from the publication American Standard Sizes of Clay Flue Linings, A 62.4-1947, *American Standards Association.*

Directions for building chimneys for fireplaces are contained in Department of Agriculture Farmers' Bulletin No. 1889.

It is considered bad practice to connect any heating device to a fireplace flue unless the fireplace is effectively sealed.

REFERENCES

[1] Notes on Power Plant Design, by E. F. Miller and James Holt (*Massachusetts Institute of Technology*, 1930).

[2] A.S.H.V.E. Research Report No. 1105—Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik and N. Ivanovic (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 35) and for more complete discussion see Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

[3] Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach and J. T. Duck (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 351).

[4] *National Bureau of Standards Commercial Standards:* CS101-43 Oil-Burning Space Heaters Equipped With Vaporizing Pot-Type Burners, CS75-42 Automatic Mechanical Oil Burners Designed for Domestic Installations, CS(E)104-43 Warm Air Furnaces Equipped With Vaporizing Pot-Type Burners; and *Trade Standards:* TS3536a Solid-Fuel Burning Forced Air Furnaces, TS3518 Oil-Burning Floor Furnaces Equipped With Vaporizing Pot-Type Burners.

[5] American Standard Requirements for Installation of Domestic Gas Conversion Burners, A.S.A. Z21.8, 1948, *American Standards Association*).

[6] Comfort Heating, 1938, p. 71 (*American Gas Association*).

[7] Comfort Heating, 1938, p. 74 (*American Gas Association*).

[8] Chimneys and Draft (Chapter 32 in Winter Air Conditioning, by S. Konzo, published by *National Warm Air Heating and Air Conditioning Association*, 1939).

CHAPTER 20

ESTIMATING FUEL CONSUMPTION FOR SPACE HEATING

Basis of Fuel Estimates, Calculated Heat Loss Method, Degree-Day Method, Estimating Fuel Consumption, Degree-Day as an Operating Unit, Maximum Demands and Load Factors, Seasonal Efficiency

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (*a*) reliable records of past heat or fuel consumptions of the building under consideration will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (*b*) where no past records exist useful data can sometimes be obtained from records of similar types of buildings with similar plants *in the same locality;* (*c*) records of consumption, which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (*d*) estimates based on computed heat losses without the benefit of operating data are wholly dependent on the degree to which the computation represents the actual facts.

Estimates based on computed heat losses alone are especially necessary where unusual operating conditions, such as excessive ventilation, abnormal inside temperatures, heat gains from external sources, etc., are encountered or where no information is available as to former consumption as in the case of proposed buildings of unusual design.

In preparing, interpreting and evaluating heat or fuel consumption estimates it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened there is more chance that some factor not allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, namely, (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and constant temperatures are assumed for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, poor heating systems, winter

heat gains, such as sun effect, and many others. In order to apply this method the hourly heat loss from the building under maximum load, or design condition, is computed following the principles discussed in Chapters 6 and 8, and the method described and illustrated in Chapter 14.

In predicting fuel consumption for heating a building by the Calculated Heat Loss Method, the general equation is:

$$F = \frac{H(t - t_a)N}{E(t_d - t_o)C} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).

H = calculated heat loss, Btu per hour, during the design hour, based on t_o and

(generally $H = H_t + H_i$ but may on occasion equal $H_t + \frac{H_i}{2}$

t = average inside temperature maintained during heating period, Fahrenheit degrees.

t_a = average outside temperature through estimate period, Fahrenheit degrees (for cities with an Oct. 1–May 1 heating season).

t_d = inside design temperature, Fahrenheit degrees (usually 70 F).

t_o = outside design temperature, Fahrenheit degrees (see Table 1 in Chapter 14).

N = number of heating hours in estimate period (for an Oct. 1–May 1 heating season, 212 days × 24 hr = 5088).

E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.

C = heating value of one unit of fuel or energy.

Although the assumption of an Oct. 1–May 1 heating season is reasonably accurate in the well-populated New York-Chicago zone, it is not valid as far north as Minneapolis nor farther south than Washington, D. C. and St. Louis. Consequently, it is suggested that allowance be made for this variation, especially in the far north or southern cities.

Example 1. A residence building is to be heated to 70 F from 6 A.M. to 10 P.M. and 55 F from 10 P.M. to 6 A.M. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −10 F outside. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution. The heating value of steam may be taken as 1000 Btu per lb, and since it is purchased steam, the efficiency can be assumed as 100 per cent. Assume average outside temperature as 36.4 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 55)}{24}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\ (65 - 36.4)\ 5088}{1.00[70 - (-10)]1000} = 218{,}275 \text{ lb.}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at $8 per ton and with a calorific value of 11,000 Btu per lb, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. Substituting in Equation 1: $F = \frac{120{,}000\ (65 - 36.4)\ 5088}{0.55[70 - (-10)]11{,}000} = 36{,}079$ lb
18 tons, which, at $8 per ton, costs $144.

Example 3. What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 P.M. to 7 A.M. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution. The average hourly temperature is:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu.}$$

$$M = \frac{79{,}000\ (66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

2204.6 × \$0.07 = \$154.32 = estimated fuel cost per year of heating building.

Several time-saving procedures have been devised for quickly estimating the hourly Btu loss of one and two-story residences in order that fuel estimates can be predicted more quickly from Equation 1. A graphical method of calculating heat losses has been developed[1] which makes possible a quick solution if the gross wall, ceiling, or floor areas and respective transmission coefficients are known.

The Federal Housing Administration has originated a short-cut formula for residential heat loss determinations which makes use of the floor area and three selected transmission coefficients. The formula was developed to apply to detached houses approximately rectangular in shape with total exterior door and window areas equal to about 25 per cent of the floor area and with a floor area not greater than about 1500 sq ft. Equation 2 is for a one-story residence and Equation 3 is intended for two-story structures.

$$H_1 = A\ (G + U_w + U_c + U_f)\ (t_d - t_o) \tag{2}$$

$$H_2 = A\ (G + 1.2\ U_w + 0.5\ U_c + 0.5\ U_f)\ (t_d - t_o) \tag{3}$$

where

H_1 = heat loss from one-story residence, Btu per hour.

H_2 = heat loss from two-story residence, Btu per hour.

A = floor area, square feet, measured to the inside faces of enclosing walls and is the sum of the following areas: (1) all the area on each principal floor level; (2) the area of all finished habitable attic rooms, including bathrooms, toilet compartments, closets, and halls; (3) all other areas intended to be heated and not located in the basement.

G = glass and infiltration factor for ordinary construction: (0.45 for no weatherstripping or storm windows), (0.40 for weatherstripping), (0.30 for storm windows with or without weatherstripping).

U_w = coefficient of transmission for outside wall.

U_c = coefficient transmission for ceiling.

U_f = coefficient of transmission for floor.

t_i = inside design temperature, Fahrenheit degrees.

t_o = outside design temperature, Fahrenheit degrees.

Notes for application of Equations 2 and 3.

1. The calculation of heat loss from heated spaces into adjacent spaces such as attics, basementless areas, and heated or unheated garages shall be based on the assumption that the temperature of such adjacent spaces is the same as the outside design temperature.

2. For all floors over basements or other warmed spaces assume $U_f = 0$.

3. For structures having concrete slab floors laid on the ground a modified application of the formula may be made. Assume $U_f = 0$ and calculate the heat loss in accordance with the check formula. Then add the slab loss determined in accordance with the procedure developed by the National Bureau of Standards and described in BMS Report 103.

4. No basement area is to be included in the formula calculation. If finished habitable rooms in the basement are to be heated, the additional heat loss should be calculated separately and added to the amount obtained by the formula.

Both the graphical method and short-cut formulas, when used within the limitations established, have been found to give reasonably accurate results for the average residence, but if precise estimates are required, the procedure outlined in Chapter 14 should be used.

In the case of gravity warm air heating installations, the load was formerly expressed in square inches of leader pipe which can be converted into Btu per hour by multiplying the square inches of leader area by 111, 167, and 200 for first, second, and third floor respectively.

Example 4. What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[2], and with four 12 in. and two 8 in. round leaders to the first floor and six 10 in. leaders to the second floor, if the gas has a heating value of 500 Btu per cu ft, the plant operates at a 70 per cent seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hr long?

Solution. The area of the round leaders is: 12 in., 113 sq in.; 10 in., 79 sq in.; and 8 in., 50 sq in. The total Btu transmitted is:

First Floor: $[(4 \times 113) + (2 \times 50)] \times 111 = 61{,}272$ Btu per hr.
Second Floor: $(6 \times 79) \times 167 = 79{,}158$ Btu per hr.

Total 140,430 Btu per hr.

Substituting this total heat loss value as H in Equation 1 gives:

$$F = \frac{140{,}430(65 - 45)5088}{0.70(70 - 10)500} = 680{,}483 \text{ cu ft gas.}$$

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is considered by many to be of more value for practical use.

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. The *American Gas Association*[3] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the mean outside temperature. In other words, on a day when the mean temperature was 20 deg below 65 F, twice as much

gas was consumed as on a day when the temperature was 10 deg below 65 F. For any one day, when the mean temperature is less than 65 F, there are as many degree-days as there are degrees difference in temperature between the mean temperature for the day and 65 F. Degree-days may be calculated on other than the 65 F base but are seldom used and are of little value except where the inside temperature to be maintained as, for example, in warehouses, differs greatly from the usual inside temperature range of 68 F to 72 F.

Table 1 lists the average number of degree-days, which have occurred over a long period of years, by months and the yearly totals for various cities in the United States, Canada and Newfoundland. The values for United States cities were calculated by taking the difference between 65 F and the daily mean temperature computed as half the total of the daily maximum and the daily minimum temperatures. The monthly averages were obtained by adding daily degree-days for each month each year and dividing by the number of days in the month; then totaling the respective calendar monthly averages for the number of years indicated and dividing by the number of years. The total or long term yearly average degree-day value is the summation of the 12 monthly averages. Degree days for Canadian cities were supplied by the Canadian Meteorological Division of the Department of Transport and were computed from the mean temperature normals on record for the various stations.

Any attempt to apply the degree-day method of calculating fuel consumption for less than one month would be of very little value. It should be noted that this method of calculation is based on a long term average and cannot be expected to coincide with any single year in calculating fuel requirement. Individual yearly degree-day calculations will vary as much as 20 per cent above and below the long term average.

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

Studies made by the *National District Heating Association*[4] of the metered steam consumption of 163 buildings located in 22 different cities and served with steam from a district heating company, substantiate the fact that the 65 F base originally chosen by the gas industry is approximately correct.

Formula for Degree-Day Method

The general equation for calculating the probable fuel consumption by the degree-day method is:

$$F = U \times N \times D \tag{4}$$

where

F = fuel consumption for the estimate period.

U = unit fuel consumption, or quantity of fuel used per (degree-day) (*building load unit*).

N = number of *building load units* (when available use calculated hourly heat loss instead of actual amount of radiation installed).

D = number of degree-days for the estimate period.

TABLE 1. AVERAGE MONTHLY AND YEARLY DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND [a, b] (BASE 65F)

State	Station	Years	No. of Seasons	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Ala.	Anniston	05/06-40/41	36	0	0	10	135	388	600	609	513	361	152	37	1	2806
	Birmingham A	98/99-45/46	48	0	1	10	111	348	586	591	497	313	130	23	1	2611
	Mobile	98/99-45/46	48	0	0	1	44	203	377	397	314	175	52	3	0	1566
	Montgomery	98/99-45/46	48	0	0	4	71	279	484	494	405	239	85	10	0	2071
Ariz.	Flagstaff	98/99-40/41	43	43	70	244	573	847	1111	1167	970	889	668	469	190	7241
	Phoenix	98/99-45/46	48	0	0	0	18	166	384	402	263	154	47	7	0	1441
	Yuma	98/99-40/41 45/46	44	0	0	0	9	113	306	318	182	85	22	1	0	1036
Ark.	Bentonville	06/07-40/41	35	1	1	38	216	516	810	879	716	519	247	86	7	4036
	Fort Smith A	98/99-45/46	48	0	0	12	128	410	717	763	615	390	154	36	1	3226
	Little Rock A	98/99-45/46	48	0	0	11	120	383	668	704	579	367	145	31	1	3009
Calif.	Eureka	98/99-45/46	48	281	269	274	344	411	518	541	478	504	440	391	307	4758
	Fresno A	98/99-45/46	48	0	0	5	77	309	562	573	380	289	152	52	4	2403
	Independence	98/99-40/41	43	0	0	28	216	512	778	799	619	477	267	120	18	3834
	Los Angeles	98/99-45/46	48	1	0	5	43	110	225	272	235	212	158	103	27	1391
	Needles	17/18-38/39	22	0	0	0	19	217	416	447	243	124	26	3	0	1495
	Point Reyes	98/99-40/41	43	350	336	263	282	317	425	467	406	437	413	415	363	4474
	Red Bluff A	98/99-33/34 38/39-40/41 44/45-45/46	41	0	0	12	97	345	592	601	419	328	178	72	9	2653
	Sacramento	98/99-45/46	48	2	1	15	98	332	582	595	405	326	202	101	21	2680
	San Diego A	98/99-45/46	48	5	1	9	60	143	252	300	257	230	172	118	49	1596
	San Francisco	98/99-45/46	48	196	179	121	139	241	420	460	340	317	272	255	197	3137
	San Jose	06/07-40/41	35	21	21	52	151	329	512	527	383	339	249	167	72	2823
Colo.	Denver	98/99-45/46	48	8	8	126	411	716	1005	1023	897	790	516	275	64	5839
	Durango	04/05-40/41	37	25	37	201	535	861	1204	1271	1002	859	615	394	139	7143
	Grand Junction	98/99-45/46	48	1	1	59	347	743	1138	1218	883	671	377	152	23	5613
	Leadville	07/08-40/41	34	280	332	509	841	1139	1413	1470	1285	1245	990	740	434	10678
	Pueblo A	98/99-45/46	48	3	4	91	377	730	1042	1042	875	724	446	195	29	5558
Conn.	Hartford A	04/05-45/46	42	3	16	105	370	692	1065	1157	1062	859	524	213	47	6113
	New Haven A	98/99-45/46	48	3	11	88	341	658	1017	1109	1023	840	522	221	47	5880
D. C.	Washington	98/99-45/46	48	0	2	42	251	553	872	928	834	624	340	101	14	4561
Fla.	Apalachicola	13/14-45/46	33	0	0	1	23	154	300	323	252	159	38	2	0	1252
	Jacksonville	98/99-45/46	48	0	0	0	25	144	294	302	244	131	42	3	0	1185
	Key West	98/99-45/46	48	0	0	0	0	2	14	21	15	7	0	0	0	59
	Miami	11/12-45/46	35	0	0	0	0	15	41	53	45	28	3	0	0	185
	Pensacola	13/14-45/46	33	0	0	0	25	159	305	332	255	162	39	4	0	1281
	Tampa A	98/99-45/46	48	0	0	0	6	60	149	157	126	62	11	0	0	571
Ga.	Atlanta	98/99-45/46	48	0	0	12	128	392	644	660	563	382	169	33	2	2985
	Augusta	98/99-45/46	48	0	0	4	85	312	529	533	448	274	107	13	1	2306
	Macon	99/00-45/46	47	0	0	5	91	322	532	538	449	278	108	14	1	2338
	Savannah A	98/99-45/46	48	0	0	1	45	206	390	395	332	194	66	6	0	1635
	Thomasville	05/06-40/41	36	0	0	2	48	208	361	359	299	178	52	5	1	1513
Idaho	Boise A	98/99-45/46	48	9	17	136	385	717	1025	1077	840	688	440	252	92	5678
	Lewistown	00/01-32/33	33	5	9	107	378	688	932	992	779	603	371	193	52	5109
	Pocatello	98/99-45/46	48	12	21	176	475	821	1159	1224	1004	845	550	330	124	6741
Ill.	Cairo	98/99-45/46	48	0	0	26	181	493	823	878	748	512	232	60	4	3957
	Chicago	98/99-45/46	48	6	7	88	337	712	1116	1218	1080	861	531	259	67	6282
	Peoria A	05/06-45/46	41	4	8	88	350	730	1126	1231	1035	790	436	178	28	6004
	Springfield	98/99-45/46	48	0	3	65	286	664	1056	1151	977	719	377	132	16	5446
Ind.	Evansville A	98/99-45/46	48	0	1	35	211	544	888	948	822	582	288	85	6	4410
	Fort Wayne A	11/12-45/46	35	6	13	106	374	737	1107	1211	1052	864	504	217	41	6232
	Indianapolis	98/99-45/46	48	1	4	66	297	660	1032	1102	973	737	410	154	22	5458
	Royal Center	18/19-31/32	14	11	19	116	373	740	1104	1239	976	860	502	245	54	6239
	Terre Haute	12/13-45/46	34	0	3	62	270	627	993	1072	897	687	358	133	15	5117
Iowa	Charles City	04/05-45/46	42	8	24	164	480	906	1362	1535	1281	995	552	255	62	7624
	Davenport	98/99-45/46	48	2	6	91	344	748	1176	1291	1111	835	448	171	29	6252
	Des Moines	98/99-45/46	48	1	6	102	354	767	1204	1320	1132	843	446	171	29	6375
	Dubuque	98/99-45/46	48	3	12	123	402	808	1249	1380	1190	915	493	204	41	6820
	Keokuk	98/99-41/42	44	1	3	71	303	680	1077	1191	1025	761	397	136	18	5663
	Sioux City A	98/99-45/46	48	3	11	128	402	844	1273	1402	1206	909	485	202	40	6905
Kan.	Concordia	98/99-45/46	48	1	3	68	288	670	1060	1144	954	712	365	142	18	5425
	Dodge City A	98/99-45/46	48	1	3	59	275	641	998	1046	868	668	351	139	20	5069
	Iola	05/06-40/41	36	0	1	40	236	579	930	1026	817	599	282	98	8	4616
	Topeka	98/99-45/46	48	0	2	56	254	623	1013	1096	917	659	326	116	13	5075
	Wichita A	98/99-45/46	48	0	1	41	221	576	947	1016	836	604	290	103	9	4644
Ky.	Louisville	98/99-45/46	48	0	1	35	217	549	881	931	816	588	298	93	8	4417
	Lexington	98/99-40/41	43	1	3	48	258	601	916	964	862	650	352	123	14	4792
La.	New Orleans	98/99-45/46	48	0	0	0	23	145	304	323	247	129	31	1	0	1203
	Shreveport A	98/99-45/46	48	0	0	4	71	275	506	531	415	241	79	10	0	2132
Me.	Eastport	98/99-45/46	48	158	146	271	528	827	1224	1364	1238	1080	778	530	301	8445
	Greenville	07/08-40/41 42/43-45/46	38	69	113	315	643	1012	1464	1625	1443	1251	842	468	194	9439
	Portland A	98/99-45/46	48	28	48	182	466	794	1182	1309	1188	997	671	376	136	7377
Md.	Baltimore	98/99-45/46	48	0	1	33	227	526	855	921	837	637	343	95	12	4487

[a] Computed from daily temperatures recorded by United States Weather Bureau stations in cities over a varied number of seasons as indicated in the 3rd and 4th column of the table. Degree-day data for airport stations are not included in this table. The data for United States cities were computed by the United States Weather Bureau in 1946 and 1947 in accordance with the requirements of the National Joint Committee on Weather Statistics. The data for a number of the cities listed are based on readings taken at more than one official city weather station during the periods of analysis but the slight difference in the readings would not appreciably affect the resultant. Degree-days for cities in Canada and Newfoundland were supplied by the Canadian Meteorological Division, Department of Transport and were computed from mean temperature normals. *Indicates actual degree days for 1947.

[b] Letter A after station indicates city office and airport records combined.

TABLE 1. AVERAGE MONTHLY AND YEARLY DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND [a] (CONTINUED)

State	Station	Years	No. of Seasons	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Mass.	Boston A	98/99-45/46	48	7	15	98	338	647	1008	1108	1025	841	538	245	66	5936
	Fitchburg	98/99-40/41	43	12	29	144	432	774	1139	1240	1137	940	572	254	70	6743
	Nantucket	98/99-45/46	48	15	15	87	315	590	904	1010	967	866	619	366	121	5875
Mich.	Alpena	98/99-45/46	48	53	79	235	548	874	1238	1388	1321	1162	764	448	168	8278
	Detroit A	98/99-45/46	48	7	15	111	388	749	1124	1230	1134	927	566	253	56	6560
	Escanaba	98/99-45/46	48	54	84	256	572	927	1329	1499	1382	1219	808	477	170	8777
	Grand Rapids	03/04-45/46	43	8	20	128	422	764	1136	1248	1143	944	569	263	57	6702
	Houghton	00/01-40/41 42/43-45/46	45	70	94	268	582	965	1355	1535	1421	1251	820	474	195	9030
	Lansing	10/11-45/46	36	18	36	167	467	818	1190	1306	1178	995	600	294	80	7149
	Ludington	12/13-40/41	29	41	55	182	472	794	1135	1271	1183	1056	698	418	153	7458
	Marquette	98/99-45/46	48	86	99	258	555	926	1306	1465	1349	1193	794	494	220	8745
	Sault Ste. Marie A	98/99-45/46	48	91	110	285	616	965	1379	1572	1470	1291	826	487	215	9307
Minn.	Duluth	98/99-45/46	48	80	97	292	631	1066	1539	1714	1497	1254	804	515	234	9723
	Minneapolis	98/99-45/46	48	8	23	167	481	942	1415	1587	1372	1072	577	260	62	7966
	Moorhead	98/99-40/41	43	20	47	240	607	1105	1609	1815	1555	1225	679	327	98	9327
	St. Paul	98/99-32/33 37/38-40/41	39	11	24	169	488	942	1412	1589	1371	1078	573	258	60	7975
Miss.	Corinth	09/10-40/41	32	0	1	13	142	418	669	696	570	396	149	32	1	3087
	Meridian	00/01-45/46	46	0	0	6	99	322	525	539	440	274	107	17	1	2330
	Vicksburg	98/99-45/46	48	0	0	5	76	267	483	503	407	236	82	10	0	2069
Mo.	Columbia	98/99-45/46	48	0	3	62	266	621	1000	1076	916	655	337	120	14	5070
	Hannibal	98/99-40/41	43	1	3	66	288	652	1037	1139	980	710	374	128	15	5393
	Kansas City A	98/99-45/46	48	0	2	51	239	598	993	1077	909	651	322	108	12	4962
	Saint Louis	98/99-45/46	48	0	1	38	215	558	925	998	855	607	300	91	8	4596
	Springfield A	98/99-45/46	48	1	2	48	232	561	908	971	827	596	302	109	12	4569
Mont.	Billings A	09/10-45/46	37	14	31	223	530	889	1215	1310	1102	923	555	315	106	7213
	Havre	98/99-45/46	48	27	54	275	592	1012	1376	1532	1358	1102	614	341	133	8416
	Helena A	98/99-45/46	48	43	66	291	596	944	1252	1347	1157	990	639	413	192	7930
	Kalispell	99/00-45/46	47	66	102	332	636	968	1235	1339	1135	956	630	413	220	8032
	Miles City A	98/99-45/46	48	7	20	188	510	918	1322	1461	1267	997	545	275	81	7591
	Missoula	92/93-45/46	54	37	56	275	606	951	1235	1331	1072	903	590	369	179	7604
Neb.	Drexel	15/16-25/26	11	4	6	95	405	788	1271	1353	1096	843	493	219	38	6611
	Lincoln	98/99-45/46	48	1	5	85	325	732	1144	1242	1056	792	407	166	25	5980
	North Platte	98/99-45/46	48	4	9	131	410	799	1163	1227	1039	846	480	227	49	6384
	Omaha A	98/99-45/46	48	1	4	84	324	744	1169	1280	1088	810	410	157	24	6095
	Valentine	98/99-45/46	48	8	19	167	479	877	1246	1349	1166	962	563	287	74	7197
Nev.	Reno A	05/06-45/46	41	8	18	140	407	697	959	1007	791	702	498	301	93	5621
	Tonopah	14/15-40/41	27	5	7	105	388	713	1010	1075	870	749	522	286	82	5812
	Winnemucca	98/99-45/46	48	10	23	188	494	801	1090	1128	884	768	536	321	114	6357
N. H.	Concord A	03/04-45/46	43	18	49	189	497	823	1228	1345	1213	993	637	308	100	7400
N. J.	Atlantic City	98/99-45/46	48	1	2	39	247	546	867	946	887	750	485	208	37	5015
	Cape May	98/99-31/32	34	1	2	38	221	527	852	936	876	737	459	188	33	4870
	Newark A	98/99-23/24 35/36-40/41	32	1	6	65	295	635	980	1083	1002	794	448	162	29	5500
	Sandy Hook	15/16-40/41	26	1	2	40	268	579	921	1016	973	833	499	206	31	5369
	Trenton	14/15-45/46	32	1	6	63	301	604	957	1033	923	748	441	154	25	5256
N. M.	Albuquerque A	19/20-45/46	27	0	0	27	258	646	913	955	708	592	322	91	5	4517
	Roswell	05/06-45/46	41	0	0	26	191	512	781	773	585	459	199	50	2	3578
	Sante Fe	98/99-45/46	43	12	15	129	451	772	1071	1094	892	786	544	297	60	6123
N. Y.	Albany	98/99-45/46	48	4	15	117	411	753	1143	1271	1169	948	551	220	46	6648
	Binghamton	98/99-45/46	48	15	37	148	448	767	1137	1236	1155	950	584	267	74	6818
	Buffalo A	98/99-45/46	48	15	24	126	413	745	1110	1226	1165	995	668	348	90	6925
	Canton	06/07-45/46	40	27	61	219	550	898	1368	1516	1385	1139	695	340	107	8305
	Ithaca	99/00-42/43	44	17	40	156	451	770	1129	1236	1156	978	606	292	83	6914
	New York	98/99-45/46	48	1	4	50	272	594	940	1028	953	771	465	172	30	5280
	Oswego	98/99-45/46	48	20	33	147	440	762	1151	1275	1188	1015	665	366	124	7186
	Rochester A	98/99-45/46	48	10	26	132	423	751	1123	1227	1155	967	605	282	71	6772
	Syracuse A	03/04-45/46	43	13	32	146	437	760	1147	1255	1167	972	611	283	76	6899
N.C.	Ashville	02/03-45/46	44	2	3	49	279	565	800	817	719	558	315	114	15	4236
	Charlotte	98/99-45/46	48	0	1	17	148	420	684	700	600	413	198	39	4	3224
	Hatteras	98/99-45/46	48	0	0	1	61	273	500	570	530	391	193	34	1	2554
	Manteo	04/05-28/29	25	0	0	7	113	358	595	642	594	469	249	75	7	3109
	Raleigh	98/99-45/46	48	0	1	17	153	415	681	702	615	429	210	46	6	3275
	Wilmington	98/99-45/46	48	0	0	5	90	306	520	531	479	320	144	24	1	2420
N. D.	Bismarck	98/99-45/46	48	21	44	244	595	1057	1520	1704	1479	1181	643	340	109	8937
	Devils Lake	04/05-45/46	42	42	76	295	687	1186	1676	1906	1615	1313	752	411	145	10104
	Grand Forks	12/13-40/41 42/43-45/46	33	32	60	274	663	1160	1681	1895	1608	1298	718	359	123	9871
	Williston	98/99-45/46	48	28	61	285	637	1104	1545	1733	1513	1226	671	368	130	9301
Ohio	Cincinnati	98/99-45/46	48	1	3	53	273	611	960	1008	898	668	369	130	16	4990
	Cleveland	98/99-45/46	48	7	14	93	354	684	1045	1143	1067	876	553	252	56	6144
	Columbus	98/99-45/46	48	2	6	68	314	670	1019	1081	975	749	432	165	25	5506
	Dayton	11/12-42/43 45/46	33	2	6	71	309	660	992	1079	939	752	416	163	23	5412
	Sandusky	98/99-45/46	48	3	8	82	347	695	1067	1155	1066	862	536	232	42	6095
	Toledo	98/99-45/46	48	5	13	100	370	718	1097	1189	1083	887	533	227	47	6269
Okla.	Broken Arrow	18/19-30/31	13	0	0	28	169	513	803	881	646	506	212	61	5	3826
	Oklahoma City	98/99-45/46	48	0	0	22	153	455	792	846	684	457	200	58	3	3670
Ore.	Baker	98/99-45/46	48	54	72	259	534	848	1161	1222	984	827	601	418	217	7197
	Medford A	11/12-40/41 45/46	31	7	10	99	345	632	837	844	636	556	387	223	74	4650
	Portland	98/99-45/46	48	27	28	106	292	538	725	775	616	533	368	237	108	4353
	Roseburg	98/99-45/46	48	23	26	116	316	541	714	730	582	530	386	257	111	4332

Table 1. Average Monthly and Yearly Degree-Days for Cities in the United States, Canada and Newfoundland [a] (Concluded)

State	Station	Years	No. of Seasons	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Pa.	Erie	98/99-45/46	48	8	17	101	367	692	1049	1159	1105	922	591	284	68	6363
	Harrisburg A	98/99-45/46	48	1	6	67	314	637	990	1073	973	757	425	146	23	5412
	Philadelphia	98/99-45/46	48	0	2	36	235	544	884	962	881	685	378	115	17	4739
	Pittsburgh	98/99-45/46	48	3	7	69	322	651	982	1042	964	751	444	166	29	5430
	Reading	12/13-45/46	34	1	5	69	301	606	957	1038	929	736	423	144	23	5232
	Scranton	00/01-45/46	46	6	22	117	400	717	1074	1162	1071	862	523	213	51	6218
R. I.	Block Island	98/99-45/46	48	10	11	79	313	596	919	1030	984	875	618	354	108	5897
	Narragansett Pier	98/99-17/18	20	1	26	121	366	691	1012	1113	1074	916	622	342	113	6397
	Providence	04/05-45/46	42	6	16	101	358	668	1020	1106	1027	847	538	237	60	5984
S. C.	Charleston	98/99-45/46	48	0	0	1	47	225	428	452	384	239	83	7	0	1866
	Columbia	98/99-45/46	48	0	0	6	95	327	560	568	482	305	126	18	1	2488
	Due West	21/22-31/32	11	0	0	9	142	393	594	651	491	411	158	39	2	2890
	Greenville A	17/18-45/46	29	0	1	13	127	410	650	684	551	403	179	40	1	3059
S. D.	Huron	98/99-45/46	48	10	20	159	502	962	1409	1572	1353	1039	573	271	70	7940
	Pierre	98/99-40/41 42/43-45/46	47	4	11	136	438	887	1317	1460	1253	971	516	238	52	7283
	Rapid City	98/99-45/46	48	15	28	192	495	842	1178	1280	1140	981	598	339	109	7197
Tenn.	Chattanooga	98/99-45/46	48	0	0	13	150	432	691	711	604	412	185	39	1	3238
	Knoxville A	98/99-45/46	48	0	0	20	189	498	756	774	666	470	226	56	3	3658
	Memphis A	98/99-45/46	48	0	0	14	126	387	670	716	600	386	157	33	1	3090
	Nashville	98/99-45/46	48	0	0	20	170	469	748	788	675	467	218	55	3	3613
Texas	Abilene A	98/99-45/46	48	0	0	10	96	332	603	619	483	296	110	23	1	2573
	Amarillo A	98/99-45/46	48	1	2	42	221	548	854	861	719	546	284	107	11	4196
	Austin A	26/27-45/46	20	0	0	2	31	227	410	458	315	185	46	5	0	1679
	Brownsville A	08/09-45/46	38	0	0	0	8	65	176	191	111	65	11	1	0	628
	Corpus Christi	98/99-45/46	48	0	0	0	11	102	255	282	204	93	17	1	0	965
	Dallas A	13/14-45/46	33	0	0	6	70	293	574	600	437	281	91	15	0	2367
	Del Rio	05/06-45/46	41	0	0	2	35	203	413	413	262	139	31	3	0	1501
	El Paso A	98/99-45/46	48	0	0	6	88	366	615	615	432	291	104	14	1	2532
	Fort Worth A	98/99-45/46	48	0	0	5	79	285	553	586	463	270	97	16	1	2355
	Galveston	98/99-45/46	48	0	0	0	14	123	290	334	255	130	27	1	0	1174
	Houston	09/10-45/46	37	0	0	1	27	160	331	361	247	150	36	2	0	1315
	Palestine	98/99-45/46	48	0	0	4	67	261	496	512	401	236	80	11	0	2068
	Point Arthur	17/18-45/46	29	0	0	1	27	177	328	375	254	151	37	2	0	1352
	San Antonio A	98/99-45/46	48	0	0	1	31	171	366	390	287	148	37	4	0	1435
	Taylor	01/02-40/41	40	0	0	2	56	234	462	494	375	214	64	8	0	1909
Utah	Modena	00/01-45/46	46	6	11	156	499	832	1142	1190	944	816	567	338	97	6598
	Salt Lake City	98/99-45/46	48	3	5	98	371	712	1033	1093	871	716	446	236	66	5650
Vt.	Burlington	06/07-45/46	40	23	51	209	530	870	1313	1467	1338	1111	694	339	106	8051
	Northfield	98/99-42/43	45	62	112	283	602	947	1389	1524	1384	1176	754	405	166	8804
Va.	Cape Henry	98/99-45/46	48	0	0	7	125	398	676	731	682	526	301	86	6	3538
	Lynchburg A	98/99-45/46	48	1	2	37	230	521	799	829	732	537	287	81	12	4068
	Norfolk	98/99-45/46	48	0	0	9	129	392	668	712	650	483	254	62	5	3364
	Richmond	98/99-45/46	48	0	1	27	196	486	780	814	722	538	278	72	8	3922
	Wytheville	02/03-40/41	39	7	13	82	352	662	916	945	836	677	410	168	35	5103
Wash.	North Head	02/03-45/46	44	251	229	255	350	491	642	697	597	610	505	428	312	5367
	Seattle	98/99-45/46	48	67	69	170	365	554	704	759	637	595	436	299	160	4815
	Spokane A	98/99-45/46	48	20	37	184	480	817	1061	1139	931	756	490	285	118	6318
	Tacoma	98/99-45/46	48	71	75	190	390	581	737	786	658	612	455	313	171	5039
	Tatoosh Island	98/99-45/46	48	301	295	325	421	534	654	716	627	643	537	454	350	5857
	Walla Walla	98/99-45/46	48	5	10	90	315	662	910	981	770	571	354	186	56	4910
	Yakima	09/10-45/46	37	8	17	124	395	778	1050	1125	837	624	374	193	60	5585
W. Va.	Elkins A	98/99-45/46	48	15	23	115	403	722	1003	1033	947	763	489	229	58	5800
	Parkersburg	98/99-45/46	48	1	3	56	286	617	930	977	882	660	369	129	18	4928
Wis.	Green Bay	98/99-40/41	48	17	38	179	494	889	1329	1493	1329	1087	658	327	91	7931
	La Crosse	98/99-45/46	48	7	22	157	454	864	1339	1492	1281	990	531	232	52	7421
	Madison	04/05-45/46	42	8	20	145	452	857	1296	1451	1246	1002	588	274	66	7405
	Milwaukee	98/99-45/46	48	13	17	124	411	786	1203	1329	1177	959	617	341	102	7079
	Wausau	15/16-40/41	26	26	58	216	568	982	1427	1594	1381	1147	680	315	100	8494
Wyo.	Cheyenne A	98/99-45/46	48	40	46	251	587	876	1144	1187	1064	996	720	460	165	7536
	Lander	98/99-45/46	48	27	43	265	623	1021	1400	1427	1197	1006	669	410	155	8243
	Yellowstone Park	04/05-40/41	37	125	173	424	759	1079	1386	1464	1252	1165	841	603	334	9605
Alta.	Calgary	—	—	108	167	432	722	1122	1426	1609	1355	1215	750	480	264	9,650
	Edmonton	—	—	70	167	441	750	1224	1593	1810	1504	1299	777	428	222	10,285
B. C.	Vancouver	—	—	43	65	234	459	657	818	893	736	682	498	326	162	5,573
	Victoria	—	—	155	158	264	446	462	738	815	689	651	504	366	237	5,485
	Prince Rupert	—	—	282	229	342	546	702	893	933	804	809	645	521	357	7,063
Man.	Churchill	—	—	350	391	696	1187	1773	2356	2604	2288	2204	1530	1097	672	17,148
	Winnipeg	—	—	27*	34	330	744	1302	1829	2111	1775	1531	822	397	78	10,980
N. B.	Moncton	—	—	8*	56	282	595	936	1373	1528	1392	1190	798	474	180	8,812
	Saint John	—	—	124	112	270	558	870	1271	1417	1266	1132	792	505	261	8,578
N. S.	Halifax	—	—	12	12	189	493	783	1141	1280	1168	1065	768	493	210	7,614
Ont.	Fort William	—	—	62	155	354	722	1143	1596	1807	1582	1386	888	567	234	10,496
	Hamilton	—	—	3*	9*	108	471	816	1178	1305	1187	1063	651	322	6	7,119
	London	—	—	34*	11*	126	508	843	1200	1336	1240	1073	642	307	105	7,425
	Ottawa	—	—	29*	22*	204	595	984	1494	1646	1459	1256	726	310	91	8,816
	Toronto	—	—	16	35	168	504	817	1155	1304	1206	1072	669	341	87	7,374
	Windsor	—	—	19*	5*	54	425	795	1172	1283	1148	995	582	251	73	6,802
P.E.I.	Charlottetown	—	—	5*	57*	222	539	858	1246	1463	1338	1203	858	536	213	8,538
P. Q.	Montreal	—	—	7*	20*	180	561	948	1407	1587	1392	1209	702	298	88	8,399
	Quebec	—	—	17*	43	276	651	1050	1534	1696	1481	1311	849	428	102	9,438
Sask.	Saskatoon	—	—	16*	68	396	781	1305	1767	2027	1635	1445	725	397	138	10,700
Y. T.	Dawson	—	—	167	322	687	1209	1908	2440	2666	2159	1879	1092	580	246	15,355

Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the unit fuel consumptions per degree-day and are obtained as a result of the collection of operating information. Certain of this information is presented later, but before referring to these data attention is directed to the nature of the unit.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel or steam per degree-day per square foot of radiation, cubic foot of heated building space, or thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per (degree-day) (square foot of floor area). In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation, and some discrimination and judgment. If the volume basis is used, the net heated space is preferable to the gross building cubage since gross cubage includes outer walls and certain portions of attic and basement space which are usually unheated. In the absence of data on net heated volume a figure of 80 per cent of the gross volume may be used to obtain the estimated net heated volume. The volume basis has been rather widely used primarily because of its facility in application. In industrial buildings it is usually easier to obtain the correct volume of a given building than to measure and evaluate the heating capacity of its heating system or calculate its maximum hourly Btu loss. The comparison of buildings on a straight volume basis does not allow for variation in exposure, type of construction, ratio of exposed area to cubical contents, and type of occupancy. It is considered inaccurate for purposes of estimating fuel consumption unless the buildings are of very similar nature.

The calculated heat loss or its equivalent square feet of calculated radiator surface may be used as the unit. The use of the unit equivalent direct radiation is of questionable value when referring to heat transfer surfaces used in warm air furnace or central air conditioning systems. Where steam or hot water radiation is already installed, care should be exercised in using the unit equivalent direct radiation basis for estimating, since actual installed radiation may differ considerably from the exact radiation requirements. In view of all these considerations it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable, although the one most widely used seems to be *units of fuel per degree-day per square foot of equivalent direct radiator surface.* The equivalent heating load for the hot water supply is not included in the latter unit, but it generally includes the piping load.

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 2 for various gas heating values, and different types and sizes of heating plants. They are based on an inside design temperature of 70 F and an outside design temperature of 0 F, and apply only to these condi-

TABLE 2. UNIT FUEL CONSUMPTION CONSTANTS (U) FOR GAS[a]
Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 60 F.

Heating Value of Gas Btu per Cu Ft	Hot Water			Steam			Warm Air	
	Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 300 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0.231	0.220	0.855	0.820
535	0.132	0.126	0.120	0.226	0.215	0.206	0.800	0.766
800	0.089	0.085	0.081	0.151	0.144	0.137	0.534	0.513
1000	0.071	0.068	0.065	0.121	0.115	0.110	0.428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000708	0.000675	0.000642	0.00121	0.00115	0.00110	0.00428	0.00409

[a] Abstracted from Comfort Heating, *American Gas Association*, 1938.

tions. For other outside design conditions corrections must be made as given in Table 5.

The factors in Table 2, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten per cent addition or deduction in these cases is recommended by *A.G.A.* publications. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

For gas heating values other than those given in Table 2, simply interpolate or extrapolate. It will also be noted that Table 2 applies only to small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 5. Estimate the gas required to heat a building located in Chicago, Ill., which has 6282 degree-days and a gas heating value of 800 Btu per cu ft. The calcu-

TABLE 3. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR OIL[b]
Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F

Unit[c]	Efficiency in Per Cent				
	40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator	0.00172	0.00137	0.00114	0.00098	0.00086
Gal Oil per Sq Ft Hot Water Radiator	0.00108	0.00086	0.00072	0.00062	0.00054
Gal Oil per 1000 Btu per Hour Heat Loss	0.00715	0.00571	0.00476	0.00409	0.00358

[a] Based on a heating value of 140,000 Btu per gallon.
[b] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.
[c] Per degree-day.

TABLE 4. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR COAL[b]
Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F.

UNITS[c]	EFFICIENCY IN PER CENT				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator..	0.0200	0.0160	0.0133	0.0114	0.0100
Lb Coal per Sq Ft Hot Water Radiator....	0.0125	0.0100	0.0084	0.0072	0.0063
Lb Coal per 1000 Btu per Hour Heat Loss....	0.0825	0.0666	0.0550	0.0471	0.0412

[a] Based on a heating value of 12,000 Btu per pound.
[b] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H.B Hotchkiss.
[c] Per degree-day.

lated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of −10 F and 70 F.

Solution. From Table 2, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.085 cu ft of gas per (degree-day) (square foot of hot water radiation). From Table 5, the correction factor is 0.875 for −10 F outside design temperature, hence 0.875 × 0.085 = 0.07438. By Equation 4,

$$F = 0.07438 \times 1000 \times 6282 = 467{,}255 \text{ cu ft.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 2, are given in Table 3. The factors in Table 3 apply only to an inside design temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 3 must be multiplied by the values in Table 5 as explained under Estimating Gas Consumption.

Values given in Table 3 assume the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in Table 3 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 6. Estimate the seasonal oil consumption of an oil-fired boiler in a building located in Minneapolis having a calculated heat loss of 192,000 Btu per hr, burning 144,000 Btu per gal oil and operating at a seasonal efficiency of 60 per cent. The outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F.

Solution. From Table 3, under 60 per cent efficiency and opposite the bottom column, the value of U is found to be 0.00476 gal per 1000 Btu hourly heat loss for 0 F outside temperature.

TABLE 5. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES[a]

OUTSIDE DESIGN TEMP. F DEG.....	−20	−10	0	+10	20
Correction Factor..............	0.778	0.875	1.000	1.167	1.400

[a] The multipliers in Table 5, which are high for mild climates and low for cold regions, are not in error as might appear The unit figures in Tables 2, 3, and 4 are per square foot of radiator or thousand Btu heat loss per degree-day. For equivalent buildings and heating seasons, those in warm climates have lower design heat losses and smaller radiator quantities than those in cold cities. Consequently, the *unit* figure in quantity of fuel per (square foot of radiator) (degree-day), is larger for warm localities than for colder regions. Since the northern cities have more radiator surface per given building and a higher seasonal degree-day total than cities in the south, the total fuel per season will be larger for the northern city.

The correction factor for -20 F outside design temperature from Table 5 is 0.778. Solving, $0.778 \times 0.00476 = 0.00370$. Making a further correction for the heating value:

$0.0037 \times \frac{140,000}{144,000} = 0.0036$ gal per 1000 Btu per hr calculated heat loss per degree-day.

From Table 1, the normal degree-days for Minneapolis are 7966. Since U is expressed in 1000 Btu, N is equal to 192. Substituting in Equation 4:

$$F = 0.0036 \times 7966 \times 192 = 5506 \text{ gal.}$$

Estimating Coal or Coke Consumption

Coal or coke consumption estimates are made by following exactly the same procedure as for oil. Values of U are given in Table 4 which only apply to an inside design temperature of 70 F and an outside design temperature of 0 F. A correction must be made for other conditions by use of the multiplying factors in Table 5. Data in Table 4 are based on 12,000 Btu per lb coal, and for other heating values of coal they must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

Example 7. A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hr. Based on an inside design temperature of 70 F and an outside design temperature of -20 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per lb fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

Solution. From Table 4, U is 0.0666 lb of coal per 1000 Btu per hr heat loss. Correcting for the outside design temperature of -20 F from Table 5, the value of U is $0.778 \times 0.0666 = 0.0518$. From Table 1, D is 8745 and from the problem, N is 240.

Substituting in Equation 4:

$$F = 0.0518 \times 240 \times 8745 = 108,718 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 1, the average number of degree-days for November, December, and January in Marquette are 926, 1306, and 1465, a total of 3697. The yearly total is 8745, so that during these three months the estimated consumption is:

$$\frac{3697}{8745} \times 108,718 = 45,961 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is generally assumed at 100 per cent. If for low pressure steam an average heating value of 1000 Btu per pound of steam is used, no correction is necessary. In comparing values from different cities, correction should be made for design temperature (see Table 5) when the unit figures are in terms of square feet of radiation but not when the values are in terms of building volume or floor space.

Where the heat loss is calculated in Btu per (hour) (*degree difference in temperature*) the simple Equation 5 may be used:

(5)

where

F = pounds of steam required for estimate period.
H = calculated heat loss, Btu per (hour) (degree difference).
D = number of degree days for the period of estimation.
1000 = Btu delivered per pound of steam condensed.

In this method the number of degree-days automatically takes care of average inside and outside temperature difference. When degree days are taken from Table 1, an average inside temperature of approximately 65 F is assumed throughout the period. If an average inside temperature other than approximately 65 F is to be used, the number of degree-days should be obtained for the new base.

Example 8. An eight-story building in Pittsburgh maintains daytime temperatures of 70 F but allows night temperature to drop to not lower than 60 F. Its calculated heat loss is 10,500 Btu per (hr) (*degree temperature difference*). What is the estimated average yearly steam consumption for building heating?

Solution. Since the average inside temperature is approximately 65 F, the degree-days from Table 1, based on 65 F may be used. Therefore, from Table 1, Pittsburgh has 5430 degree-days per normal season. Inserting in Equation 5

$$F = \frac{10{,}500 \times 24 \times 5435}{1000} = 1{,}368{,}360 \text{ lb of steam.}$$

TABLE 6. STEAM CONSUMPTION OF BUILDINGS WITH VARIOUS TYPES OF OCCUPANCY[a]

TYPE OF BUILDING	NO. BLDGS.	AVERAGE VOLUME HEATED SPACE 1000 CU FT	STEAM FOR HEATING	AVERAGE HOURS OF OCCUPANCY
			Lb per DD per 1000 Cu Ft	
Office	334	2160	0.685	12.1
Office and Bank	49	3000	0.577	13.1
Office and Printing	8	1895	1.230	17.7
Office and Theater	7	4950	0.412	12.9
Office and Stores or Shops	26	1615	0.617	13.2
Bank	16	806	0.786	11.7
Department Store	63	3400	0.385	11.1
Stores	73	310	0.624	10.4
Loft	63	865	0.588	10.0
Warehouse	24	2230	0.459	9.4
Hotel and Club	73	1795	0.990	22.3
Apartment or Residence	51	1425	0.962	21.8
Theater	22	1240	0.482	12.9
Garage	13	1540	0.202	21.4
Manufacturing	19	1350	0.808	9.5
Church	9	656	0.532	7.9
Hospital	4	3306	1.194	22.0
School	8	1115	0.592	11.5
Municipal or Federal	15	3215	0.587	15.6
Lodge, Gym, Hall or Auditorium	12	880	0.390	12.4
Miscellaneous	7	1387	0.479	21.4

[a] Principles of Economical Heating, *National Association of Building Owners and Managers.*

Consideration has been given to the difference in steam utilization of different types of buildings and Table 6 shows actual average units for these various types. These figures were obtained from operating results in 896 buildings located in all sections of the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating hot water is not included in the values given in Table 6.

Example 9. A store in Philadelphia with a heating system designed to maintain 70 F inside in 0 F weather has 250,000 cu ft of heated space. What would be the estimated average yearly steam consumption of purchased steam for heating?

Solution. According to Table 6, a store would use 0.624 lb of steam per degree-day per 1000 cu ft heated space. From Table 1, Philadelphia has 4739 degree-days per normal year. Inserting in Equation 4:

$$F = 0.624 \times 250 \times 4739 = 739{,}284 \text{ lb of steam.}$$

Degree-Day as an Operating Unit

The degree-day is also widely used as a means of comparing the efficiency of the fuel consumption of one period with another *for the same building*. Since the fuel consumption is proportional to the weather (degree-days) and since the periods to be compared may not have the same weather conditions, the comparison can be made only after the fuel consumptions have been computed on a comparable weather basis, that is, upon the actual number of degree-days occurring for a given month and year in the city under consideration. Since fuel consumption is proportional to the number of degree-days, plant operators frequently compute each month the fuel burned per degree-day by the heating plant. The resulting unit figure, by eliminating the outside temperature variable, indicates whether the operating efficiency of the plant is above or below the previous month or year.

The figures in Table 7 illustrate a typical example of a method of using the degree-day for making heating comparisons for one building for two consecutive heating seasons. The heat quantity figures inserted are pounds of steam, but a similar comparison could be made using pounds of coal, gallons of oil, or cubic feet of gas.

For such a comparison, a two-year record is often used, as shown in Table 7. The year under consideration may then be compared, month by month, with the previous year. Column 3, Consumption for Heating, would be used if the same fuel is used for heating and process steam. Some reasonable figure must be assumed for the process requirement and should be deducted from the amount shown in column 2. This would leave in column

Table 7. Heat Consumption Record for Comparison

Heating Season	Col. 1	Col. 2 Total Consumption	Col. 3 Consumption for Heating	Col. 4 Avg Mean Temp.	Col. 5 Deg Days 65 F Base	Col. 6 Lb/Deg Day	Col. 7 Lb/Deg Day/ M Cu Ft
1942-43	Sept.	337,500	170,500	65	146	1,170	0.575
	Oct.	834,200	667,200	53	339	1,966	0.970
	Nov.	1,446,600	1,279,600	44	641	1,990	0.982
	Dec.	2,176,400	2,009,400	25	1,233	1,630	0.804
	Jan.	2,332,200	2,165,200	22	1,297	1,670	0.822
	Feb.	2,131,100	1,964,100	28	1,106	1,775	0.888
	Mar.	2,021,900	1,854,900	31	1,032	1,799	0.885
	Apr.	1,241,500	1,074,500	43	647	1,660	0.818
	May	672,500	505,500	55	303	1,670	0.822
	June	258,600	91,600	....	50	1,830	0.905
	July	188,400	....	....	....	....	....
	Aug.	180,100	....	....	....	....	....
	Total	13,821,000	....	....	....	....	....
1943-44	Sept.	330,200	146,200	61	167	875	0.431
	Oct.	887,100	703,100	52	410	1,718	0.845
	Nov.	1,525,200	1,341,200	39	812	1,653	0.815
	Dec.	2,045,500	1,861,500	28	1,120	1,660	0.817
	Jan.	1,933,400	1,749,400	30	1,044	1,670	0.825
	Feb.	1,990,200	1,806,200	30	1,111	1,624	0.800
	Mar.	1,984,100	1,800,100	31	1,021	1,760	0.868

If, for example, the heat consumption in March, 1943, is compared with that in March, 1944, it will be found that in the latter the steam consumption is 1799 − 1760 = 39 lb less which is a decrease of 2.2 per cent.

TABLE 8. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS

Building Classification	Load Factor	Lb of Demand per (Hour) (Sq Ft of Equivalent Installed Radiator Surface)
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

3 only the fuel chargeable to heating. The degree-day values in column 5 are obtainable from the local Weather Bureau. Figures in column 6 are obtained by dividing corresponding values in column 3 by the degree days in column 5. The heating index in column 6 is, then, a figure of heat consumption, corrected for outdoor temperature, and should be relatively constant month by month. Column 7 in Table 7 may be used if the heat consumption is to be compared on a building volume basis with average values shown in Table 6.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 8[5].

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization. Thus, in Table 8, the theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

SEASONAL EFFICIENCY

The task of predicting fuel consumption within reasonably accurate limits is a simple one where sufficient experience data are available for the fuel in question. Such data can be analyzed to the point where average unit factors can be determined and expressed in such terms as, for example, cubic feet of gas actually burned per (square foot of calculated steam radiator surface) (degree-day). The unit U can be inserted directly in Equation 4 without reference to efficiency. Such experience factors are available for gas (see Table 2) and for district steam (Table 6), but not for coal or oil.

Since values of U are not available for oil or coal, an assumed seasonal efficiency E must be used. Selection of a value for this E must be made with caution, for its use implies a meaning not commonly associated with the word efficiency and consequently is frequently misleading.

The input of heat to a building consists not only of the energy in the fuel but that from occupants, the sun, appliances, processes, and all other sources. In many cases these make up, over a period, an important percentage of the total heat required, and if they are not taken into account a calculation of *efficiency* can show a figure over 100 per cent.

For this and other reasons the actual *seasonal efficiency* is a difficult thing to determine. Published data are widely scattered and insufficient. From the available published material it is found that the seasonal efficiency varies over a wide range, depending on the fuel used, and it varies widely even for a given fuel. For example, in a recent survey of 30 houses in one locality there was found a variation of from 45 to 75 per cent in the utilization *efficiency* depending on the fuel[6].

REFERENCES

[1] Graphical Method of Calculating Heat Losses, by Paul D. Close (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 345).

[2] Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems (11th edition), and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (*National Warm Air Heating and Air Condition Association*).

[3] See Industrial Gas Series, House Heating (*third edition*), published by the *American Gas Association.*

[4] Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

[5] The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 171).

[6] Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 185).

CHAPTER 21

GRAVITY WARM AIR SYSTEMS

Warm Air Leaders, Stacks, and Registers; Return Air Grilles, Ducts, and Connections; Outline of Design Procedure

WARM air heating systems of the *gravity* type are described in this chapter[1]. In these systems the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head.

A gravity warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard. A sectional view of a typical plant showing good installation practice is given in Fig. 1.

The air supply to the furnace is usually taken entirely from inside the building through one or more recirculating ducts, although in some cases an outside air supply duct is provided.

WARM AIR LEADERS, STACKS, AND REGISTERS

In a gravity circulating warm-air furnace system, the size of the leader pipe to a given room depends upon the length of the leader and the temperature of the warm air entering the room at the register. For most successful operation, the furnace should be centrally located with respect to register and stack positions so that the leaders will be of uniform length and as short as possible, in which case the frictional resistance to air flow and the temperature loss from the ducts will be about the same for all runs.

In the Standard Code for Installation of Gravity Warm Air Heating Systems, the design was originally based on the heat carrying capacities per square inch of leader pipe area with register air temperatures of 175 F. In a recent revision of the entire design procedure, as shown in the section entitled *Outline of Design Procedure*, the carrying capacities of leader pipes have been expressed directly in terms of Btu per hour.

In general it is advisable to use two or more leader pipes to rooms re-requiring more than the capacity of a 12 in. round pipe. The tops of all sizes of leader pipes should be cut into the furnace bonnet at the same elevation, and from this point there should be a uniform upgrade of at least 1 in. per foot of run. Leaders over 12 ft in length, or having a large number of elbow fittings should be avoided if possible. In cases where such leaders are necessary, it is recommended that smooth transition fittings be used, and that duct insulation be applied. Asbestos paper, unless of the corrugated type, should not be considered as insulation. To assist

in balancing the air distribution of the system, a damper should be placed in each leader pipe except one, this latter leader preferably being connected to a room heated at all times, such as a living room.

In a gravity circulating system, the ratio of stack to leader area is quite

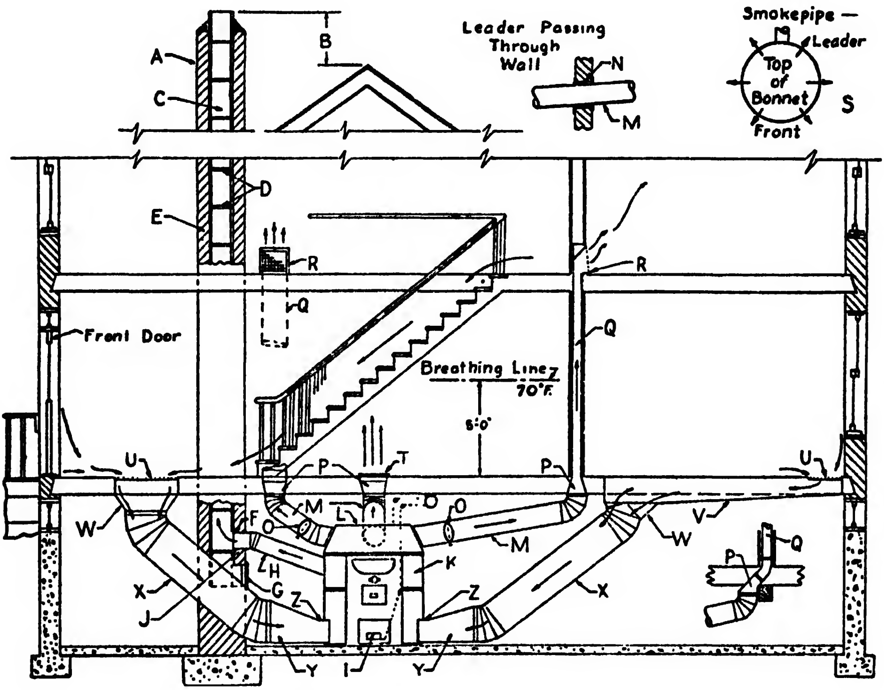


FIG. 1. A SECTIONAL VIEW OF A TYPICAL PLANT SHOWING GOOD INSTALLATION PRACTICE[a]

A. House chimney, no bends nor offsets.
B. Top of chimney at least 2 ft above ridge of roof.
C. Flue lining, fireclay.
D. All joints air tight.
E. At least 8 in. brick.
F. No other connection beside that to furnace.
G. Cleanout frame and door, airtight.
H. Smoke pipe, end flush with inner surface of flue.
I. Draft door.
J. Use flue thimble.
K. Casing body.
L. Casing hood or bonnet, top of all leader collars on same level.
M. Round leader, pitch 1 in. per foot.
N. Sleeve with air space around leader where passing through wall.
O. Dampers in all leaders, except one
P. Transition fittings.
Q. Rectangular wall stack.
R. Baseboard register.
S. Distribute pipes equally around bonnet.
T. Floor register.
U. Return air face.
V. Panning under joist.
W. Transition collar.
X. Round return pipe.
Y. Transition shoe.
Z. Top of shoe at casing not above grate level.

[a] From *N.W.A.H.&A.C.Assn.* Standard Code Application Manual.

important, although little is gained by providing wall stacks with areas in excess of 75 per cent of their connected leader pipe area. In most cases a 3½ in. × 12 in. stack is the largest which can be installed in normal wall construction. Hence, any room having a heat loss much in excess of 9000 Btu per hr, will require two or more stacks, or one oversized stack built into a 6 in. studding space, providing the design register temperature is to be retained at the value of 175 F as recommended.

Registers used for discharging warm air into rooms should have a net area not less than the area of the leader pipe to which the register is attached. First story registers should be connected through boot and register box extensions having areas at least equal to leader areas. Upper story registers should be of the same width as the wall stack, and should be placed either in the baseboard or sidewall, preferably without offsets. First story registers may be of the baseboard or floor type, with the former location preferred. High sidewall registers in gravity systems deliver more warm air into the room than do baseboard registers, but most of the additional air merely results in high temperatures at the ceiling.

RETURN AIR GRILLES, DUCTS, AND CONNECTIONS

The placement and number of return grilles will depend upon the size, details, and exposure of the house. Small compactly built houses may be adequately served by a single return grille effectively placed in the central hall. It is usually desirable to have two or more returns, provided that in two-story residences one return is placed to effectively receive the return air at the foot of the stairs. A return air connection must be carried to any room whose floor level is below that of adjacent rooms.

The return air grilles should have free areas at least equal to the ducts to which they connect and should be installed in the floor, or in the baseboard with the top edge of the grille not more than about 14 in. above the floor line. Frictional resistance in the return air system is as detrimental as is resistance in the warm-air system, so that care should be exercised in locating return air grilles which require long return ducts.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus, in rooms having only small windows the grilles can be brought as close to the furnace as possible, but if the room has large window exposure the grille should be located near the exposure. The frictional resistance of the long ducts used in parallel with short return ducts must be reduced to compensate for the length. Return ducts from upstairs rooms may be necessary in spaces which are closed off from the rest of the house or which have much outdoor exposure. Return grilles on different floor levels should not be connected to the same vertical return duct.

The ducts through which air is returned to the furnace should be designed to minimize resistance to air flow. They should be of ample area, in excess of the total area of warm-air pipes, and should be streamlined. Horizontal ducts should pitch at least ½ in. per foot downward toward the furnace, avoiding fittings which would require lifting of the return air after the duct has passed under some obstacle.

Ducts returning air to the furnace should avoid heat sources which tend to reheat the return air. If the duct must be run over the top of the furnace, or above the vent pipe from the furnace, insulation should be interposed between the heat source and the duct.

Circulation of air is facilitated if the air can slide down a pipe inclined at approximately 45 deg and into a furnace shoe connection having a cross-sectional area equal to that of the pipe. The top of the return shoe should enter the casing below the level of the grate in the case of a coal furnace, and not more than 14 in. above the floor in the case of oil or gas furnaces. In order to accomplish this the shoe is made wide.

OUTLINE OF DESIGN PROCEDURE

The data underlying the design procedure are given in detail in a circular[2] issued by the University of Illinois. In this procedure the design of the warm-air duct system is considered as an entire unit, so that for a given heat loss the sizes of leaders, stacks, boots, stackheads, and registers are all correlated. Similarly in the case of return ducts, the selection specifies a complete unit consisting of return grille, return duct, and shoe connection.

Recommended Standard Sizes

For the purpose of simplification and standardization, selected combinations of commercial sizes of warm air pipes, return air pipes, ducts, grilles, fittings, and registers are designated as Combination Numbers. The numbers assigned and the combinations selected as standard are listed in the following Tables 1 to 4 inclusive[3].

TABLE 1. FIRST STORY WARM-AIR DUCTS[a]

Combination No.	Leader Pipe Diameter, In.	Register Size, In.		
		Floor	Baseboard	
			Size	Extension
1	8	8 x 10	10 x 8	2¼
2	9	9 x 12	12 x 8	2¼
3	10	10 x 12	12 x 9	3¼
4	12	12 x 14	13 x 11	5¼
5	14	14 x 16	....	......

[a] When the calculations indicate a requirement for a given room greater than Combination No. 4, two or more smaller units totalling the required capacity are recommended.

TABLE 2. SECOND STORY WARM-AIR DUCTS—SINGLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack[b] Size In.	Register Size, In.			
			Floor	Baseboard		Sidewall
				Size	Extension	
11	8	10 x 3¼	8 x 10	10 x 8	2¼	10 x 8
12	9	12 x 3¼	9 x 12	12 x 8	2¼	12 x 8
14	10	14 x 3¼	10 x 12	12 x 8	2¼	12 x 8
15	12	12 x 5¼	....	12 x 9	3¼	....
16	12	14 x 5¼	....	13 x 11	5¼	....

[b] Recommended stack sizes. Tables may also be applied to 3 in. and 3½ in. stack depths.

TABLE 3. SECOND STORY WARM-AIR DUCTS—DOUBLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack Size, In.		Register Size, In.			
		Internal	External	Floor	Baseboard		Sidewall
					Size	Extension	
21	8	2½[c] x 10	3⅛[c] x 10⅝	8 x 10	10 x 8	2¼	10 x 8
22	8	3 x 10	3⅝ x 10⅝	8 x 10	10 x 8	2¼	10 x 8
23	9	2½[c] x 12	3⅛[c] x 12⅝	9 x 12	12 x 8	2¼	12 x 8
24	9	3 x 12	3⅝ x 12⅝	9 x 12	12 x 8	2¼	12 x 8

[c] Commercial sizes vary ⅛ in. from values shown.

TABLE 4. RETURN AIR DUCTS

Combination No.	Duct Dia. In.	Area at Shoe Connection, Sq In.	Metal Grille Sizes — Choose One: A	B	C	When Joist Lining is Used[d]: No. of Joists Lined	Minimum[e] Depth, In.	When Duct is Used — Choose One	
31	10			8 x 14	10 x 12	1	7	14 x 6	12 x 8
32	12		6 x 30	8 x 24	12 x 14	1	9	22 x 6	16 x 8
33	14	170	8 x 30	10 x 24	14 x 16	1	12	28 x 6	22 x 8
34	16	220	10 x 30	12 x 24		2	8	28 x 8	22 x 10
35	18	280	12 x 30	14 x 24		2	10	36 x 8	28 x 10
36	20	340	14 x 30	18 x 24		2	12.5	36 x 10	30 x 12
37	22	420	18 x 30			2	15.0	42 x 10	36 x 12
38	24	500	20 x 30			2	18.0	42 x 12	36 x 14

[d] Based on 14 in. space between joists.

[e] Use full depth of joist except when joist depth is less than minimum depth required, *when pan must be used.*

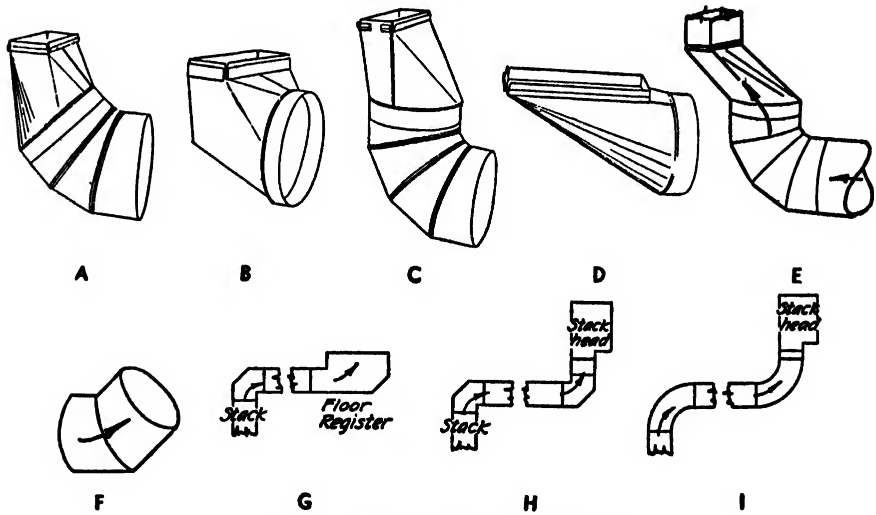


FIG. 2. TYPICAL WARM AIR BOOTS

TABLE 5. RESISTANCES OF WARM AIR BOOT COMBINATIONS EXPRESSED IN ELBOW EQUIVALENTS

Warm Air Boot	Name of Combination	Equivalent No. of 90-Deg Elbows
A	45-Deg Angle Boot and 45-Deg Elbow	1
B	90-Deg Angle Boot	1
C	Universal Boot and 90-Deg Elbow	1
D	End Boot	2
E	Offset Boot	2½
F	45-Deg Angle	½
G	Floor Register—Second Story	3
H	Offset	3
I	Offset	2

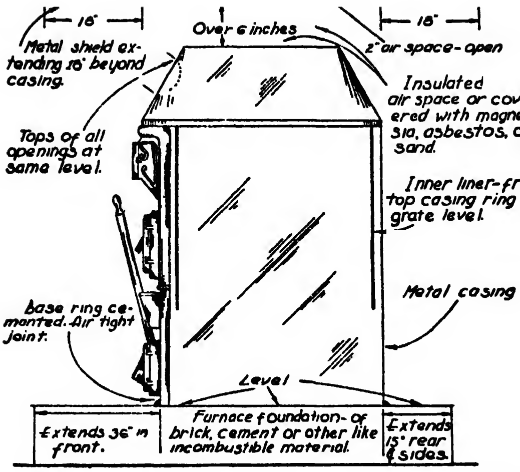


Fig. 3. Details of Furnace Bonnet, Casing, and Foundation (From Gravity Code and Manual)

The selected types of boots are shown in Fig. 2 and their resistances expressed in equivalent elbows are shown in Table 5. It is essential that free areas be maintained throughout fittings.

Figs. 3 and 4 show recommended practice as given in the *N.W.A.H. & A.C. Assn.* For construction, design features, and ratings of gravity furnaces see Chapter 18.

Carrying Capacity

The Btu carrying capacities of the selected warm air and return air combinations are shown in Tables 6, 7 and 8.

The selected types of return air ducts and fittings are shown in Fig. 5.

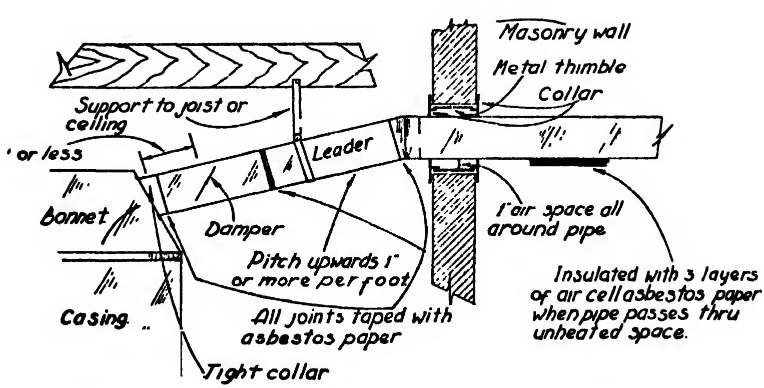


Fig. 4. Details of Bonnet and Leader of Gravity Warm-Air Furnace (From Gravity Code and Manual)

Table 6. Warm Air Carrying Capacity, Btu Delivered, First Story Registers[a]

Length of Leader Pipe—in Feet

Combination No.	No. of Elbows	4 Ft	6 Ft	8 Ft	10 Ft	12 Ft	14 Ft	16 Ft	18 Ft	20 Ft	22 Ft	24 Ft
1	1	6,020	5,850	5,680	5,510	5,340	5,170	5,000	4,830	4,660	4,490	4,320
2		7,620	7,400	7,180	6,970	6,760	6,540	6,320	6,110	5,890	5,680	5,460
3		9,400	9,140	8,870	8,600	8,340	8,070	7,810	7,540	7,270	7,010	6,740
4		13,350	12,970	12,590	12,210	11,830	11,450	11,080	10,700	10,320	9,950	9,560
5		17,520	17,020	16,530	16,040	15,550	15,050	14,550	14,050	13,560	13,060	12,560
1	2	5,850	5,660	5,490	5,330	5,160	5,000	4,840	4,670	4,510	4,340	4,180
2		7,360	7,150	6,940	6,730	6,520	6,320	6,110	5,910	5,700	5,500	5,290
3		9,090	8,840	8,580	8,320	8,060	7,800	7,550	7,290	7,040	6,780	6,520
4		12,910	12,540	12,170	11,800	11,430	11,060	10,690	10,320	9,950	9,580	9,210
5		16,940	16,450	15,990	15,500	15,040	14,550	14,080	13,600	13,120	12,650	12,150
1	3	5,620	5,460	5,310	5,150	4,990	4,830	4,670	4,510	4,350	4,190	4,030
2		7,120	6,910	6,710	6,510	6,310	6,110	5,900	5,700	5,500	5,300	5,100
3		8,780	8,530	8,280	8,030	7,780	7,530	7,290	7,040	6,800	6,550	6,300
4		12,450	12,100	11,750	11,400	11,050	10,700	10,350	10,000	9,650	9,300	8,950
5		16,360	15,900	15,440	14,970	14,510	14,050	13,600	13,130	12,660	12,200	11,750
1	4	5,420	5,260	5,110	4,960	4,800	4,650	4,500	4,350	4,190	4,040	3,890
2		6,860	6,660	6,460	6,270	6,080	5,890	5,690	5,500	5,300	5,110	4,910
3		8,460	8,200	7,980	7,740	7,500	7,260	7,020	6,780	6,550	6,310	6,070
4		12,010	11,670	11,330	10,990	10,650	10,310	9,970	9,630	9,290	8,950	8,610
5		15,770	15,320	14,880	14,420	13,990	13,540	13,100	12,650	12,200	11,750	11,310
1	5	5,240	5,090	4,940	4,790	4,640	4,500	4,350	4,200	4,050	3,910	3,760
2		6,630	6,440	6,250	6,060	5,880	5,690	5,500	5,320	5,130	4,940	4,750
3		8,180	7,950	7,720	7,490	7,260	7,030	6,800	6,560	6,330	6,100	5,860
4		11,610	11,290	10,950	10,620	10,300	9,970	9,640	9,320	8,990	8,660	8,320
5		15,250	14,800	14,380	13,950	13,520	13,090	12,650	12,230	11,800	11,370	10,940

[a] Additional values for 6 and 7 elbows are given in original Manual.

Table 7. Warm Air Carrying Capacity, Btu Delivered, Second Story Registers[a]

Length of Leader Pipe—in Feet

Combination No. [b,c]	No. of Elbows	4 Ft	6 Ft	8 Ft	10 Ft	12 Ft	14 Ft	16 Ft	18 Ft	20 Ft	22 Ft	24 Ft
11-22	1	8,370	8,140	7,900	7,670	7,430	7,190	6,950	6,710	6,470	6,240	6,000
12-24		10,040	9,760	9,470	9,190	8,900	8,620	8,330	8,050	7,770	7,480	7,200
14		11,710	11,380	11,050	10,720	10,390	10,060	9,720	9,390	9,060	8,730	8,400
15		16,200	15,750	15,300	14,840	14,380	13,920	13,460	13,000	12,550	12,100	11,640
16		18,920	18,390	17,850	17,310	16,780	16,240	15,710	15,180	14,640	14,100	13,570
11-22	2	7,940	7,720	7,500	7,280	7,050	6,830	6,600	6,370	6,150	5,930	5,700
12-24		9,540	9,270	9,000	8,730	8,460	8,190	7,920	7,650	7,380	7,110	6,840
14		11,120	10,810	10,500	10,180	9,870	9,550	9,230	8,920	8,610	8,290	7,980
15		15,400	14,970	14,530	14,100	13,670	13,230	12,800	12,360	11,930	11,500	11,070
16		17,980	17,470	16,960	16,450	15,950	15,430	14,830	14,420	13,910	13,400	12,890
11-22	3	7,530	7,320	7,110	6,900	6,680	6,470	6,250	6,040	5,830	5,620	5,400
12-24		9,030	8,780	8,520	8,270	8,010	7,750	7,500	7,240	6,990	6,730	6,470
14		10,530	10,240	9,940	9,650	9,350	9,050	8,750	8,450	8,160	7,860	7,560
15		14,580	14,180	13,780	13,370	12,950	12,530	12,120	11,710	11,300	10,890	10,480
16		17,040	16,550	16,070	15,580	15,110	14,620	14,140	13,660	13,180	12,700	12,210
11-22	4	7,120	6,920	6,720	6,520	6,310	6,110	5,900	5,700	5,500	5,300	5,100
12-24		8,530	8,290	8,050	7,810	7,570	7,330	7,080	6,840	6,600	6,360	6,120
14		9,950	9,670	9,390	9,110	8,830	8,550	8,260	7,980	7,700	7,420	7,140
15		13,780	13,390	13,000	12,610	12,220	11,830	11,440	11,050	10,670	10,280	9,890
16		16,080	15,620	15,170	14,710	14,260	13,810	13,350	12,900	12,440	11,980	11,530
11-22	5	6,700	6,510	6,320	6,130	5,940	5,750	5,560	5,370	5,180	4,990	4,800
12-24		8,040	7,810	7,580	7,350	7,130	6,900	6,670	6,440	6,220	5,990	5,760
14		9,370	9,110	8,850	8,580	8,310	8,050	7,780	7,510	7,250	6,980	6,720
15		12,970	12,600	12,240	11,870	11,500	11,140	10,770	10,400	10,010	9,680	9,310
16		15,140	14,710	14,280	13,850	13,420	13,000	12,570	12,140	11,710	11,280	10,850

[a] When floor registers are used, see Fig. 2.

[b] No. 21 for Btu values multiply 11-22 values by 0.83.

[c] No. 23 for Btu values multiply 12-24 values by 0.83.

TABLE 8. RETURN AIR—CARRYING CAPACITY—BTU SERVICED

Return Air Combination No.	Duct Dia In.	Type A Btu per Hr	Types B and C Btu per Hr	Type D Btu per Hr	Type E Btu per Hr	Type F Btu per Hr	Return Air Combination No.
31	10	11,300	9,500	7,800	5,000	7,800	31
32	12	16,300	13,700	11,300	7,200	11,300	32
33	14	22,200	18,700	15,300	9,800	15,300	33
34	16	29,000	24,400	20,000	12,800	20,000	34
35	18	36,700	30,800	25,300	16,200	25,300	35
36	20	45,300	38,000	31,300	20,000	31,300	36
37	22	54,800	46,000	37,800	24,100	37,800	37
38	24	65,200	54,800	45,000	28,700	45,000	38

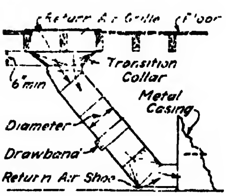
Design Procedure

The steps to be taken in designing a gravity warm air duct system are:

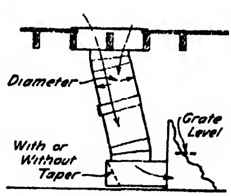
1. Calculate the heat loss from each room as explained in Chapters 6, 8 and 14.

2. Prepare a layout showing (*a*) furnace, (*b*) chimney connection, (*c*) warm air registers (whether floor, baseboard or wall), (*d*) return air grilles.

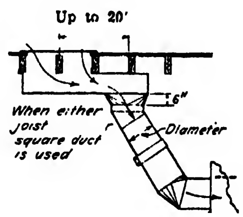
3. Indicate on each warm air run (using symbols shown in Fig. 6): (*a*) whether the room to be heated is on the first or second story, (*b*) the approximate length of leader pipe in the basement, (*c*) the number of right angle elbows required, including the elbow at the boot connection (see Fig. 2), (*d*) whether the register is to be located in the floor, in the baseboard, or in the wall.



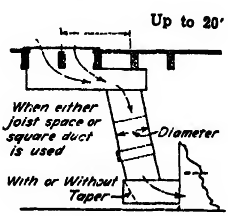
Type A



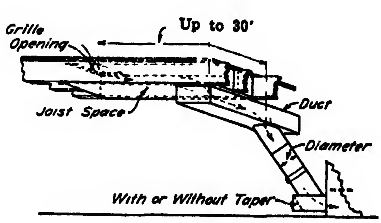
Type B



Type C



Type D



Type E

Same as Type E except that collar and shoe are same as for Type A.

Type F

Note: For Types C, D, E, and F return-air duct systems, reduce the carrying capacities shown in Table 8 by 1 per cent for each 4 ft additional length in the horizontal run.

FIG. 5. TYPICAL ARRANGEMENTS OF RETURN-AIR DUCT SYSTEMS

4. Show the number and proposed locations of return air grilles and the type of return air system (see Fig. 5).

5. From Table 6, for first story, or from Table 7 for second story, select the combination number for the warm air system which will supply the heat required to each room, with the number of elbows and length of leader pipe previously determined. Then, using the combination number as found, read directly in Tables 1, 2, or 3 the leader, stack, and register sizes required.

6. From Table 8 select the combination number for the return air system to correspond with the Btu serviced and the type of return air system. Then from Table 4 select the duct and grille sizes, etc., corresponding to the same combination number.

7. Select a furnace having a register delivery, in Btu per hour, equal to the total heat loss from the structure.

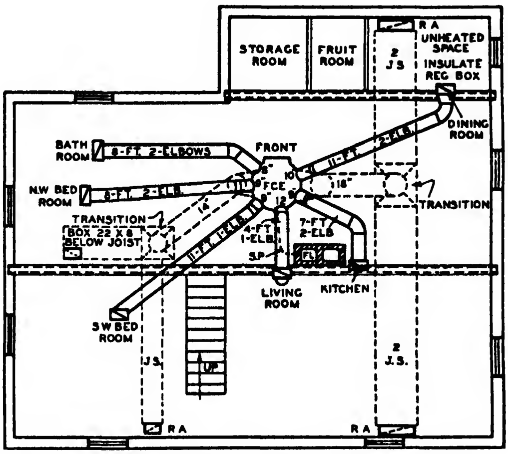


Fig. 6. Typical Basement Line Drawing

Design Examples

Examples 1 and 2 will illustrate the use of the tables in selecting warm air and return system sizes.

Example 1. For a room which has a heat loss of 22,500 Btu per hr select the size of first story warm air system. There are three elbows and the leader is approximately 10 ft long.

Solution: Since 22,500 Btu is beyond the capacities shown in Table 6, it is necessary to select two units of 11,250 each. From Table 6 in 10 ft leader column and in section for three elbows, find 11,400 as nearest capacity which corresponds to Combination Number 4 in first column. Refer to Combination Number 4 in Table 1 and find that the leader should be 12 in. in diameter and should be used with a 12 × 14 in. floor register or a 13 × 11 in. baseboard register with a 5¼ in. extension.

Example 2. What is the size of a return system of Type D which is to service 35,000 Btu per hr?

Solution: From Table 8 find Combination Number 37 which will service 37,800 Btu per hr. Refer to Table 4 to find that Combination Number 37 will require a 22-in. diameter duct, a shoe area of 420 sq in., a metal grille 18 × 30 in., a duct 42 × 10 in. or 36 × 12 in. If joist lining is used the minimum depth should be 15 in. for two 2-joist spaces 14 in. wide, or 10 in. for three joist spaces.

REFERENCES

[1] The engineering data were obtained from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo. See also Gravity Code and Manual for Gravity Warm Air Heating Systems, published by the *National Warm Air Heating and Air Conditioning Association.*

[2] Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm-Air Heating Plants, by A. P. Kratz and S. Konzo (*University of Illinois, Engineering Experiment Station Circular* 45, Dec., 1942).

[3] Gravity Code and Manual for Gravity Warm Air Heating Systems, Second Edition, 1945, *National Warm Air Heating and Air Conditioning Association.*

CHAPTER 22

MECHANICAL WARM AIR SYSTEMS

Air Distribution, Standard Combinations of Parts, Simplified Design of Heating System, Automatic Controls, Design Procedure for Larger Systems, Adjustment of System for Continuous Air Circulation, Cooling Methods

IN mechanical warm air or fan furnace heating systems[1], the air circulation is effected by motor-driven centrifugal fans, commonly referred to as blowers, instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom, as in gravity systems described in Chapter 21. The advantages of mechanical systems, as compared with gravity systems, are:

1. The furnace need not be centrally located but may be placed in any part of the basement.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view where desired.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

8. Ventilation air may be positively introduced and heated.

The construction features of mechanical warm air furnace units and discussions of the function and selection of the various parts, such as the furnace, casings, motors, filters and controls are included in Chapter 18.

AIR DISTRIBUTION

The conditions of comfort obtained in a room are influenced greatly by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. One method is to locate the supply register near the floor, or high in the side wall, so that the warm air from the register *blankets* a cold wall, and mixes with the cold air descending from the exposed walls and glass. Another method is to locate the supply openings near the floor, or high in the side wall, on the inside wall and the return openings near the greatest outside exposure. In any case the warm air registers should be located so that the air stream never discharges directly into space that will normally be occupied by people at rest. Tests in the Warm Air Research Residence[2] have indicated that continuous blower operation gave better results than intermittent operation.

Register and Grille Openings

Supply registers located in the floor require attention to keep them clean and are usually avoided. Tests conducted in the Warm Air Research Residence at the *University of Illinois* have indicated that comparable results are obtainable with either high side wall or baseboard registers, if proper registers and air velocities are selected. Baseboard registers should be of a deflecting-diffuser type which throw the air downward toward the floor and diffuse it at the same time. For baseboard registers air temperatures under 125 F and air velocities over 500 fpm should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing

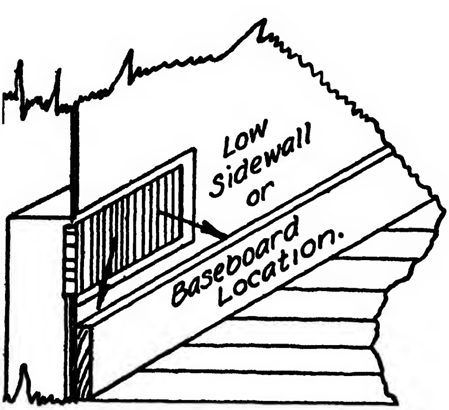


Fig. 1. Recommended Type of Baseboard and Low Sidewall Registers[a]

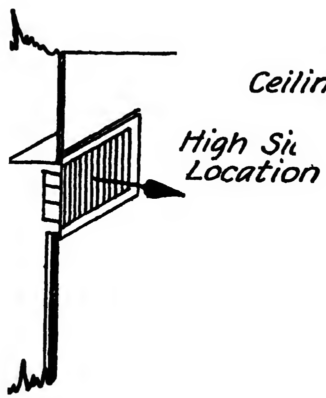


Fig. 2. Recommended Type of High Sidewall Registers[b]

[a] Vertical bars with adjustable deflection, or fixed vertical bars with deflections to right and left not exceeding about 22 deg. For low sidewall location, the deflection for horizontal, multiple valve, registers should not exceed 22 deg. For baseboard locations, the deflection for horizontal, multiple valve, registers should not exceed about 10 deg.

[b] Horizontal valves, in back or front, to give downward deflections not to exceed from 15 to 22 deg.

type registers should be used to insure best results. Register air velocities should be such that the air velocity will be about 50 fpm three quarters of the distance from the register to the opposite wall.

Velocities through registers may be reduced by the use of registers larger than the connecting ducts. Diffusers should be used to spread the air uniformly over the register face. Basic rules for the location and selection of registers are given in Section C of Manual No. 7 of the *National Warm Air Heating and Air Conditioning Association.*

Registers should be well proportioned and decorated to harmonize with the trim. Air supply registers should be equipped with dampers and all registers should be sealed against leakage around edges. The register types shown in Figs. 1 and 2 have been recommended as standard by the *National Warm Air Heating and Air Conditioning Association.*

Return air grilles may be located in hallways, near entrance doors, under windows, in exposed corners, or inside walls, depending on location of supply registers. Baseboard returns are preferable to floor grilles.

TABLE 1. WARM AIR DUCT SYSTEM COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination No.	Stack Size In.	Branch Pipe Size, In.		Register Size, In. (See Figs. 1 and 2)		Required Increase in Width of Trunk Duct, In.
		Round	Rectangular	Base-board High or Low Sidewall	Floor Registers[a]	
1	2	3	4	5	6	7
41	10 x 3¼	6	4 x 8	10 x 6	8 x 10[a]	1
42	10 x 3¼	6	4 x 8	10 x 6	8 x 10[a]	2
43	12 x 3¼	7	5 x 8	12 x 6	9 x 12[a]	3
44	14 x 3¼	8	6 x 8	14 x 6	9 x 12[a] or longer	4
45	10 x 3¼ (2-Stacks)	9	8 x 8	(2) 10 x 6 or (1) 24 x 6	10 x 12[a]	5
46	12 x 3¼ (2-Stacks)	10	10 x 8	(2) 12 x 6 or (1) 30 x 6	12 x 14[a]	7

[a] Use these items only when the building construction or capacity requirements necessitate the use of floor registers. The sizes listed for floor registers correspond to the standard sizes for gravity warm air furnace systems, except for the sizes of the floor box collars. The use of standard blind boxes is suggested. A 12 x 5¼ in. stack may be used on Combination 45 and a 14 x 5¼ in. stack on Combination 46 with floor registers.

Dampers

Suitable dampers for air direction or volume control are essential to any duct system. Special care must be used in the design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. Three types of dampers are commonly used. *Volume dampers* are used to completely cut off or reduce the flow through pipes. *Splitter dampers* are used where a branch is taken off from a main trunk. *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. It is essential that a damper with positive locking device be provided for each main or duct branch. Labels placed on ducts should indicate the room being served. Damper positions should be marked for summer and winter operation, and to avoid tampering.

Ducts

The ducts may be either round or rectangular in cross section. The radii of elbows should preferably be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts. Warm air ducts passing through cold spaces, or where located in exposed walls, should have ½ to 2 in. of insulation.

Special attention should be given to the problem of noise elimination. The metal duct connection to and from the furnace casing and fan housing should be broken by strips of canvas. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. Installation of a fan directly under a cold air grille is usually avoided.

Table 2. Return-Air Duct System Combinations of Parts Selected as Standard

Combination No.	Return-Air Intake Size, In.		Riser Size, In. Where Stack is Used in Stud Space	Branch Pipe Size, In.		When Joist Lining is Used[b] Number of Joist Spaces Lined and Minimum Depth of Space Required	Required Increase in Width of Trunk Duct (for 8 In. Depth of Duct), In.
	Baseboard	Floor[a]		Round	Rectangular		
1	2	3	4	5	6	7	8
51	10 x 6	6 x 10 or 4 x 14	10 x $3\frac{1}{4}$[e]	6	4 x 8	1 space of 3 in. depth	1
52	10 x 6	6 x 10 or 4 x 14	10 x $3\frac{1}{4}$[e]	6	4 x 8	1 space of 3 in. depth	2
53	12 x 6	6 x 12 or 6 x 14	12 x $3\frac{1}{4}$[d]	7	5 x 8	1 space of 4 in. depth	3
54	14 x 6	6 x 14	14 x $3\frac{1}{4}$[d]	8	6 x 8	1 space of 5 in. depth	4
55	24 x 6 or 30 x 6	6 x 30	Two stacks each 10 x $3\frac{1}{4}$[e]	9	8 x 8	1 space of 6 in. depth or 2 spaces of 3 in. depth	5
56	30 x 6	6 x 30	Two stacks 12 x $3\frac{1}{4}$[d]	10	10 x 8	1 space of 7 in. depth or 2 spaces of 4 in. depth	7
57	—	8 x 30	—	12	15 x 8	1 space of 9 in. depth or 2 spaces of 5 in. depth	12

[a] Use these items only when building construction, or capacities, require the use of floor intakes. The sizes listed correspond to standard sizes for gravity installations, except floor box collars. The use of standard blind boxes is suggested.

[b] Based on 14 in. space between joists. Use full depth of joist, except when joist depth is less than minimum depth required, in which case a *drop pan must be used.* This may occur when two or more return ducts are connected to the same joist space.

[c] If it is desired to use 14 in. x $3\frac{1}{4}$ in. stud space, it makes no difference whether this space has protruding keys or not.

[d] If it is desired to use 14 in. x $3\frac{1}{4}$ in. stud space, the plaster base must be smooth, without any protruding plaster keys to interfere with the flow of air.

STANDARD COMBINATIONS OF PARTS

The combinations of parts selected as standard by the *National Warm Air Heating and Air Conditioning Association* are shown in Tables 1 and 2. A method for selecting these combinations is indicated in the following section *Simplified Method of Design.*

SIMPLIFIED METHOD OF DESIGN

A simplified method for selecting the combinations of branches, boots, stacks, and registers, is given in Manual No. 7 of the *National Warm Air Heating and Air Conditioning Association.* In this method the sizes of the branch ducts are obtained from two tables giving their Btu capacities. The proper combination of parts for each branch can be determined if the following information is available.

a. Location of room, that is, whether on first or second story.

b. Actual length of duct from bonnet to boot, in feet.

c. Btu loss from room to be heated.

d. Equivalent lengths in feet of all fittings and of the register. Fig. 3 shows the values of equivalent lengths of fittings commonly used for domestic systems.

This simplified method is applicable to structures having heat losses not in excess of approximately 120,000 Btu per hour. The capacities shown in Tables 3 and 4 are based upon the most reliable data pertaining to friction losses and temperature drops in ducts. They are also based upon a 100 deg temperature rise of the air, and a static pressure available for overcoming friction losses in the external duct system alone of 0.20 in. water gage. The use of this method assumes that the fan in the fan-furnace assembly will be capable not only of overcoming the resistance of the external duct system alone, but also the resistances imposed by the blower inlet, the filter, and the furnace casing. The combination numbers shown in the right hand column of Tables 3 and 4 correspond to those given in Tables 1 and 2. Tables 3 and 4 are also applicable for the selection of the return air branches. A depth of 8 in. has been adopted as the standard for the trunk ducts. The width of a trunk duct serving two branches is determined by adding to the width of the remote branch the value shown in column 7 of Table 1, or column 8 of Table 2.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in a properly designed system can be largely eliminated through proper care in the planning and installation of the control system[3]; desirable controls usually employed are:

1. A *thermostat* located in a living room where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. The thermostat location should not be on an outside wall, in a bedroom, bath room or sun room, or in a location where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct, register or chimney.

2. A *fan switch* control located in the bonnet to start blower operations at temperatures between 110 and 130 F, and to stop the blower at about 25 to 30 deg below the cut-in point. The lower settings are used for high side wall register installations, and the higher settings for baseboard register installations. For most satisfactory results these settings should be as low as is feasible.

3. A *protective high limit switch* located in the bonnet to stop the system independently of the thermostat if the bonnet temperature exceeds 175 F.

4. On oil and gas burner installations, a protective control should be included which will stop the system if the fire is extinguished or if there is a failure of the ignition system.

5. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying, or a time interval contactor is used that will start the stoker to run a few minutes out of each hour.

6. A *humidistat* to regulate the moisture supplied to the rooms, located either in one of the rooms or in the main return duct near the furnace.

DESIGN PROCEDURE FOR LARGE SYSTEMS[4]

For buildings having a heat loss in excess of 120,000 Btu per hour the design procedure given in Manual No. 9 of the *NWAH & ACA*, may be used except where ventilation air volume exceeds volume required to supply calculated heat loss. This procedure consists of:

1. Calculation of design heat losses from individual spaces in the structure. (See Chapter 14.)

Table 3. Capacity Tables for Warm-Air and Return-Air Branches[a, b]

FIRST STORY

For UNINSULATED Metal Ducts	Actual Length (from Bonnet to Boot) or, (from Return Plenum to Boot) in Feet							Warm Air Combination No.	Return Air Combination No.
	1 to 7 ft.	8 to 12 ft.	13 to 17 ft.	18 to 24 ft.	25 to 34 ft.	35 to 44 ft.	45 to 54 ft.		
	Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	Col. G		
Section A.	7200	6700	6100	5600	4800	4100	3500	41	51
	12500	11700	10800	9900	8500	7400	6400	42	52
40 to 69	16000	15000	14000	13000	11300	9900	8700	43	53
Equivalent Ft for Fittings and Register	19100	18000	17000	16000	14200	12500	11000	44	54
	25000	23400	21600	19800	17000	14800	12800	45[c]	55
	32000	30000	28000	26000	22600	19800	17400	46[c]	56
	80000	75000	70000	65000	56500	49500	43500	—	57[c]
Section B.	5500	5100	4800	4500	3900	3400	3000	41	51
	9900	9200	8600	8100	7100	6200	5400	42	52
70 to 99	13100	12300	11600	10900	9700	8500	7500	43	53
Equivalent Feet	16300	15400	14500	13700	12200	10800	9500	44	54
	19800	18400	17200	16200	14200	12400	10800	45[c]	55
	26200	24600	23200	21800	19400	17000	15000	46[c]	56
	65500	61500	58000	54500	48500	42500	37500	—	57[c]
Section C.	4600	4300	4100	3800	3300	3000	2700	41	51
	8500	7900	7400	6900	6100	5300	4700	42	52
100 to 129	11300	10600	10000	9400	8400	7400	6500	43	53
Equivalent Feet	14300	13500	12700	11900	10500	9300	8300	44	54
	17000	15800	14800	13800	12200	10600	9400	45[c]	55
	22600	21200	20000	18800	16800	14800	13000	46[c]	56
	56200	52600	50200	47300	42000	37000	32400	—	57[c]
Section D.	4100	3800	3600	3400	2900	2600	2300	41	51
	7300	6900	6400	6000	5300	4700	4200	42	52
130 to 164	9800	9100	8600	8100	7200	6400	5600	43	53
Equivalent Feet	12300	11700	11000	10300	9100	8100	7100	44	54
	14600	13800	12800	12000	10600	9600	8400	45[c]	55
	19600	18200	17200	16200	14400	12800	11200	46[c]	56
	49200	45800	41900	40300	36000	31800	28100	—	57[c]
Section E.	3800	3500	3300	3100	2700	2400	2100	41	51
	6500	6100	5700	5400	4800	4300	3800	42	52
165 to 200	8800	8200	7700	7200	6400	5700	5000	43	53
Equivalent Feet	11000	10500	9900	9300	8300	7300	6400	44	54
	13000	12200	11400	10800	9600	8600	7600	45[c]	55
	17600	16400	15400	14400	12800	11400	10000	46[c]	56
	44100	41500	38800	36000	32000	28200	24700	—	57[c]

For INSULATED Ducts	Col. A	Col. B	Col. C	Col. D	Col. E	For Ducts that are COMPLETELY INSULATED with ½ In. Thick Insulation from Bonnet to Boot, use these Column Headings.
	1 to 9 ft.	10 to 17 ft.	18 to 24 ft.	25 to 34 ft.	35 to 54 ft.	

[a] These tables are for use in sizing both the warm air and the return air branches.

[b] Frictional resistances and temperature drops in ducts have both been accounted for in these tables.

[c] Use these items only when the building construction, or capacity requirements, necessitate the use of two adjoining stacks or floor registers.

Table 4. Capacity Tables for Warm-Air and Return Air Branches[a,b]

SECOND STORY

For Uninsulated Metal Ducts	Actual Length (from Bonnet to Boot) or, (from Return Plenum to Boot) in Feet: 1 to 7 ft. Col. A	8 to 12 ft. Col. B	13 to 17 ft. Col. C	18 to 24 ft. Col. D	25 to 34 ft. Col. E	35 to 44 ft. Col. F	45 to 54 ft. Col. G	Warm Air Combination No.	Return Air Combination No.
Section A.	6300	5700	5200	4800	4100	3500	3100	41	51
	10900	10000	9200	8500	7300	6400	5600	42	52
40 to 69	14000	13000	12100	11400	10000	8800	7800	43	53
Equivalent Ft for	17000	15900	14900	13900	12400	11100	9900	44	54
Fittings	21800	20000	18400	17000	14600	12800	11200	45[c]	55
and	28000	26000	24200	22400	20000	17600	15600	46[c]	56
Register	70000	65000	60500	57000	50000	44000	39000	—	57[c]
Section B.	5000	4600	4300	4000	3400	3000	2700	41	51
	9000	8200	7600	7100	6200	5400	4700	42	52
70 to 99	11900	11000	10300	9600	8400	7500	6700	43	53
Equivalent Feet	14800	13900	13000	12200	10800	9600	8500	44	54
	18000	16400	15200	14200	12400	10800	9400	45[c]	55
	23900	22000	20600	19200	16800	15000	13400	46[c]	56
	59500	55000	51500	48000	42000	37500	33500	—	57[c]
Section C.	4200	3900	3700	3500	3000	2600	2400	41	51
	7700	7200	6700	6200	5400	4700	4100	42	52
100 to 129	10400	9700	9000	8400	7400	6500	5800	43	53
Equivalent Feet	13000	12100	11300	10600	9400	8300	7400	44	54
	15400	14400	13400	12400	10800	9400	8200	45[c]	55
	21700	19400	18000	16800	14800	13000	11600	46[c]	56
	52200	48400	45300	42300	37000	32500	29000	—	57[c]
Section D.	3800	3500	3200	3000	2700	2300	2100	41	51
	6800	6300	5800	5500	4800	4200	3700	42	52
130 to 164	9100	8400	7900	7400	6300	5700	5000	43	53
Equivalent Feet	11400	10500	9800	9200	8100	7200	6400	44	54
	13600	12600	11600	11000	9600	8400	7400	45[c]	55
	18200	16800	15800	18400	12600	11400	12800	46[c]	56
	45500	42000	39600	36800	32200	28200	25100	—	57[c]
Section E.	3500	3200	2900	2700	2500	2100	1900	41	51
	6100	5700	5300	5000	4400	3900	3400	42	52
165 to 200	8200	7600	7100	6700	5700	5100	4500	43	53
Equivalent Feet	10300	9500	8900	8300	7400	6500	5800	44	54
	12200	11400	10600	10000	8800	7800	6800	45[c]	55
	16400	15200	14200	13400	11400	10200	9000	46[c]	56
	41000	38300	35800	33400	29000	25800	22800	—	57[c]

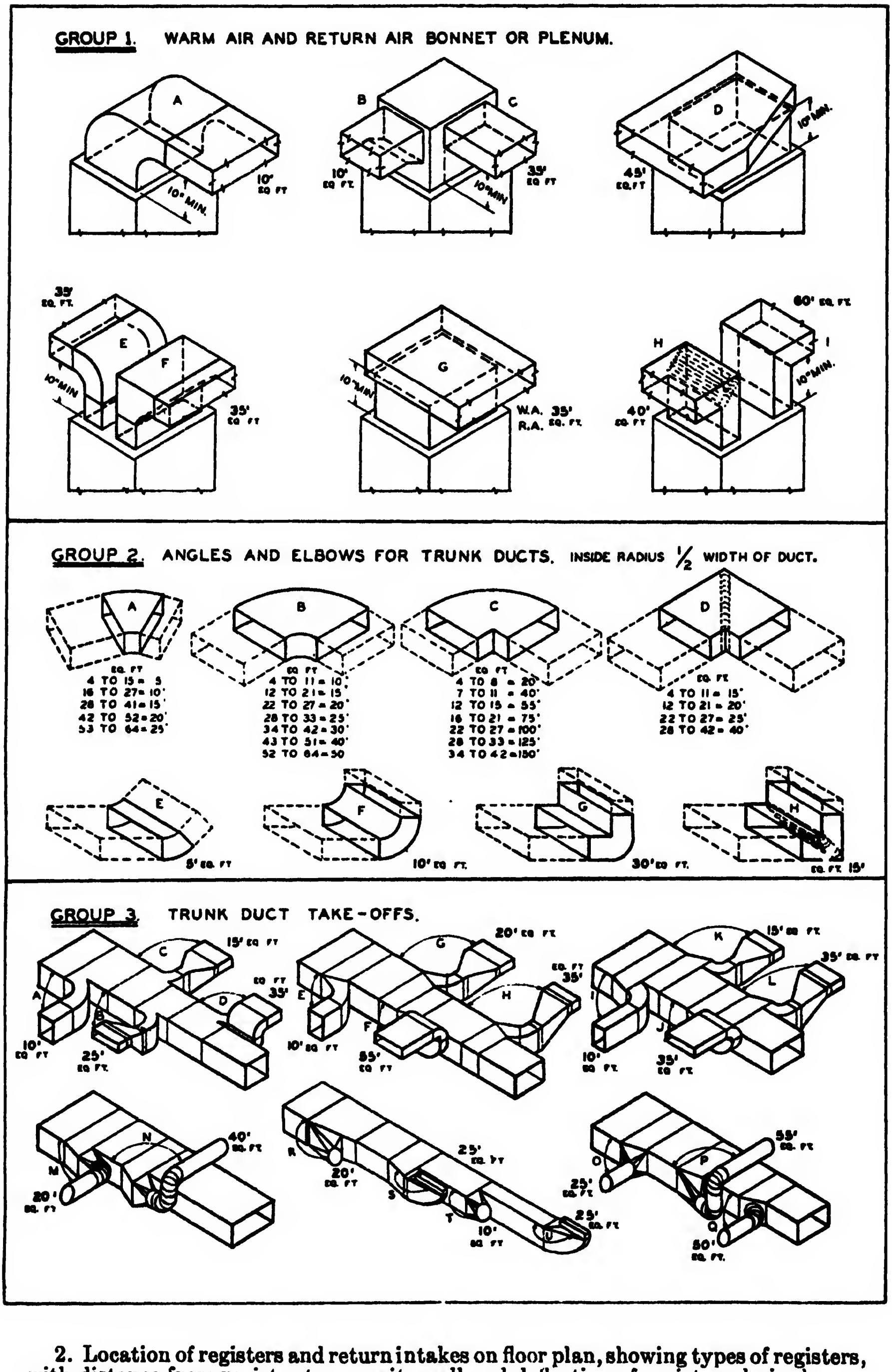
For Insulated Ducts	Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	For Ducts that are COMPLETELY INSULATED with ½ In. Thick Insulation from Bonnet to Boot Use these Column Headings.
	1 to 8 ft.	9 to 14 ft.	15 to 20 ft.	21 to 27 ft.	28 to 42 ft.	43 to 54 ft.	

[a] These tables are for use in sizing both the warm air and the return air branches.

[b] Frictional resistance and temperature drops in ducts have both been accounted for in these tables.

[c] Use these items only when the building construction, or capacity requirements, necessitate the use of two adjoining stacks or floor registers.

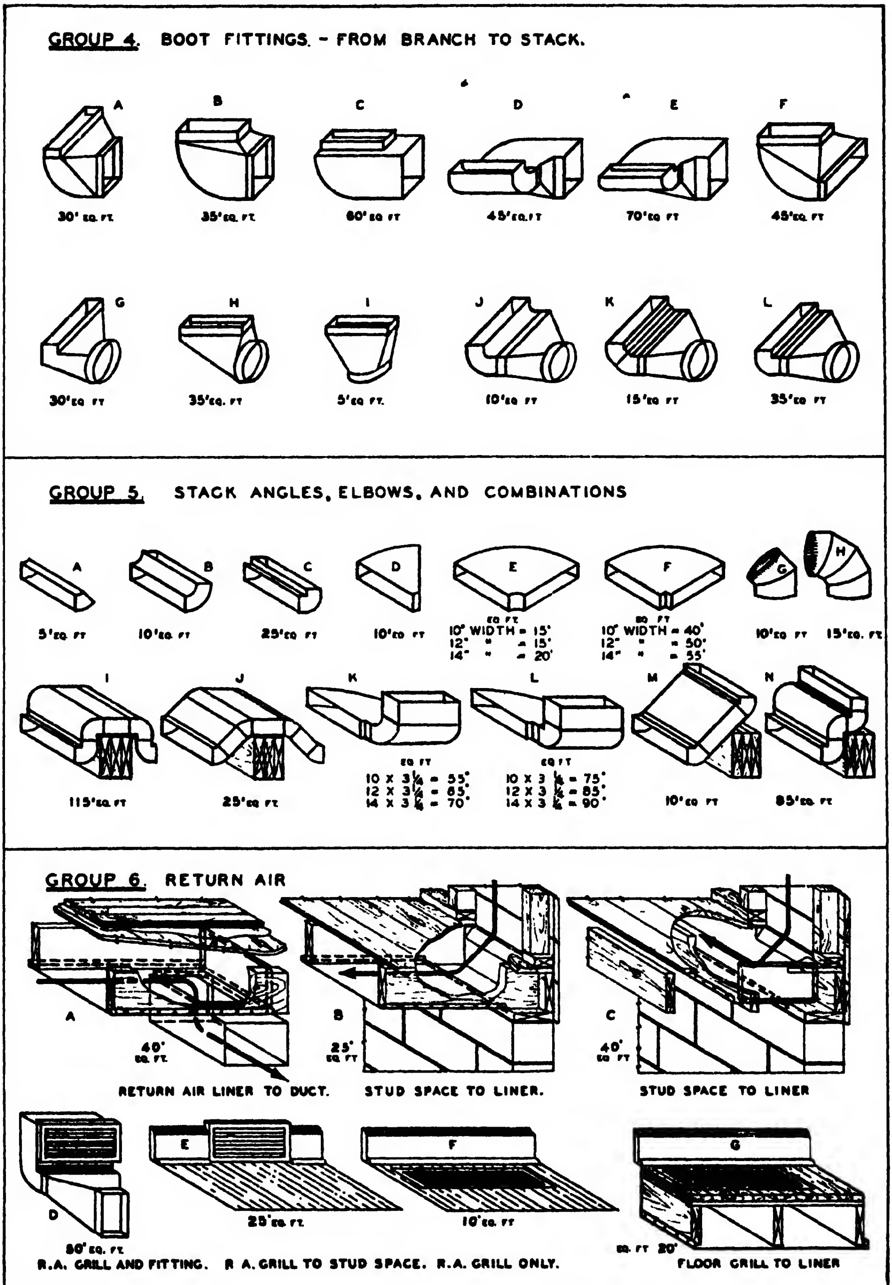
Fig. 3. Equivalent Length of Fittings



2. Location of registers and return intakes on floor plan, showing types of registers, with distance from register to opposite wall and deflection of registers desired.

3. Laying out a proposed duct system for both warm air and return air sides of the system and including details of types of fittings and the actual and equivalent

Fig. 3. Equivalent Length of Fittings (Continued)



lengths of each branch line from bonnet to register, without sizes. (See Fig. 3, Groups 1 through 6, for equivalent length of fittings.)

4. Determination of bonnet temperature.

If rating sheet for furnace-blower unit specifies a *fixed value of bonnet temperature*, enter table at this value. If not specified, use the following procedure: Use Table

FIG. 3. EQUIVALENT LENGTH OF FITTINGS (Concluded)

GROUP 7. REGISTERS (INCLUDING LOSSES IN STACKHEAD AND VELOCITY PRESSURE).

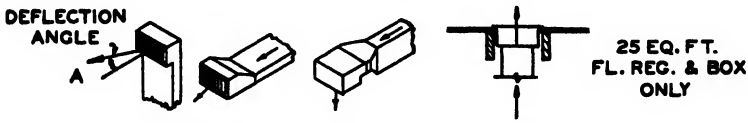


TABLE 4A. EQUIVALENT LENGTH FOR REGISTERS

Deflection Angle A	0°	15°	22°	30°	45°
Baseboard, High or Low Sidewall Registers	Eq. Ft. 35	40	45	60	115
Floor Registers with Box only	25				

For 2-way deflection registers, add the vertical and horizontal deflection angles together and multiply by 0.7. Select closest angle A in Table 4A.

5 for buildings having a heat loss between 120,000 and 350,000 Btu per hr or Table 6 for buildings having a heat loss greater than 350,000 Btu per hr. Select *shortest actual length* and read downward in nearest column in Tables 5 or 6 until lower heavy diagonal line is reached, but do not cross line. Run horizontally to left to obtain value of *maximum bonnet temperature* in first column. Also select *longest actual length* and read downward in nearest column in Tables 5 or 6, until upper heavy diagonal line is just crossed. Run horizontally to left to obtain value for *minimum bonnet temperature* in first column. Select as the design bonnet temperature any value between maximum and minimum.

5. Determination of air volume to be delivered through each register and the respective register air temperatures.

Using Tables 5 or 6 and the design bonnet temperature selected, find the values of cfm per 1000 Btu for each duct length.

6. Selection of register sizes and pressure losses to produce necessary throw, for the air volumes handled.

Use Tables 7 or 8 to obtain required free area and pressure loss of register.

7. Design of duct system.

A. *Warm air branches.*

a. Use Table 9 to select maximum bonnet pressure *usually* required for the trunk carrying the maximum volume of air (cfm). If the maximum bonnet pressure is not high enough to accomodate the pressure loss through the registers, use a higher bonnet pressure. If the register pressure is critically large, it may be necessary to reduce it by either using two registers in place of one, or using smaller deflection angles.

b. Obtain actual duct loss by subtracting register pressure loss from maximum bonnet pressure.

c. Obtain the pressure drop in each duct per 100 ft by use of Table 10.

d. By means of Table 10 (Friction chart) determine duct size, cfm, and pressure drop per 100 ft of duct.

B. *Return air branches.*

a. Select a low value of actual duct loss obtained from step b under item A for the suction loss of return duct system.

b. Proceed in sizing return air branches by the same method described for the warm air branches.

C. *Trunk ducts for warm air and return air sides of system.*

a. Add air volumes of branches to be handled by each trunk duct.

b. The friction loss per 100 ft of trunk duct is determined by taking the smaller of the two values for friction loss for the two ducts meeting at the junction.

c. Determine trunk duct size by using Friction Chart, volume (cfm), and pressure drop per 100 ft of trunk duct.

Table 5. Selection of Bonnet Temperature, Register Temperature and Register Delivery (CFM)[a, b]
(For Buildings Having a Heat Loss between 120,000 and 350,000 BTU/Hr.)
Register Delivery Volume in CFM per 1000 BTU per Hr.

Temp. of Bonnet °F	Linear Distance from Bonnet to Register, in Feet																							
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240
110	108[a] 22.8[b]	107 23.6	106 24.2	104 25.4	103 26.1	102 26.9	101 27.3	100 28.1	99 29.2	98 30.0	97 30.8	96 31.8	95 32.8	94 34.0	93 34.5	92 35.7	91 37.0	90 38.5	90 39.2	89 39.9	89 40.6	88 41.3	88 42.2	87 43.2
120	118 18.9	116 19.6	114 20.2	112 21.0	111 21.7	109 22.5	108 23.0	106 24.1	105 24.9	103 25.7	102 26.7	101 27.3	100 28.0	99 28.8	98 30.0	97 30.8	96 31.5	95 32.9	94 33.4	93 34.6	92 35.7	91 37.0	90 38.2	90 39.2
130	128 16.5	126 17.0	123 17.6	121 18.2	119 18.8	117 19.4	115 20.0	113 20.7	111 21.4	110 22.1	108 22.9	107 23.7	105 24.6	104 25.5	103 26.1	102 26.8	101 27.7	99 28.7	98 29.6	97 30.6	96 31.5	95 32.3	94 33.1	93 34.0
140	137 14.6	134 15.1	131 15.7	129 16.2	126 16.8	124 17.3	122 17.9	120 18.5	118 19.1	116 19.7	114 20.3	112 21.1	110 22.1	109 22.5	107 23.4	106 24.2	105 25.1	103 25.8	102 26.7	101 27.3	100 28.1	99 28.9	98 30.0	97 30.8
150	147 12.9	143 13.5	140 14.0	137 14.6	134 15.1	131 15.7	129 16.2	127 16.7	124 17.3	122 17.9	120 18.5	118 19.0	116 19.6	114 20.3	112 21.1	110 21.7	109 22.5	107 23.4	106 24.2	105 25.1	103 25.8	102 26.7	101 27.3	100 28.1
160	156 11.9	152 12.3	149 12.7	145 13.2	142 13.7	139 14.2	136 14.8	133 15.3	130 15.9	128 16.5	126 17.0	123 17.5	121 18.1	119 18.7	117 19.3	115 19.9	113 20.7	112 21.4	110 22.1	108 22.9	107 23.6	106 24.5	109 25.4	103 26.1
170	166 10.9	161 11.3	157 11.7	153 12.1	150 12.6	146 13.1	143 13.6	140 14.1	137 14.6	134 15.1	131 15.7	128 16.3	126 16.8	124 17.4	122 17.9	120 18.5	118 19.2	116 19.8	114 20.4	112 21.1	110 22.1	109 22.6	107 23.4	106 24.2
180	175 10.0	170 10.5	166 10.9	161 11.3	157 11.7	153 12.1	150 12.6	146 13.1	143 13.6	140 14.1	137 14.6	135 15.1	131 15.7	128 16.3	126 16.8	124 17.4	122 18.0	120 18.6	118 19.1	116 19.7	115 20.2	113 20.7	111 21.5	110 22.3
190	184 9.4	179 9.8	174 10.2	169 10.6	165 10.9	160 11.3	156 11.8	153 12.2	149 12.7	145 13.2	142 13.7	139 14.2	136 14.7	133 15.3	130 15.9	128 16.4	126 16.9	124 17.5	121 18.1	119 18.7	117 19.3	115 19.9	113 20.7	112 21.4
200	193 8.9	187 9.2	182 9.6	176 9.9	171 10.3	167 10.7	163 11.2	158 11.6	154 12.0	151 12.4	147 12.9	144 13.4	140 13.9	138 14.5	135 15.1	132 15.7	129 16.2	127 16.6	125 17.2	122 17.8	120 18.4	118 19.0	116 19.6	114 20.3

[a] Register temperature, Fahrenheit. (Use upper value in each group).
[b] Register delivery volume in cubic feet per minute per 1000 Btu (Use lower value in each group).

Table 6. Selection of Bonnet Temperature, Register Temperature and Register Delivery (CFM)[a, b]
(For Buildings Having a Heat Loss Greater than 350,000 BTU/Hr.)
Register Delivery Volume in CFM per 1000 BTU per Hr.

Bonnet Temp. °F.	Linear Distance from Bonnet to Register, in Feet																								
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240	250
140	138[a]	136	134	132	130	128	127	126	125	124	123	121	120	119	118	117	116	115	114	113	112	111	110	109	108
	14.4[b]	14.8	15.2	15.6	16.0	16.4	16.6	16.9	17.1	17.4	17.6	18.2	18.4	18.7	19.0	19.3	19.7	20.0	20.4	20.8	21.2	21.6	22.1	22.5	23.1
150	148	146	143	141	139	137	135	133	131	129	128	127	126	124	123	122	120	119	118	117	116	115	114	113	112
	12.8	13.1	13.5	13.9	14.2	14.6	15.0	15.4	15.8	16.2	16.4	16.6	16.9	17.4	17.6	17.9	18.4	18.7	19.0	19.3	19.7	20.0	20.4	20.8	21.2
160	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123	121	120	119	118	117	116
	11.6	12.0	12.3	12.6	13.0	13.2	13.5	14.0	14.4	14.8	15.2	15.6	16.0	16.4	16.6	16.9	17.1	17.4	17.6	18.2	18.4	18.7	19.0	19.3	19.7
170	167	164	161	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123	121	120	119
	10.8	11.0	11.3	11.6	12.0	12.3	12.6	13.0	13.2	13.5	14.0	14.4	14.8	15.2	15.6	16.0	16.4	16.6	16.9	17.1	17.4	17.6	18.2	18.4	18.7
180	177	173	170	167	164	161	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123
	9.9	10.2	10.5	10.8	11.0	11.3	11.6	12.0	12.3	12.6	13.0	13.2	13.5	14.0	14.4	14.8	15.2	15.6	16.0	16.4	16.6	16.9	17.1	17.4	17.6
190	186	182	178	175	172	168	165	162	159	156	153	151	148	146	143	141	139	137	135	133	131	129	128	127	126
	9.3	9.6	9.8	10.0	10.3	10.7	10.9	11.2	11.5	11.8	12.2	12.4	12.8	13.1	13.5	13.9	14.2	14.6	15.0	15.4	15.8	16.2	16.4	16.6	16.9
200	196	191	187	183	179	176	172	169	166	163	160	157	154	151	149	147	144	142	140	138	135	133	131	130	128
	8.8	9.0	9.2	9.5	9.8	10.0	10.3	10.6	10.8	11.1	11.4	11.7	12.1	12.4	12.7	13.0	13.4	13.7	14.0	14.4	15.0	15.4	15.8	16.0	16.4

[a] Register temperature, Fahrenheit (Use upper value in each group).
[b] Register delivery volume in cubic feet per minute per 1000 Btu (Use lower value in each group).

Table 7. Determination of Free Area and Pressure Loss of Register for 22 Deg Deflection of Air[a, b, c, d, e]

Register Free Area in Square Inches (Upper value in each group)
Pressure Loss in Inches of Water (Lower value in each group)

Cfm	Residential Use: Reg. Size or Equivalent	Residential Use: Pressure Loss
Up to 59	10 x 4 10 x 6	.01 .01
60-69	10 x 6 12 x 4	.01 .02
70-99	10 x 6 12 x 6	.02 .02

Cfm	Residential Use: Reg. Size or Equivalent	Residential Use: Pressure Loss
100-119	12 x 6 14 x 4	.02 .02
120-129	12 x 6 14 x 6	.02 .02
130-169	14 x 6	.02
170-189	14 x 8	.02

Cfm	Distance From Register of Opposite Wall: Up to 18	19-21	22-24	25-27	28-30	31-34	35-39	40-49	50-59
190-209	68 0.02	49 0.03	38 0.05	29 0.08	24 0.11	19 0.16			
210-229	82 0.02	60 0.03	46 0.04	35 0.06	28 0.10	23 0.13			
230-249		71 0.02	54 0.04	42 0.06	34 0.08	28 0.11	22 0.17		
250-269		84 0.02	63 0.03	49 0.05	40 0.07	32 0.10	25 0.16		
270-299		100 0.02	76 0.03	60 0.04	48 0.06	38 0.09	30 0.13		
300-339			96 0.02	73 0.04	60 0.05	48 0.07	37 0.11		
340-379			122 0.02	95 0.03	78 0.04	61 0.06	47 0.09	33 0.18	
380-419				117 0.02	94 0.03	75 0.05	58 0.08	39 0.15	
420-459				142 0.02	113 0.03	91 0.04	70 0.06	47 0.13	
460-499				163 0.02	135 0.03	108 0.04	83 0.06	56 0.11	
500-539					159 0.02	126 0.03	97 0.05	66 0.10	
540-579					185 0.02	147 0.03	113 0.04	77 0.08	
580-619					212 0.02	169 0.03	130 0.04	88 0.08	51 0.18
620-659						192 0.02	147 0.03	100 0.07	59 0.15
660-699						217 0.02	166 0.03	113 0.06	75 0.13

Cfm	Distance From Register to Opposite Wall: 31-34	35-39	40-49	50-59	60-69	70-79	80-89
700-739	243 0.02	186 0.03	127 0.06	85 0.11			
740-779	271 0.02	208 0.03	141 0.05	94 0.10			
780-819		230 0.02	157 0.05	105 0.09			
820-859		254 0.02	173 0.04	115 0.08			
860-899		278 0.02	190 0.04	126 0.08			
900-939		304 0.02	207 0.04	138 0.07			
940-979		332 0.02	226 0.03	151 0.07	108 0.12		
980-1019		360 0.02	245 0.03	163 0.06	118 0.11		
1020-1059		390 0.02	265 0.03	177 0.06	127 0.11		
1060-1099			285 0.03	190 0.06	137 0.10		
1100-1139			307 0.03	204 0.05	147 0.09		
1140-1179			330 0.02	220 0.05	158 0.09		
1180-1219			353 0.02	235 0.04	169 0.08		
1220-1259			376 0.02	251 0.04	180 0.08		
1260-1299			401 0.02	267 0.04	192 0.07	136 0.13	
1300-1339			426 0.02	284 0.04	204 0.07	144 0.13	
1340-1379			452 0.02	301 0.04	217 0.06	154 0.12	109 0.22
1380-1419			479 0.02	319 0.04	230 0.06	163 0.12	116 0.21
1420-1459			508 0.02	338 0.03	244 0.06	172 0.11	122 0.20
1460-1500			536 0.02	356 0.03	256 0.06	182 0.10	129 0.19

A B A′ B′

[a] If register selected based on distance from register to opposite wall is unsatisfactory on account of size or pressure loss, it is permissible to shift one or more spaces left or right in the tables to obtain a more suitable register. If requirements fall in blank space, select two registers in place of one and divide CFM capacity between the two registers.

[b] Pressure loss is based on FLAT FACE ADJUSTABLE BAR TYPE and does NOT include stackhead.

[c] Values on the right of line A and A′ should not be used in applications such as churches, auditoriums, and concert halls.

[d] Values on right of line B and B′ should not be used in applications such as residential work, motion picture theaters, court rooms and schools.

[e] For floor and baseboard registers where a velocity of approximately 300 FPM is used, the free area $= \frac{CFM \times 144}{300}$ or approximately, $\frac{CFM}{2}$. Assume a pressure loss of .01.

Table 8. Determination of Free Area and Pressure Loss of Register for No Deflection of Air[a, b, c, d, e]

Register Free Area in Square Inches (Upper value in each group)
Pressure Loss in Inches of Water Column (Lower value in each group)

CFM	Distance from Register to Opposite Wall								
	19–21	22–24	25–27	28–30	31–34	35–39	40–49	50–59	60–69
190[f]–209	62 0.02	47 0.03	37 0.04	30 0.06	24 0.09	18 0.15			
210–229	76 0.02	58 0.02	45 0.04	36 0.05	29 0.08	22 0.12			
230–249		69 0.02	53 0.03	43 0.04	34 0.07	26 0.11			
250–269		81 0.02	63 0.03	50 0.04	40 0.06	31 0.09	21 0.18		
270–299		93 0.02	73 0.02	59 0.03	47 0.05	36 0.08	24 0.16		
300–339			95 0.02	77 0.03	61 0.04	46 0.06	32 0.12		
340–379			120 0.02	97 0.02	77 0.03	59 0.05	40 0.10		
380–419				119 0.02	95 0.03	73 0.04	50 0.08		
420–459				149 0.02	114 0.03	88 0.04	60 0.07		
460–499				171 0.02	136 0.02	105 0.03	71 0.06		
500–539					160 0.02	123 0.03	84 0.05		
540–579					186 0.02	143 0.02	97 0.05	65 0.09	47 0.17
580–619					213 0.02	164 0.02	111 0.04	74 0.08	54 0.15
620–659						180 0.02	127 0.04	85 0.07	61 0.14

CFM	Distance from Register to Opposite Wall					
	35–39	40–49	50–59	60–69	70–79	80–89
660–699	210 0.02	143 0.03	95 0.07	69 0.12	49 0.22	
700–739	236 0.02	160 0.03	107 0.06	77 0.11	54 0.21	
740–779	262 0.02	179 0.03	119 0.06	86 0.10	61 0.18	
780–819	291 0.02	198 0.03	132 0.05	95 0.09	67 0.17	
820–859	320 0.02	218 0.03	145 0.05	105 0.08	74 0.15	
860–899	352 0.02	240 0.02	160 0.04	115 0.08	81 0.14	
900–939	385 0.01	262 0.02	175 0.04	126 0.07	89 0.13	
940–979	419 0.01	285 0.02	190 0.04	137 0.07	97 0.12	
980–1019	455 0.01	309 0.02	206 0.04	148 0.06	105 0.11	
1020–1059	493 0.01	335 0.02	223 0.03	161 0.06	113 0.11	
1060–1099	530 0.01	361 0.02	241 0.03	173 0.06	123 0.10	
1100–1139	571 0.01	388 0.02	258 0.03	186 0.05	132 0.09	93 0.17
1140–1179		416 0.02	277 0.03	199 0.05	141 0.08	100 0.16
1180–1219		446 0.02	297 0.03	214 0.04	151 0.08	107 0.15
1220–1259		476 0.02	317 0.03	228 0.04	161 0.07	116 0.14
1260–1299		507 0.01	338 0.02	243 0.04	172 0.07	122 0.13
1300–1339		539 0.01	359 0.02	258 0.04	183 0.07	130 0.12
1340–1379			382 0.02	274 0.04	195 0.07	138 0.12
1380–1419			403 0.02	290 0.03	206 0.06	146 0.12
1420–1459				308 0.03	218 0.06	155 0.10
1460–1500				324 0.03	230 0.06	163 0.10

[a] If register selected based on distance from register to opposite wall is unsatisfactory on account of size or pressure loss, it is permissible to shift one or more spaces left or right in the tables to obtain a more suitable register. If requirements fall in blank space, select two registers in place of one and divide CFM capacity between the two registers.

[b] Pressure loss is based on FLAT FACE ADJUSTABLE BAR TYPE and does NOT include stackhead.

[c] Values on the right of line A and A′ should not be used in applications such as churches, auditoriums, and concert halls.

[d] Values on right of line B and B′ should not be used in applications such as residential work, motion picture theaters, court rooms and schools.

[e] For floor and baseboard registers where a velocity of approximately 300 FPM is used, the free area $= \frac{CFM \times 144}{300}$ or approximately, $\frac{CFM}{2}$. Assume a pressure loss of .01.

[f] For any volume under 190 CFM use table 7.

8. Selection of Blower.
   A. Determine total cfm air delivery (the sum of all branch cfm values).
   B. Determine static pressure requirements.
      a. For furnace-blower combination units it is the sum of bonnet pressure and suction pressure.
      b. For blowers separately selected from the furnace it is the sum of bonnet pressure, filter loss, casing loss, losses through air washers, coils, and other devices.
9. Selection of Furnace.
   A. Determine register delivery (the sum of room Btu losses).
   B. Determine bonnet capacity.
      Bonnet Capacity = (total cfm) × (temperature rise) × 1.089
   C. Allowance for pick-up load.

ADJUSTMENT OF SYSTEM FOR CONTINUOUS AIR CIRCULATION

This procedure applies to the adjustment of an automatically fired forced warm-air heating system to provide continuous air circulation when the control arrangement is of the type where the room thermostat controls the fire and the blower control (fan switch) in the furnace bonnet or warm air plenum controls the blower operation. This procedure as outlined in detail in Manual No. 6 of the *National Warm Air Heating and Air Conditioning Association,* is as follows:

Table 9. Suggested Bonnet Pressure

Inches of Water

Total CFM Through Any One Duct	Suggested Bonnet Pressure In. Water	Total CMF Through Any One Duct	Suggested Bonnet Pressure In. Water
800–1000	0.10	3500 to 5000	0.15
1000–1200	0.10	5000 to 7500	0.25
1200–1800	0.10	7500 to 10,000	0.375
1800–2400	0.13	10,000 to 12,000	0.500
2400 to 3500	0.14	12,000 to 14,000	0.750

1. Adjust the fuel input in proper relation to the heat loss of the structure.
2. Determine the temperature rise through the furnace.
3. Adjust the air volume to produce a temperature rise through the furnace of about 100 deg.
4. Adjust the fan switch differential to a minimum of about 15 deg.
5. Adjust the fan switch cut-out point as low as practicable.
6. Adjust the room thermostat temperature differential to a minimum which will cause the burner to cycle frequently.
7. Balance the system by adjusting dampers to produce even temperature distribution between rooms.
8. Set the room thermostat at the desired room temperature.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the circulating basement air. A more positive cooling effect may be obtained by the use of an air washer where the temperature of the city or well water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Table 10. Pressure Drop in Duct

Inches of Water per 100 Feet of Duct Length

Equivalent Length of Duct (Ft)	Total Pressure Drop in Duct (In. of Water)																
	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17	0.18	0.19	0.20
35–44	0.10	0.13	0.15	0.18	0.20	0.23	0.25	0.28	0.30	0.33	0.35	0.38	0.40	0.43	0.45	0.48	0.50
45–54	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34	0.36	0.38	0.40
55–64	0.07	0.08	0.10	0.12	0.13	0.15	0.17	0.18	0.20	0.22	0.23	0.25	0.27	0.28	0.30	0.32	0.33
65–74	0.06	0.07	0.09	0.10	0.11	0.13	0.14	0.16	0.17	0.19	0.20	0.21	0.23	0.24	0.26	0.28	0.29
75–84	0.05	0.06	0.08	0.09	0.10	0.11	0.13	0.14	0.15	0.16	0.18	0.19	0.20	0.21	0.23	0.24	0.25
85–94	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.16	0.17	0.18	0.19	0.20	0.21	0.22
95–104	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17	0.18	0.19	0.20
105–114	0.04	0.05	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.15	0.16	0.17	0.18
115–129	0.03	0.04	0.05	0.06	0.07	0.08	0.08	0.09	0.10	0.11	0.12	0.12	0.13	0.14	0.15	0.16	0.17
130–149	0.03	0.04	0.04	0.05	0.06	0.07	0.07	0.08	0.09	0.09	0.10	0.11	0.11	0.12	0.13	0.14	0.14
150–169	0.03	0.03	0.04	0.04	0.05	0.06	0.06	0.07	0.08	0.08	0.09	0.09	0.10	0.11	0.11	0.12	0.13
170–189	0.02	0.03	0.03	0.04	0.04	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.11	0.11
190–214	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.10
215–239	0.02	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09
240–264	0.02	0.02	0.02	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.06	0.06	0.06	0.07	0.07	0.08	0.08
265–289	0.01	0.02	0.02	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.07
290–324	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.06	0.07
325–374	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.06
375–424	0.01	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05	0.05	0.05
425–474	0.01	0.01	0.01	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05
475–524	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04
525–574	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.04
575–625	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03

Equivalent Length of Duct (Ft)	Total Pressure Drop in Duct (In. of Water)																
	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.30	0.32	0.34	0.36	0.38	0.40	0.45	0.50
35–44	0.53	0.55	0.58	0.60	0.63	0.65	0.68	0.70	0.73	0.75	0.80	0.85	0.90	0.95	1.00	1.13	1.25
45–54	0.42	0.44	0.46	0.48	0.50	0.52	0.54	0.56	0.58	0.60	0.64	0.68	0.72	0.76	0.80	0.90	1.00
55–64	0.35	0.37	0.39	0.40	0.42	0.43	0.45	0.47	0.48	0.50	0.53	0.57	0.60	0.64	0.67	0.75	0.83
65–74	0.30	0.32	0.33	0.34	0.36	0.37	0.39	0.40	0.42	0.43	0.46	0.49	0.52	0.54	0.57	0.64	0.72
75–84	0.26	0.28	0.29	0.30	0.31	0.33	0.34	0.35	0.36	0.38	0.40	0.43	0.45	0.48	0.50	0.56	0.63
85–94	0.23	0.25	0.26	0.27	0.28	0.29	0.30	0.31	0.32	0.33	0.36	0.38	0.40	0.42	0.45	0.50	0.56
95–104	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.30	0.32	0.33	0.36	0.38	0.40	0.45	0.50
105–114	0.19	0.20	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.31	0.33	0.35	0.37	0.41	0.46
115–129	0.18	0.18	0.19	0.20	0.21	0.22	0.23	0.23	0.24	0.25	0.27	0.28	0.30	0.32	0.33	0.38	0.47
130–149	0.15	0.16	0.16	0.17	0.18	0.19	0.19	0.20	0.21	0.21	0.23	0.24	0.26	0.27	0.29	0.32	0.36
150–169	0.13	0.14	0.14	0.15	0.16	0.16	0.17	0.18	0.18	0.19	0.20	0.21	0.23	0.24	0.25	0.28	0.31
170–189	0.12	0.12	0.13	0.13	0.14	0.15	0.15	0.16	0.16	0.17	0.18	0.19	0.20	0.21	0.22	0.25	0.28
190–214	0.11	0.11	0.12	0.12	0.13	0.13	0.14	0.14	0.15	0.15	0.16	0.17	0.18	0.19	0.20	0.23	0.25
215–239	0.09	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.13	0.13	0.14	0.15	0.16	0.17	0.18	0.20	0.22
240–264	0.08	0.09	0.09	0.10	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.14	0.15	0.15	0.16	0.18	0.20
265–289	0.08	0.08	0.08	0.09	0.09	0.10	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.14	0.15	0.16	0.18
290–324	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.12	0.13	0.13	0.15	0.17
325–374	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.13	0.14
375–424	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.11	0.13
425–474	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.11
475–524	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.10
525–574	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.07	0.07	0.08	0.09
575–625	0.04	0.04	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.07	0.08	0.08

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface and fan capacity are necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with a warm air system and to cool the building by this method, pro-

vided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 38, 39 and 43.)

Conclusions drawn from studies[5] conducted in the University of Illinois Research Residence, subject to the limitations of the test are:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the additional frictional resistance of the coil to flow of air must be given consideration.

9. Cooling the structure by introducing large quantities of outdoor air at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure which may be used for the design of a summer cooling system in a forced-air installation is:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 6 and 15.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In University of Illinois Research Residence tests a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room.

5. Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the sensible and latent heat to be removed by the cooling unit.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter.

8. Determine pressure loss in duct system and select fan having proper capacity.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 25.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

REFERENCES

[1] Specifications for the furnace unit and the installed duct system are shown in The Yardstick, Manual No. 8, and Manual No. 7 published by *National Warm Air Heating and Air Conditioning Association.*

[2] Performance of a Forced Warm-Air Heating System as Affected by Changes in Volume and Temperature of Air Recirculated, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 393).

[3] Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

[4] Proposed Design Procedure for Large Mechanical Warm Air Heating Systems, by S. Konzo, R. J. Martin, D. S. Levinson, and R. W. Roose, A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, May, 1947, p. 108.

[5] Summer Cooling in the Research Residence, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (University of Illinois *Engineering Experiment Station Bulletins* Nos. 290, 305 and 321). A.S.H.V.E. RESEARCH REPORT No. 1177—Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P. Kratz, S. Konzo and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 203).

CHAPTER 23

STEAM HEATING SYSTEMS AND PIPING

Classification of Steam Heating Systems by Types; One-pipe; Two-pipe, Sub-atmospheric and Orifice Systems; Sizing Piping for Steam Heating Systems; Pressure Reducing Valves; Boiler Connections; Condensate Return Pumps; Vacuum Heating Pumps; Traps; Drips; Connections to Heating Units; Control Valves

STEAM heating systems may be classified according to any one of, or combination of, the following features: (1) by the piping arrangement, (2) by the pressure or vacuum conditions obtained in operation, (3) by the method of returning condensate to the boiler.

1. *By Piping Arrangement.* A steam heating system is known as a *one pipe system* when a single main serves the dual purpose of supplying steam to the heating unit and conveying condensate from it. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

A steam heating system is known as a *two-pipe system* when each heating unit is provided with two piping connections and when steam and condensate flow in separate mains and branches.

Heating systems may also be described as *up-flow* or *down-flow* depending on the direction of steam flow in the risers; and as a *dry-return* or a *wet-return* depending on whether the condensate mains are above or below the water line of the boiler or condensate receiver.

2. *By Pressure or Vacuum Conditions.* Steam heating systems may also be classified as *high pressure*, *low pressure*, *vapor*, and *vacuum* systems, depending on the pressure conditions under which the system is designed to operate.

A system is known as a *high pressure system* when the operating pressures employed are above 15 psig; as a *low pressure system* when pressures vary from 0 to 15 psig; as a *vapor system* when the system operates under both vacuum and low pressure conditions without the use of a vacuum pump; and as a *vacuum system* when the system operates under vacuum and low pressure conditions with the use of vacuum pump.

When automatic controls are employed to vary the pressure conditions in the system in accordance with outside weather conditions, the system may be known as a *sub-atmospheric*, *differential*, or *synchronized* system. These latter classifications are proprietary designations.

When orifices are employed on the inlets to the heating units the system may be known as an *orifice system.*

3. *By Method of Returning Condensate.* When condensate is returned to the boiler by gravity, the system is known as a *gravity return system.* In this system all heating units must be elevated sufficiently above the water line of the boiler, so that the condensate can flow freely to the boiler. Elevation of the heating units above the water line must therefore be sufficient to overcome pressure drops due to flow as well as pressure differences due to operation.

Referring to Fig. 1 it will be noted that the boiler and wet return form a U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction and resistance to the flow of steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of ½ psi, and utilizes a Hartford return connection instead of a check valve on the return,

the rise in the water level at the far end of the return due to the difference in steam pressure would be ⅛ of 28 in. (28 in. head being equal to one pound per square inch), or 3½ in. Adding 3 in. to overcome the resistance of the return main and 6 in. as a factor of safety for heating up gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ psi, and with a check in the return, would require ½ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

When conditions are such that condensate cannot be returned to the boiler by the action of gravity, and either traps or pumps must be employed, the system is known as a *mechanical return system.* There are three general types of mechanical condensate return devices in common use: (*a*) the alternating return trap (*b*) the condensate return pump, and (*c*) the vacuum return pump.

In systems where pressure conditions in the system vary between that of a gravity return and a forced return system, a boiler return trap or alternating receiver is employed and the system may be known as an *alternating return system.*

When condensate is pumped to the boiler under pressures of the atmosphere or above, the system is known as a *condensate pump return system.*

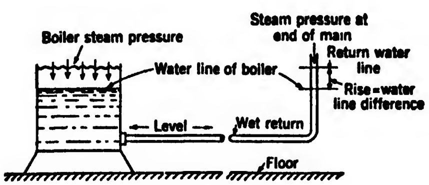


FIG. 1. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

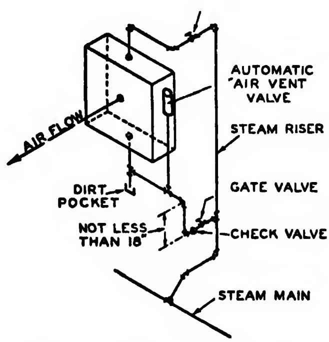


FIG. 2. TYPICAL TWO-PIPE CONNECTIONS TO UNIT HEATERS IN ONE-PIPE AIR VENT SYSTEMS

When condensate is pumped to the boiler under vacuum conditions, the system is known as a *vacuum pump return system.*

In either the *condensate* or *vacuum pump systems* it is highly desirable to arrange for gravity flow to a receiver and to the pump. The pump then forces condensate into the boiler against its pressure.

ONE-PIPE SYSTEMS

One-pipe systems, as previously defined, are systems in which steam and condensate flow in the same pipe. Radiators and other heating units, in general, have only one piping connection from main to unit, although it is possible to employ two connections to the same main as indicated in Fig. 2. Unit heaters in one-pipe systems may also have separate connections to the wet return as shown in Fig. 5 Chapter 26.

There are several variations in the piping arrangement of a one-pipe system as follows:

1. Up-feed one-pipe systems where the radiators and other heating units are located above the supply mains. The mains in this instance convey both steam and condensate. Such a system is illustrated in Fig. 3. Typical connections to radiator or risers are illustrated in Fig. 4 and method of changing sizes of mains in Fig. 5.

2. Up-feed one-pipe systems where the radiators and other heating units are located above the mains and the mains are dripped at each radiator connection to a wet

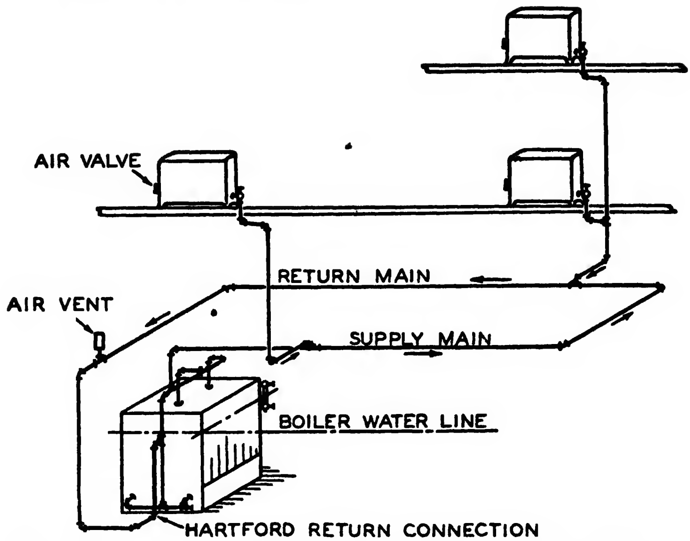


FIG. 3. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

return, so that the steam main carries a minimum of the condensate. This system is illustrated in Fig. 6. Typical connections to radiators and risers are illustrated in Fig. 7. Up-feed systems are not recommended for systems higher than four stories.

3. Down-feed one-pipe systems, where the radiators and other heating units are located below the supply main. In this arrangement only risers and connections to heating units convey both steam and condensate, and both are flowing in the same direction. The steam main is kept relatively free of condensate by dripping through the drop risers.

Each radiator or heating unit in a one-pipe system must be supplied with a thermostatic air valve which functions to relieve air from the heating unit under pressure, and to close when steam itself heats the thermostatic element of the valve.

To improve steam circulation in one-pipe systems quick vent air valves should be provided at the ends and at intermediate points where the steam main is brought to a higher elevation or where dropped below the water line. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 6, to prevent possible damage to their mechanisms by water.

Air valves are of two general types, the pressure and the vacuum types. The pressure type permits the inflow of atmospheric air to the system when the steam pressure in the system falls below atmospheric pressure. The vacuum type, which contains a small check valve, prevents the air from flowing back to the system and thereby maintains vacuum conditions in the system, and a consequent evaporation or generation of steam

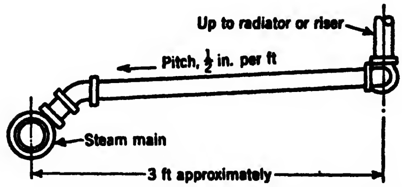


FIG. 4. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

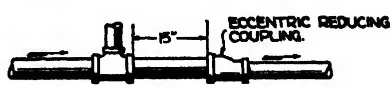


FIG. 5. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

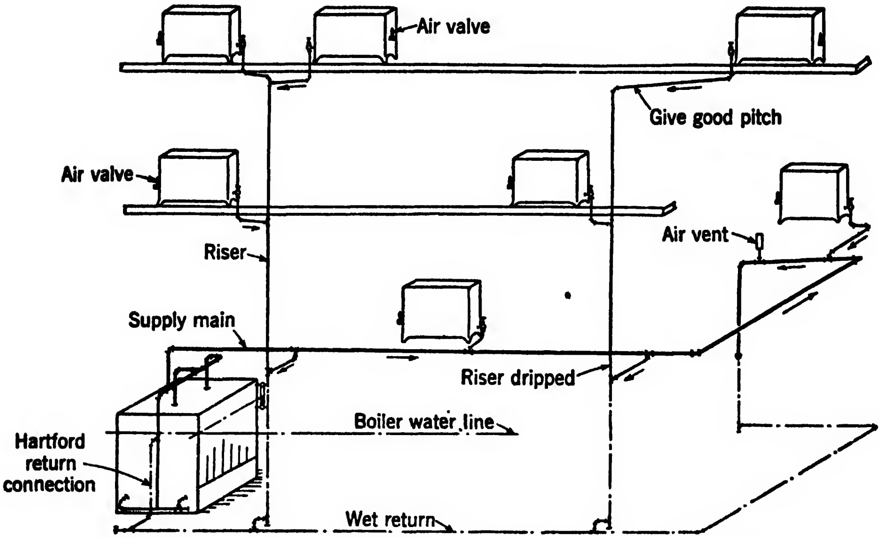


FIG. 6. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

or vapor at sub-atmospheric pressures, and at consequent lower temperatures. Systems which use vacuum valves are known as *vapor* or *vacuum one-pipe systems*. The vapor or vacuum systems will maintain a more uniform temperature condition than the pressure systems.

Each heating unit in a one-pipe system may also be provided with a valve on the connection to the unit, although this is not essential except to shut the unit off in case it is not desired for heating. Valves on one-pipe systems must be either fully opened or fully closed. No throttling or modulating position can be maintained, since if a valve is partially closed condensate will not drain from the unit. This condition is dangerous because it may create a low water condition in the boiler with consequent burning or cracking of the boiler, or of creating a hazard due to the freezing of the water logged heating unit itself.

TWO-PIPE SYSTEMS

Two-pipe systems, as previously defined, are systems in which steam and condensate flow in separate pipes. Two-pipe systems operate under either high pressure, low pressure, vapor, or vacuum conditions. Either the up-flow or the down-flow arrangement of mains may be employed.

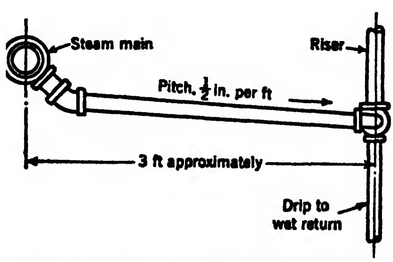


FIG. 7. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

Two-Pipe High Pressure Systems

Two-pipe high pressure systems operate at pressures above 15 psig, usually from 30 to 150 psig. Large industrial type buildings, which are equipped with unit heaters or large built up fan units, usually use high pressure steam systems.

Fig. 8 illustrates a typical high pressure system. Because of the high pressures and the great differential between steam and return mains, it is possible to locate returns above the heating units and lift the condensate to these returns.

The condensate can be flashed into steam in low pressure mains if any are available or passed through an economizer heater before being discharged to a vented receiver. It is of course necessary to provide for the elimination of air from high pressure systems, the same as in low pressure systems.

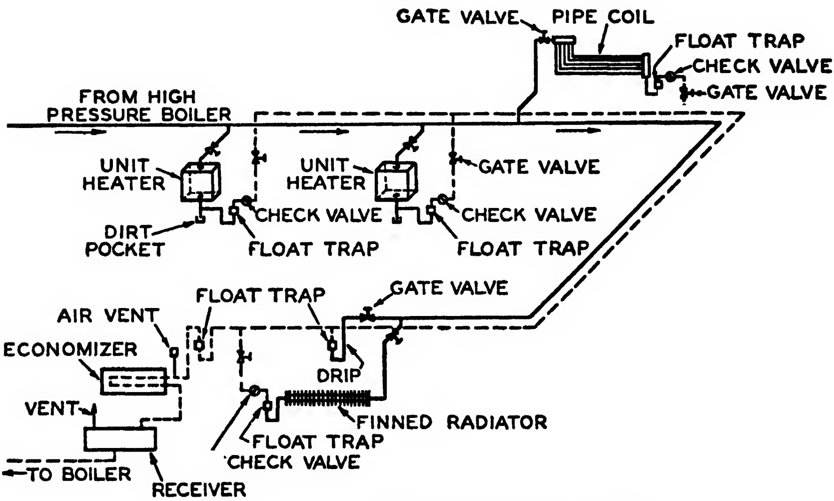


Fig. 8. Typical High Pressure Heating System

Return traps used on high pressure systems are usually of the bucket, inverted bucket, float or impulse type.

Two-Pipe Low Pressure Systems

Low pressure systems operate at pressures of 0 to 15 psig. The piping arrangement of both up-feed and down-feed low pressure systems is identical with those of two-pipe vapor systems described in the following section. The only difference between the two systems is in the type of air valve used. The air valves used in low pressure systems do not contain the check discs and hence the system cannot operate under vacuum conditions. The low pressure systems are not as popular as the vapor systems, because they have the disadvantage of not holding heat when the rate of steam generation is diminishing. They also have the disadvantage of corroding to a greater extent than vapor systems due to the continued presence of new air in the system.

Low pressure systems have the advantage however of returning condensate to the boiler readily and not retaining it in the piping as may be possible in vapor systems.

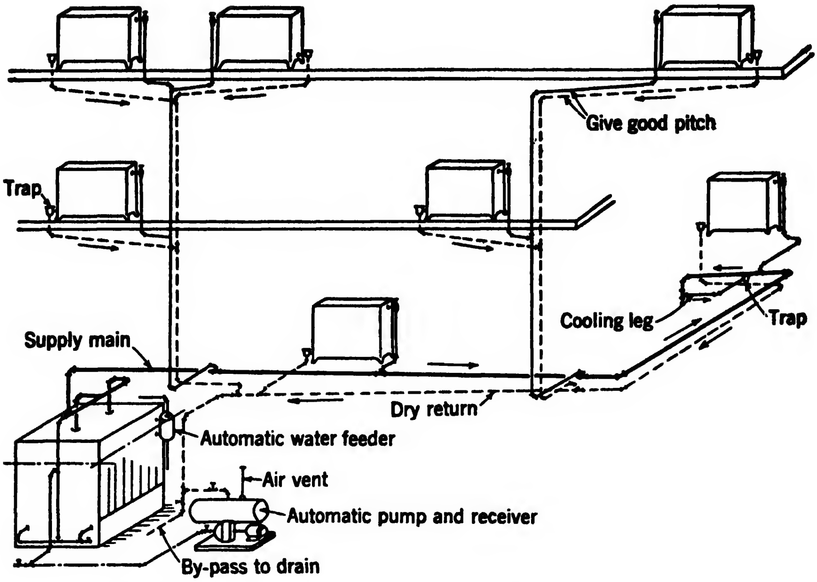


FIG. 9. TYPICAL INSTALLATION USING CONDENSATE PUMP

Fig. 9 illustrates a typical low pressure system with condensate pump.

Two-Pipe Vapor Systems

Two-pipe vapor systems operate at pressures varying from 30 in. vacuum to 15 psig without the use of a vacuum pump. A typical two-pipe up-feed vapor system is shown in Fig. 10, and a typical two-pipe down-feed system is illustrated in Fig. 11. The method of dripping drop risers in a down-feed system is illustrated in Fig. 12. Radiators discharge their condensate and air through thermostatic traps to the dry return main. Air is eliminated, when the system is under pressure, at the ends of the supply and return

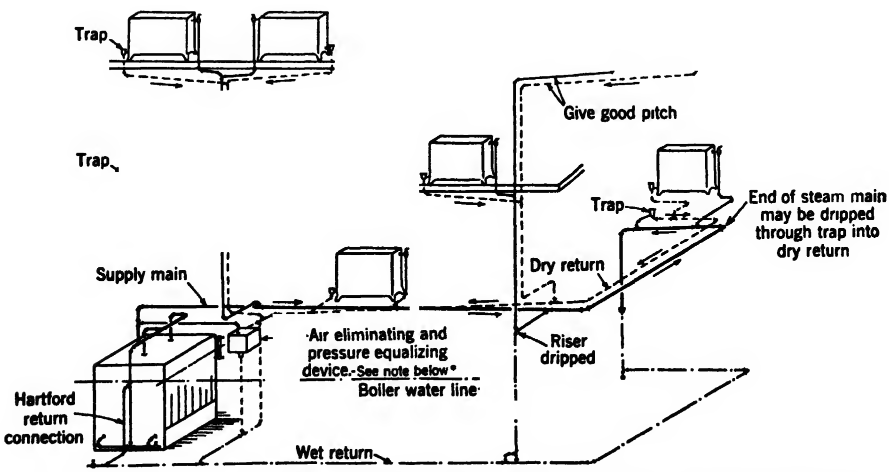


FIG. 10. TYPICAL UP-FEED TWO-PIPE SYSTEM WITH AUTOMATIC RETURN TRAP*

* Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

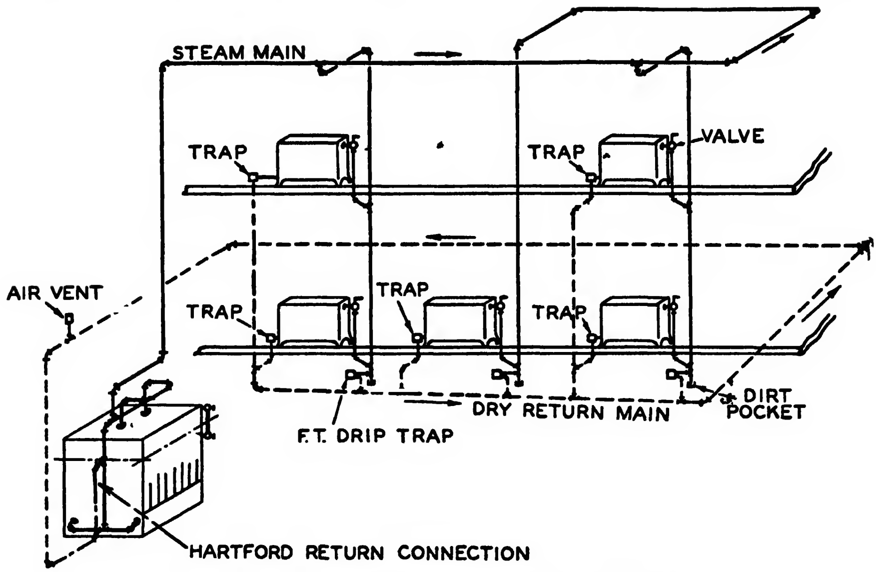


FIG. 11. TYPICAL DOWN-FEED TWO-PIPE SYSTEM

mains just before they drop to the wet return. The air valves are of the float and thermostatic type which opens when cool air contracts the thermostatic element and closes when steam expands the element. The float element of the valve closes the valve when, due to pressure differences, water rises to the point of overflow in the main. The air valves are also provided with a small check disc which closes to prevent the inflow of air to the system when the pressure drops below atmospheric pressure. This enables the system to operate under vacuum conditions at lower steam temperatures for a period of four to eight hours depending on the tightness of the system.

Vapor systems may also be provided with an automatic return trap or alternating receiver which automatically returns condensate to the boiler

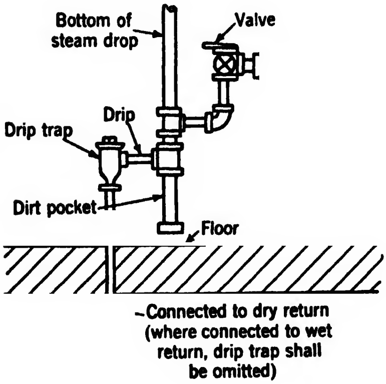


FIG. 12. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

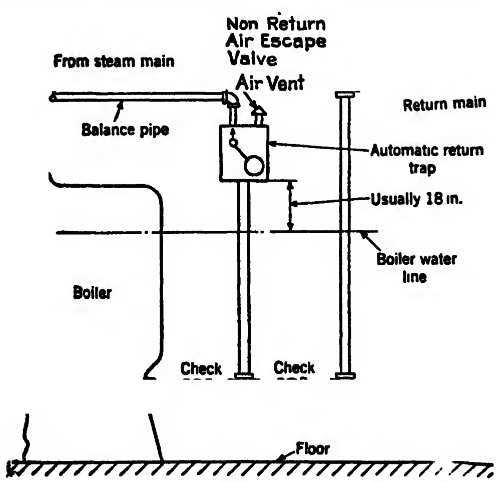


FIG. 13. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

when the boiler is steaming under pressure conditions which would prevent the return of condensate by gravity. The typical connections for an automatic return trap are illustrated in Fig. 13.

Each heating unit in a vapor system, as in all two-pipe systems, is provided with a graduated or modulating valve which permits the control of heat in the radiator by varying the opening of the valve.

Two-Pipe Vacuum Systems

Vacuum systems operate under conditions of both low pressure and vacuum, but employ the use of a vacuum pump to insure maintenance of sub-atmospheric pressures.

A typical two-pipe up-feed vacuum system is illustrated in Fig. 14, and a down-feed arrangement in Fig. 15.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum

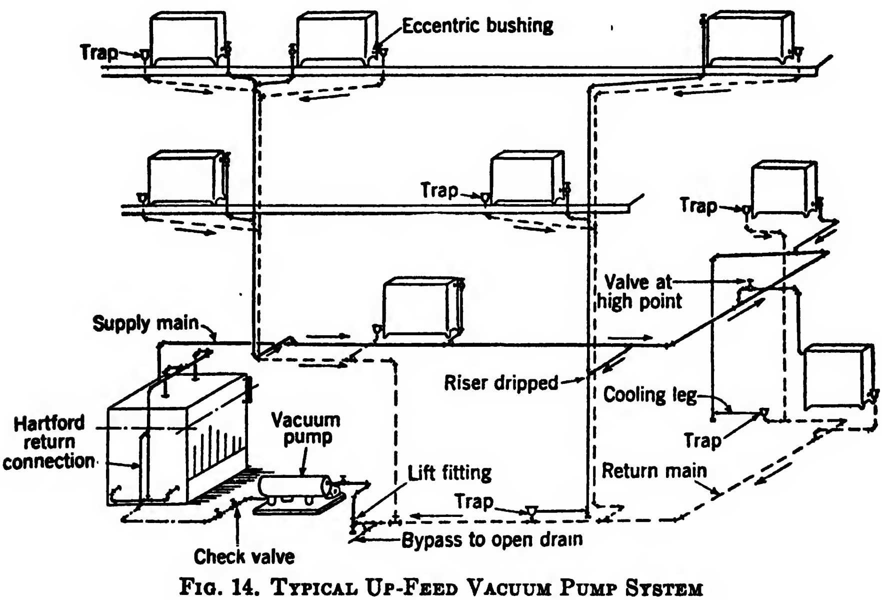


Fig. 14. Typical Up-Feed Vacuum Pump System

pump withdraws the air and water from the system, separates the air from the water and expels it to atmosphere and pumps the water back to the boiler, or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height, by the suction of the vacuum pump, by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the amount of vacuum maintained. It is preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a

float control for the pump, at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 16. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 17. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate danger of freezing.

TWO-PIPE SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat

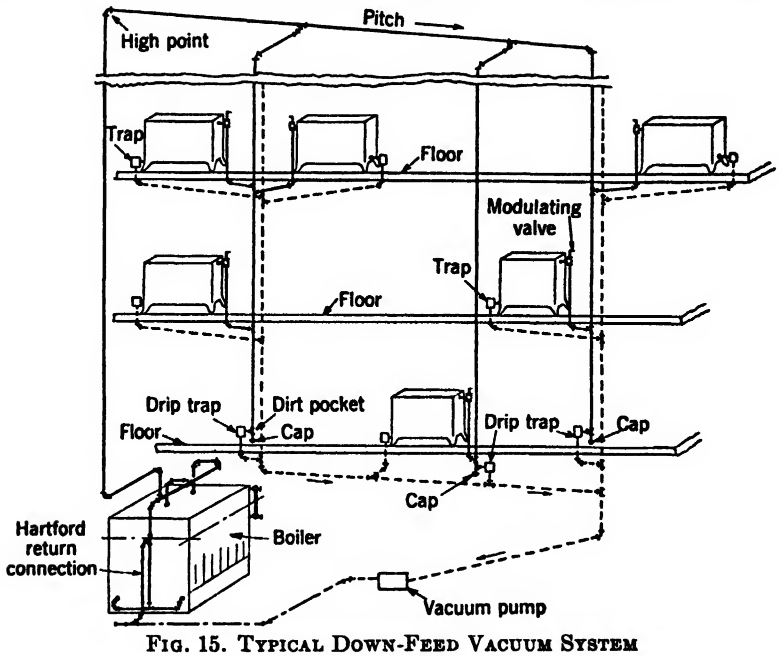


FIG. 15. TYPICAL DOWN-FEED VACUUM SYSTEM

output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and specific volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg, after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler.

The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. In some systems radiator supply valves include adjustable orifices or are equipped with regulating orifice plates. The sizes of orifices used are larger than for other types of orifice systems because for equal radiator sizes the volume flowing is larger. Orifices are omitted on some systems. Radiator traps and drips are designed to operate at any pressure from 15 psig to 26 in. Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self-induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump.

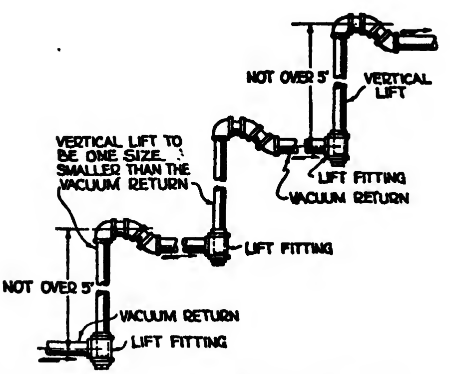


Fig. 16. Method of Making Lifts on Vacuum Systems when Distance is Over 5 ft

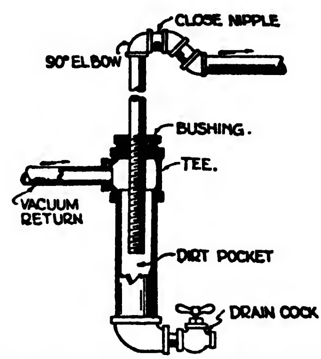


Fig. 17. Detail of Main Return Lift at Vacuum Pump

One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump and equipped with float control so that the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Some applications of sub-atmospheric systems are proprietary.

TWO-PIPE ORIFICE SYSTEMS

Orifice steam heating systems may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic traps but use thermostatic or combination float and thermostatic traps on all drip points. A return condensate pump with receiver vented to

atmosphere, a return line vacuum pump, or a return trap, is generally used to return the condensate to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure differential maintained.

The principle on which these systems operate is based on the fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in size as to exactly fill a radiator with 2 psig on one side and $\frac{1}{4}$ psig on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$

Should the steam pressure be dropped to $\frac{1}{4}$ psig on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portions of the radiator down to the point where the main pressure equals the back pressure in the radiator provided the supply pipe pressures may be controlled sufficiently close. If orifices are designed on a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main, which maintains a determined pressure by varying the vacuum in the return lines, or by varying the pressure in the supply lines and the vacuum in the returns. The valves are frequently manually set from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outside and inside temperatures. Since the range through which the pressures may be varied is usually from 0 to 4 psig, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

A recommended orifice schedule is shown in Table 1. Some systems use orifices not only in radiator inlets but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure was 3 psig and the pressure at the end of the main was 2 psig, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced. Some orifice systems are proprietary.

SIZING PIPING FOR STEAM HEATING SYSTEMS

The functions of the piping system are the distribution of the steam, the return of the condensate and, in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be

Table 1. Orifice Capacities for Low Pressure Steam Systems

This table is based on data from actual tests[a]

Orifice Diameter 64ths of an Inch	6 in. Hg Differential	5 in. Hg Differential	4 in. Hg Differential	2 in. Hg Differential	1 in. Hg Differential
	Capacity Expressed in Square Feet E D R				
7	18–23	16–21	15–19	10–13	
8	23–29	21–27	19–25	13–17	8–11
9	29–36	27–33	25–30	17–21	11–14
10	36–44	33–40	30–37	21–26	14–17
11	44–52	40–48	37–44	26–31	17–20
12	52–62	48–57	44–51	31–37	20–24
13	62–72	57–66	51–59	37–43	24–28
14	72–83	66–76	59–67	43–49	28–32
15	83–94	76–86	67–76	49–56	32–37
16	94–106	86–97	76–86	56–64	37–42
17	106–119	97–109	86–97	64–72	42–47
18	119–133	109–122	97–108	72–80	47–52
19	133–148	122–135	108–120	80–88	52–58
20	148–163	135–149	120–133	88–98	58–64
21	163–179	149–164	133–145	98–107	64–71
	Capacity Expressed in Pounds per Hour				
7	4.5–5.8	4.0–5.3	3.8–4.8	2.5–3.3	
8	5.8–7.3	5.3–6.8	4.8–6.3	3.3–4.3	2.0–2.8
9	7.3–9.0	6.8–8.3	6.3–7.5	4.3–5.3	2.8–3.5
10	9.0–11.0	8.3–10.0	7.5–9.3	5.3–6.5	3.5–4.3
11	11.0–13.0	10.0–12.0	9.3–11.0	6.5–7.8	4.3–5.0
12	13.0–15.5	12.0–14.3	11.0–12.8	7.8–9.3	5.0–6.0
13	15.5–18.0	14.3–16.5	12.8–14.8	9:3–10.8	6.0–7.0
14	18.0–20.8	16.5–19.0	14.8–16.8	10.8–12.3	7.0–8.0
15	20.8–23.5	19.0–21.5	16.8–19.0	12.3–14.0	8.0–9.3
16	23.5–26.5	21.5–24.3	19.0–21.5	14.0–16.0	9.3–10.5
17	26.5–29.8	24.3–27.3	21.5–24.3	16.0–18.0	10.5–11.8
18	29.8–33.3	27.3–30.5	24.3–27.0	18.0–20.0	11.8–13.0
19	33.3–37.0	30.5–33.8	27.0–30.0	20 0–22.0	13.0–14.5
20	37.0–40.8	33.8–37.3	30.0–33.3	22.0–24.5	14.5–16.0
21	40.8–44.8	37.3–41.0	33.3 36.3	24.5–26.8	16.0–17.8

Note.—The radiator orifice plates recommended in this table are made of brass stampings 0.023 in. thick cup-shaped to be inserted in radiator valve unions.

[a] Flow of Steam Through Orifices into Radiators, by S S. Sanford and C. B. Sprenger (A S.H.V.E. Transactions, Vol. 37, 1931, p. 371).

rapid, uniform and without noise, and the release of air should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement it is desirable to maintain equivalent resistances in the supply and return piping to and from a radiator. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler tends to obtain such a result. The condensate which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive re-vaporization, such as occurs where condensate is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

It is important that steam piping systems distribute steam not only at full *design* load but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside

temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the maximum operating load for severe weather due to the necessity of raising the temperature of the metal in the system to the steam temperature and the building to the *design* indoor temperature. Investigations of the return of condensate have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under constant conditions as they are continually changing due to variation in load. As the system is being filled with steam the pressures existing in various locations may be different from those which exist for appreciable periods at other locations although at equilibrium conditions the pressures are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air permit uniform

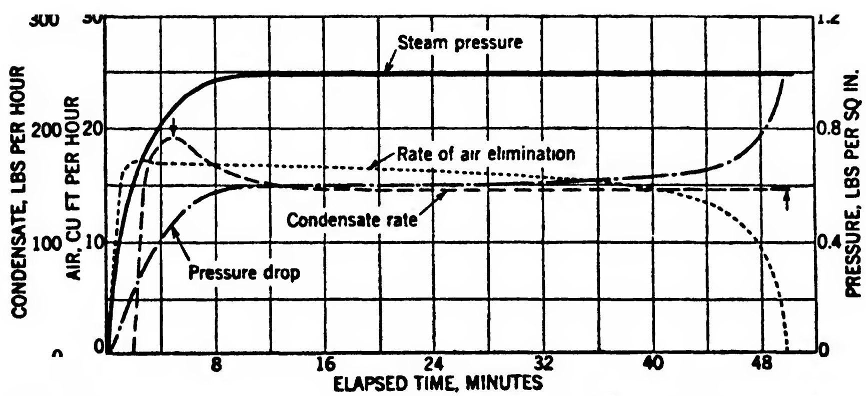


FIG. 18. RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 18 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming-up period when the rate of air elimination and condensation is high are clearly indicated in these curves.

It is evident that the condensate flow during the initial warming-up period reaches a peak which is greater than the constant condensing rate eventually reached when the pressure becomes uniform. Moreover, the peak condensing rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

Steam Flow

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship

Table 2. Flow of Steam in Pipes

P = loss in pressure in pounds per square inch.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367$$

Pressure Loss in Ounces	Col. 1	Pipe Size		Internal Area of Pipe Sq Inches	Col. 2	Avg Steam Press. psig	Col. 3	Length of Pipe in Feet	Col. 4
	$5220\sqrt{\frac{P}{100}}$	Nominal	Actual Internal Diameter		$\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$		$\sqrt{d}$		$\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2.036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12.730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60.3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0.557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200.3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition.

Example 1: 1 oz drop — 2 in. pipe — 1.3 lb press. — 100 ft equivalent length:

130.5 × 3.710 × 0.201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 2 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a] Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.
[b] The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

has been established by Babcock in the formula given at the top of Table 2. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

Pipe Sizes

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and at the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. The direction of flow of the condensate, whether against or with the steam.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered such as initial pressure and pressure required at the end of the line, but it is most important that: (1) the total pressure drop does not exceed the initial gage pressure of the system and in actual practice it should never exceed one-half of the initial gage pressure; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric, which normally operate under controlled partial vacua and orifice and vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the rise in water due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry return, and the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial gage pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Table 3 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensate present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and

the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Reaming Important

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided. The second is the care used in reaming the ends of the pipe after cutting. The effect of both of these factors increases as the

TABLE 3. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft. Velocity in Ft per Sec

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.
	Capacity Expressed in Square Feet E D R															
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30
	Capacity Expressed in Pounds per Hour															
¾	6.3	12	7.6	14	9.3	18	10.1	19	10.6	20	11.5	21	11.9	22	12.3	23
1	11.5	12	13.2	15	15.8	17	17.5	20	18.8	22	20.8	23	22.0	25	22.6	26
1¼	26.2	18	29.3	20	33.3	23	36.1	25	38.5	27	41.3	28	43.2	29	44.6	31
1½	35.7	18	39.8	21	45.3	23	49.1	25	52.3	27	56.0	28	58.7	30	60.7	31
2	59.0	19	65.9	20	74.9	23	81.4	25	86.6	27	92.4	28	97.1	29	100.3	30

[a] Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

pipe size decreases. According to A.S.H.V.E. Research Laboratory tests, either of these factors may affect the capacity of a 1-in. pipe as much as 20 per cent. The third factor is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 4 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent*

length of run as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it may be necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLES FOR PIPE SIZING FOR LOW PRESSURE SYSTEMS[2]

Tables 5, 6, and 7 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation (*abbreviated EDR*) per hour. The drops indicated are

TABLE 4. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

Size of Pipe Inches	Length in Feet to be Added to Run				
	Standard Elbow	Side Outlet Tee	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

[a]Valve in full open position.

Example of length in feet of pipe to be added to actual length of run.

Measured Length	=	132.0 ft
4 in. Gate Valve	=	1.9 ft
4—4 in. Elbows	=	36.0 ft
Equivalent Length	=	169.9 ft

drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed and without unusual or noticeable defects.

Table 5 may be used for sizing piping for steam heating systems by pre-determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensate flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensate flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve and vertical connection sizes, and radiator and runout sizes, all of which are based on the

Table 5. Steam Pipe Capacities for Low Pressure Systems
(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size In.	Capacities of Steam Mains and Risers — Direction of Condensate Flow in Pipe Line — With the Steam in One-Pipe and Two-Pipe Systems: 1/32 psi or 1/2 Oz Drop	1/24 psi or 3/4 Oz Drop	1/16 psi or 1 Oz Drop	1/8 psi or 2 Oz Drop	1/4 psi or 4 Oz Drop	1/2 psi or 8 Oz Drop	Against the Steam Two-Pipe Only: Vertical	Horizontal	Special Capacities for One-Pipe Systems Only: Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
Capacity Expressed in Square Feet E D R											
3/4	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	34	45	28	28
1 1/4	87	100	122	173	245	346	122	75	98	62	62
1 1/2	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2 1/2	449	518	635	898	1,270	1,800	635	395	464	....	260
3	822	948	1,160	1,650	2,330	3,290	1,130	700	800	....	475
3 1/2	1,230	1,420	1,740	2,460	3,470	4,910	1,550	1,150	1,140	....	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2,040	1,700	1,520	....	1,110
5	3,210	3,710	4,550	6,430	9,090	12,900	3,600	3,150	....	....	2,180
6	5,280	6,100	7,460	10,550	14,900	21,100	5,800	5,520	....	....	....
8	11,000	12,700	15,500	21,970	31,070	43,900	12,800	13,200	....	....	....
10	20,000	23,100	28,300	40,100	56,700	80,200	21,200	24,000	....	....	....
12	32,000	37,100	45,500	64,300	91,000	129,000	32,000	40,000	....	....	....
16	61,000	69,700	84,800	121,000	170,000	242,000	68,000	92,000	....	....	....
Capacity Expressed in Pounds per Hour											
3/4	....	....	8	....	....	....	8	....	6	....	7
1	10	12	14	20	28	40	14	9	11	7	7
1 1/4	22	25	31	43	61	87	31	19	20	16	16
1 1/2	34	39	48	67	95	135	48	27	38	23	23
2	68	79	97	137	193	273	97	49	72	42	42
2 1/2	112	130	159	225	318	449	159	99	116	....	65
3	206	237	291	411	581	822	282	175	200	....	119
3 1/2	307	355	434	614	869	1,230	387	288	286	....	186
4	435	503	614	869	1,230	1,740	511	425	380	....	278
5	806	928	1,140	1,610	2,270	3,210	900	788	....	....	545
6	1,320	1,520	1,870	2,640	3,730	5,280	1,450	1,380	....	....	....
8	2,750	3,170	3,880	5,490	7,770	11,000	3,200	3,300	....	....	....
10	5,010	5,790	7,090	10,000	14,200	20,000	5,300	6,000	....	....	....
12	8,040	9,290	11,400	16,100	22,700	32,200	8,000	10,000	....	....	....
16	15,100	17,400	21,200	30,300	42,400	60,500	17,000	23,000	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—Steam at an average pressure of 1 psig is used as a basis for calculating capacities. All drops shown are in psi per 100 ft of equivalent run—based on pipe properly reamed.

[a] Do not use Column *H* for drops of 1/24 or 1/32 psi, substitute Column *C* or Column *B* as required.

[b] Do not use Column *J* for drop 1/32 psi except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c] On radiator runouts over 8 ft long increase one pipe size over that shown in Table 4.

critical velocities of the steam to permit the counter flow of condensate without noise.

Return piping may be sized with the aid of Tables 6 and 7 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 5. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-psig?

Solution. It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 psi or less. With a pressure drop of 1 psi and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ psi, while if the total drop were $\frac{1}{2}$ psi, the drop per 100 ft would be $\frac{1}{20}$ psi. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ psi per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ psi. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

TABLES FOR PIPE SIZING FOR HIGH PRESSURE SYSTEMS

Many of the recent installations of heating systems for large industrial type buildings have been designed for the use of high pressure steam, that is, without the use of pressure reducing valves. Such systems usually involve the use of unit heaters or large built-up fan units with blast heating coils. Pressures on these systems vary from 30 to 150 psi. Temperatures are controlled by a modulating or throttling type thermostatic valve controlled by the air temperature in the room, fan inlet or outlet.

Tables 8 to 11 may be used for the sizing of steam and return piping for systems of 30 and 150 psi pressure at various pressure drops. These tables are based on Babcock's formula, and have been used as the basis of design for a number of years.

SIZING PIPING FOR ONE-PIPE GRAVITY SYSTEMS

Gravity one-pipe air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized by means of Tables 5, 6 and 7 as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-psi drop (Column *D*).
2. *Where the riser runouts are not dripped* and the steam and condensate flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.
3. *For up-feed steam risers* carrying condensate back from the radiators, use Column *J*.
4. *For down-feed systems, the main risers* of which do not carry any radiator condensate, use Column *H*.
5. *For the radiator valve size and the stub connection,* use Column *K*.
6. For the *dry return main,* use Column *U*.
7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ psi. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{24}$-psi drop is being used, the steam main and dripped runouts would be

Table 6. Return Pipe Capacities for Low Pressure Systems

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size Inches	Capacity of Return Mains and Risers																	
	1/32 Psi or ½ Oz Drop per 100 Ft			1/24 Psi or ⅔ Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			⅛ Psi or 2 Oz Drop per 100 Ft			¼ Psi or 4 Oz Drop per 100 Ft			½ Psi or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	Mains																	
¾	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,980
1¼	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	......	......	3,390
1½	1,350	822	......	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,260	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2½	4,700	3,040	......	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,900
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	......	......	30,200
3½	11,000	7,880	......	13,200	8,800	13,000	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	......	......	62,190
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,900	......	......	77,400	......	......	109,000
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,000
	Risers																	
¾	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,980
1	......	450	......	......	450	976	......	450	1,200	......	450	1,700	......	450	2,400	......	......	3,390
1¼	......	990	......	......	990	1,550	......	990	1,900	......	990	2,700	......	990	3,800	......	......	5,370
1½	......	1,500	......	......	1,500	3,260	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,450	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,900
2½	......	......	......	......	......	8,710	......	......	10,700	......	......	15,200	......	......	21,400	......	......	30,200
3	......	......	......	......	......	13,000	......	......	16,000	......	......	22,700	......	......	32,000	......	......	45,200
3½	......	......	......	......	......	17,900	......	......	22,000	......	......	31,200	......	......	44,000	......	......	62,200
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,900	......	......	77,400	......	......	109,000
5	......	......	......	......	......	50,500	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,000

Table 7. Return Pipe Capacities for Low Pressure Systems

Capacity Expressed in Pounds per Hour

(Reference to this table will be made by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size Inches	Capacity of Return Mains and Risers																	
	1/32 Psi or 1/2 Oz Drop per 100 Ft			1/24 Psi or 2/3 Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			1/8 Psi or 2 Oz Drop per 100 Ft			1/4 Psi or 4 Oz Drop per 100 Ft			1/2 Psi or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
Mains																		
3/4	....	....	....	....	....	42	....	....	100	....	....	142	....	....	200	....	....	283
1	125	62	....	145	71	143	175	80	175	250	103	249	350	115	350	....	....	494
1 1/4	213	130	....	248	149	244	300	168	300	425	217	426	600	241	600	....	....	848
1 1/2	338	206	....	393	236	388	475	265	475	675	340	674	950	378	950	....	....	1,340
2	700	470	....	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	....	....	2,830
2 1/2	1,180	760	....	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	....	....	4,730
3	1,880	1,460	....	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	....	....	7,560
3 1/2	2,750	1,970	....	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	....	....	11,300
4	3,880	2,930	....	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	....	....	15,500
5	....	....	....	....	....	7,880	....	....	9,680	....	....	13,700	....	....	19,400	....	....	27,300
6	....	....	....	....	....	12,600	....	....	15,500	....	....	22,000	....	....	31,000	....	....	43,800
Risers																		
3/4	....	48	....	....	48	143	....	48	175	....	48	249	....	48	350	....	....	494
1	....	113	....	....	113	244	....	113	300	....	113	426	....	113	600	....	....	848
1 1/4	....	248	....	....	248	388	....	248	475	....	248	674	....	248	950	....	....	1,340
1 1/2	....	375	....	....	375	815	....	375	1,000	....	375	1,420	....	375	2,000	....	....	2,830
2	....	750	....	....	750	1,360	....	750	1,680	....	750	2,380	....	750	3,350	....	....	4,730
2 1/2	....	....	....	....	....	2,180	....	....	2,680	....	....	3,800	....	....	5,350	....	....	7,560
3	....	....	....	....	....	3,250	....	....	4,000	....	....	5,680	....	....	8,000	....	....	11,300
3 1/2	....	....	....	....	....	4,480	....	....	5,500	....	....	7,810	....	....	11,000	....	....	15,500
4	....	....	....	....	....	7,880	....	....	9,680	....	....	13,700	....	....	19,400	....	....	27,300
5	....	....	....	....	....	12,600	....	....	15,500	....	....	22,000	....	....	31,000	....	....	43,800

TABLE 8. STEAM PIPE CAPACITIES FOR 30 PSIG STEAM SYSTEMS[a]
Capacity Expressed in Pounds per Hour
(Steam and Condensate Flowing in Same Direction)

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length					
	1/8	1/4	1/2	3/4	1	2
3/4	15	22	31	38	45	63
1	31	46	63	77	89	125
1 1/4	69	100	141	172	199	281
1 1/2	107	154	219	267	.309	437
2	217	313	444	543	627	886
2 1/2	358	516	730	924	1,030	1,460
3	651	940	1,330	1,630	1,880	2,660
3 1/2	979	1,410	2,000	2,450	2,830	4,000
4	1,390	2,000	2,830	3,460	4,000	5,660
5	2,560	3,640	5,230	6,400	7,390	10,500
6	4,210	6,030	8,590	10,400	12,100	17,200
8	8,750	12,600	17,900	21,900	25,300	35,100
10	16,300	23,500	33,200	40,600	46,900	66,400
12	25,600	36,900	52,300	64,000	74,000	104,500

[a] Note: Steam at an average pressure of 30 psig is used as the basis for calculating the above table.

sized from Column *C*; radiator runouts and undripped riser runouts from Column *L*; up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{24}$ psi); the dry return from Column *R*; and the wet return from Column *Q*.

With a $\frac{1}{32}$-psi drop the sizing would be the same as for $\frac{1}{24}$ psi except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per foot. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

TABLE 9. STEAM PIPE CAPACITIES FOR 150 PSIG STEAM SYSTEMS[a]
Capacity Expressed in Pounds per Hour
(Steam and Condensate Flowing in Same Direction)

Pipe Size Inches	Drop in Pressure—Psi per 100 Ft in Length						
	1/8	1/4	1/2	3/4	1	2	5
3/4	29	41	58	71	82	116	184
1	58	82	117	143	165	233	369
1 1/4	130	185	262	320	370	523	827
1 1/2	203	287	407	497	575	813	1,290
2	412	583	825	1,010	1,170	1,650	2,600
2 1/2	683	959	1,360	1,650	1,920	2,710	4,290
3	1,240	1,750	2,480	3,020	3,500	4,940	7,820
3 1/2	1,860	2,630	3,720	4,550	5,250	7,420	11,700
4	2,630	3,720	5,260	6,430	7,430	10,500	16,600
5	4,860	6,880	9,730	11,900	13,800	19,500	30,800
6	7,960	11,300	16,000	19,500	22,600	31,900	50,400
8	16,600	23,500	33,200	40,600	47,000	66,400	105,000
10	30,800	43,400	61,700	75,600	87,300	123,000	195,000
12	48,600	68,800	97,300	119,000	138,000	194,000	307,500

[a] Note: Steam at an average pressure of 150 psig is used as the basis for calculating the above table.

TABLE 10. RETURN PIPE CAPACITIES FOR 30 PSIG STEAM SYSTEMS[a]
Capacity Expressed in Pounds per Hour

PIPE SIZE INCHES	DROP IN PRESSURE—POUNDS PER 100 FT IN LENGTH				
	1/8	1/4	1/2	3/4	1
3/4	115	170	245	308	365
1	230	340	490	615	730
1 1/4	485	710	1,025	1,290	1,530
1 1/2	790	1,160	1,670	2,100	2,500
2	1,580	2,360	3,400	4,300	5,050
2 1/2	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,300	12,900	15,300
3 1/2	7,200	10,600	15,300	19,200	22,800
4	10,200	15,000	21,600	27,000	32,300
5	19,000	27,800	40,300	55,500	60,000
6	31,000	45,500	65,500	83,000	98,000

[a] Note: The above table is based on steam at pressures of 0 to 4 psig.

4. Supply mains, runouts to risers, or risers, should be dripped where necessary.
5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

Example 3. Size the one-pipe gravity steam system shown in Fig. 19 assuming that this is all there is to the system or that the riser and main shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ psi the drop per 100 ft will be slightly less than $\frac{1}{16}$ psi. It would be well in this case to use $\frac{1}{24}$ psi, and this would result in the theoretical sizes indicated in Table 12. These theoretical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

SIZING PIPING FOR ONE-PIPE VAPOR SYSTEMS

Piping for one-pipe vapor systems is sized so as to permit only a few ounces pressure drop in the system. Otherwise, the method follows that outlined for sizing one-pipe gravity systems.

TABLE 11. RETURN PIPE CAPACITIES FOR 150 PSIG STEAM SYSTEMS[a]
Capacity Expressed in Pounds per Hour

PIPE SIZE INCHES	DROP IN PRESSURE—PSI PER 100 FT IN LENGTH					
	1/8	1/4	1/2	3/4	1	2
3/4	156	232	360	465	560	890
1	313	462	690	910	1,120	1,780
1 1/4	650	960	1,500	1,950	2,330	3,700
1 1/2	1,070	1,580	2,460	3,160	3,800	6,100
2	2,160	3,300	4,950	6,400	7,700	12,300
2 1/2	3,600	5,350	8,200	10,700	12,800	20,400
3	6,500	9,600	15,000	19,500	23,300	37,200
3 1/2	9,600	14,400	22,300	28,700	34,500	55,000
4	13,700	20,500	31,600	40,500	49,200	78,500
5	25,600	38,100	58,500	76,000	91,500	146,000
6	42,000	62,500	96,000	125,000	150,000	238,000

[a] Note: The above table is based on steam at pressures of 1 to 20 psig.

Table 12. Pipe Sizes for One-Pipe Up-Feed System Shown in Fig. 19

Part of System	Section of Pipe	Radiation Supplied EDR Sq Ft	Theoretical Pipe size (Inches)	Practical Pipe size (Inches)
Branches to radiators..	....	100	2	2
Branches to radiators..	....	50	1¼	1¼
Riser..........	*a* to *b*	200	2	2
Riser..........	*b* to *c*	300	2½	2½
Riser..........	*c* to *d*	400	2½	2½
Riser..........	*d* to *e*	500	3	3
Riser..........	*e* to *f*	600	3	3
Runout to riser..........	*f* to *g*	600	3½	3½
Supply main..........	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main..........	*f* to *k*	600	1¼	2
Wet return main..........	*k* to *m*	600	1	2
Wet return main..........	*m* to *n*	600	1	2
Wet return main..........	*n* to *p*	600	1	2

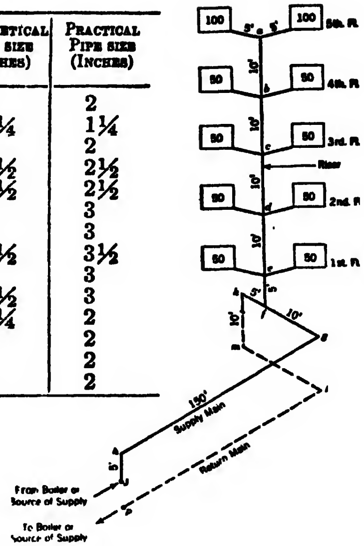


Fig. 19. Riser, Supply Main and Return Main of One-Pipe System

SIZING PIPING FOR TWO-PIPE HIGH PRESSURE SYSTEMS

Steam piping for two-pipe high pressure can be sized for greater pressure drops than that of the return piping. For a system using steam at 30 psig the total pressure drop can be 5 to 10 psi and for 150 psig systems, 25 to 30 psi.

It has been observed that the maximum total pressure in the returns of a 30-psig system is about 5 psi, and that of a 150-psig system is about 20 psi. The pressure in the return mains is, of course, caused by the discharge of traps, or by leaky traps and by flashing of condensate into steam due to the lower pressure in the return line. The usual practice in the sizing of high pressure returns has been to size on the basis of ½ psi per 100 ft of pipe for 30-psig systems, and 1 psi per 100 ft for 150-psig systems. This is an average figure which corresponds generally to several of the previously published tables for the design of high pressure return piping.

Notes on Two-Pipe High Pressure Systems

Pitch of mains should not be less than ¼ inch in 10 feet.

Pitch of horizontal runouts to risers and heating units should be not less than ½ inch per foot.

SIZING PIPING FOR TWO-PIPE LOW PRESSURE SYSTEMS

Piping for two-pipe low pressure systems is sized in the same manner as for two-pipe vapor systems, except that the pressure drop throughout the system can be based on ½ psi to 1 psi drop.

SIZING PIPING FOR TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensate return to the boiler by

gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire, and (3) that with large variations in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, Table 5, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Tables 6 and 7, Column *U*, (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* for small systems correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed $\frac{1}{8}$ psi to $\frac{1}{4}$ psi, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over $\frac{1}{8}$ psi divided by 4, or $\frac{1}{32}$ psi. In this case the steam mains would be sized from Column *B*, the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of $\frac{1}{32}$ psi. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over $\frac{1}{32}$ psi. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than $\frac{1}{4}$ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than $\frac{1}{2}$ in. per foot. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general it is not desirable to have a supply main smaller than 2 in.

4. When necessary, supply main, supply risers, or runouts to supply risers should be dripped separately into a wet return, or may be connected into the dry return through a thermostatic drip trap.

SIZING PIPING FOR TWO-PIPE VACUUM SYSTEMS

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from $\frac{1}{4}$ to $\frac{1}{2}$ psi. Systems in which the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop, while systems over 200 ft equivalent length of run, more frequently are designed for the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of $\frac{1}{2}$ psi divided by 12, or $\frac{1}{24}$ psi. In this case, the steam main would be sized from Column *C*, Table 5, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of $\frac{1}{24}$ psi). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*, Tables 6 and 7; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛ psi drop per 100 ft of equivalent run nor to exceed 1 psi total pressure drop in any system.

2. Pitch of mains should not be less than ¼ in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per foot. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in.

5. When necessary, the supply main, supply riser, or runout to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in this chapter.

7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

SIZING PIPING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized according to the quantity of steam condensed by each unit. The condensation per unit depends upon the entering temperature and the air velocity, and may be obtained from manufacturers' rating tables. Where more than one unit are placed in series, the entering air temperature for any unit will be the leaving temperature for the preceding unit.

When the amount of condensation has been obtained for each unit, the pipe sizes should be based on the length of run and the pressure drop desired, as in the case of radiators. It is generally desirable to place the indirect heating units on a separate piping system rather than to connect them to the piping which supplies direct radiation. For type of connections see section on Connections to Heating Units.

PRESSURE REDUCING VALVES

While the illustrations given in Figs. 2 to 17 inclusive indicate the various systems to be supplied by separate boiler plants, it is also possible to have steam supplied at high pressure by a boiler plant remotely located. In this case steam is supplied directly to the system or through a pressure reducing valve. Condensate can either be returned to the boiler plant or wasted to the sewer. The general arrangement of the systems fed through a pressure reducing valve will not vary from those illustrated with a boiler supply.

When high pressure steam is being supplied and lower steam pressures are required for heating, for domestic hot water, for utility services, etc., one or more pressure reducing valves (pressure regulators) are required.

These are used in two classes of service, one where the steam must be shut off tight to prevent the low pressure building up at time of no load, and the other where the low pressure lines will condense enough steam to offset normal leakage through the valve. In the latter case, double seated valves may be used in a manner that reduces the work required of the diaphragm in closing the valve and consequently the size of the diaphragm. These valves also control the low pressures more closely under conditions of varying high pressures.

Valves that shut off all steam are called *dead end* type. They are single seated, and some of them have pilot operation that provides close control of the reduced pressure. If a thermostatically controlled valve is installed

after, and near, a reducing valve in such a manner as to cut off the passage of steam, the dead end type should be used.

It is common practice when the initial steam pressure is 100 psig or higher to install two-stage reduction. If the radiation served is cast-iron, the *A.S.M.E.* code requires two reducing valves when the inlet pressure exceeds 50 psig. This makes a quieter condition of steam flow, as it is apparent that with one reduction, as for example from 150 to 2 psig, there is a smaller opening with greater velocity across the reducing valve and, consequently, more noise. A two-stage reduction also introduces a source of safety, since if one reducing valve were to build up its discharge pressure, this excess pressure would not be so great as the case might be in a one-stage reduction.

If an installation requires single seated valves and the pilot type cannot be used, it is necessary to use two-stage reduction, as single seated valves require sufficient diaphragm area to overcome the unbalanced pressure underneath the single valve. In many cases the large diameter of diaphragm required would make it impractical in construction. With a two-stage reduction the diaphragm diameter required would be reduced. If a one-stage reduction is desired, it is necessary to use a pilot controlled pressure reducing valve, where low pressures are to be maintained closely.

In making a two-stage reduction, allowance for expansion of steam on the low pressure side of the valve should be made by increasing the pipe size. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance up to 20 ft is recommended to reduce excessive hunting action of the first valve.

When the reduced pressure is approximately 15 psig or lower, the weight and lever diaphragm valve gives the best results with minimum maintenance. Above 15 psig, spring loaded diaphragm valves should be used, because of the extra weights required on weight and lever type. Pressure equalizing lines should not be connected too close to the valve. They should be connected into the bottom of the reduced pressure steam main, to allow maximum condensate to exist in the equalizing lines, or the connection can be made into the top of the main if a water accumulator is used to reduce the variation of the head of water on the diaphragm.

Care should be exercised in selecting the size of a reducing valve. The safest method is to consult the manufacturer. It is essential that sizes of piping to and from the reducing valve be such that they will pass the desired amount of steam with the maximum velocity desired. A common error is to make the size of the reducing valve the same size as that of the service, or outlet pipe size. Generally, this will make the reducing valve oversized, and bring about wire-drawing of valve and seat, due to small lift of the valve seat.

On installations where the steam requirements are relatively large and variable in mild weather or reduced demand periods, wire-drawing may occur. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 per cent proportion of maximum flow. For example, if 50,000 lb of steam per hour are required, the size of one valve is on the basis of $0.7 \times 50{,}000$ lb, or 35,000 lb, and the other on the basis of $0.3 \times 50{,}000$ lb, or 15,000 lb. During the mild or reduced demand periods, steam will flow through the smaller valve only. During the remainder of the season, the larger valve is set to control at whatever low pressure is desired, and the smaller one at a somewhat lower pressure. Thus, when steam flow is not at its maximum, the smaller valve is closed, but it opens automatically when the maximum

steam demand occurs, because this maximum demand creates a slight pressure drop in the service line.

The installation of reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair, and wherever possible with diaphragm downward, except in cases of pilot operated valves.

There should be a by-pass around each reducing valve of size equal to one-half the size of reducing valve. The globe valve in by-pass line should be of a better type of construction, and must shut off absolutely tight. A steam pressure gage, graduated up to the initial pressure, should be installed on the low pressure side. Safety valves located on the low pressure side should be set 5 psi higher than the final pressure but may be 10 psi higher than the reduced pressure if this reduced pressure is that of the first stage reduction of a double reduction. Strainers are sometimes installed on the inlet to the reducing valve but are not required before a

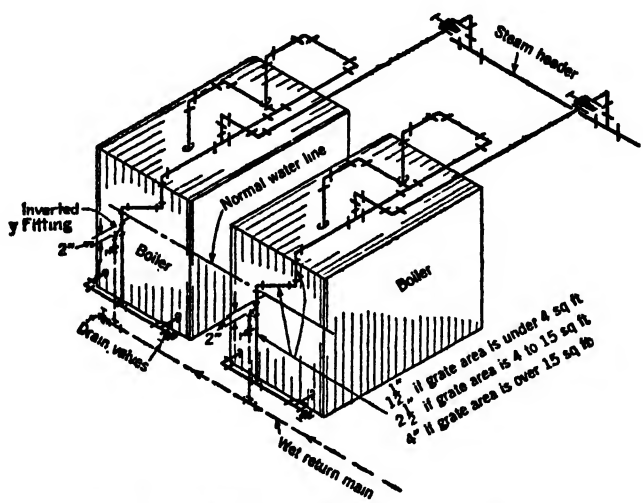


Fig. 20. The Hartford Return Connection

second-stage reduction. If a two-stage reduction is made, it is well to install a pressure gage immediately before the reducing valve of the second-stage reduction also. In sizes 3 in. and above, it is advisable to install a drip trap between the two reducing valves.

BOILER CONNECTIONS

Steam

Cast-iron, sectional heating boilers usually have several outlets in the top. Two or more outlets should be used whenever possible to reduce the velocity of the steam in the vertical uptakes from the boiler and thus to prevent water being carried over into the steam main.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return

tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. The return connection should include either a Hartford return connection or a check valve to prevent the accidental loss of boiler water to the returns with consequent danger of boiler damage. The Hartford return connection is to be preferred over the check valve because the latter is apt to stick or not close tightly and, furthermore, because the check valve offers additional resistance to the condensate coming back to the boiler, which in gravity systems would raise the water line in the far end of the wet return several inches.

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford return connection is recommended. This connection for a one- or two-boiler installation is shown in Fig. 20. The essential features of construction of a Hartford return connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple, or preferably an inverted Y-fitting connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return pressure balance connection. Equalizing pipe connections between the steam and return are given in Fig. 20, based on grate areas, but in no case shall this pipe size be less than the main return piping from the system.

Sizing Boiler Connections

Little information is available on the sizing of boiler runouts and steam headers. Although some engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runouts on the same basis as the building mains. The difference in size between the vertical uptakes from the boiler, which should be of same size as the boiler outlet tapping, and the horizontal main or runout is compensated for by the use of reducing ells.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop. The relative boiler loads should be considered, as in the case of gravity

return connections. Boiler header and piping sizes should be based on the total load.

CONDENSATE RETURN PUMPS

Condensate return pumps are used for gravity systems when the local conditions do not permit the condensate to return to the boiler under the existing static head. The return of the condensate permits the water to repeatedly go through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems in which the condensate is wasted and fresh make-up water is continually being introduced.

A generally accepted condensate pump unit for low pressure heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw, turbine and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute and the pump generally has a delivery rate of 3 to 4 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

A typical installation of a motor driven automatic condensate unit is illustrated in Fig. 9.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric systems, where the supply piping, radiation and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and non-condensable gases to atmosphere and to dispose of the condensate. Direct-acting steam-driven reciprocating vacuum pumps are sometimes used where high pressure steam is available or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return line heating pump especially developed for this service. Steam turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere and a water removal unit which discharges the condensate to the boiler. They are furnished complete with receiver, separating tank and automatic controls mounted as an integrated unit on one base. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 psi drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

For rating purposes[3] vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated for maintaining $5\frac{1}{2}$ in. Hg vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above $5\frac{1}{2}$ in.

Manufacturers of vacuum pumps specify that the standard capacity of pumps shall be 0.3 to 0.5 cfm of air removal and 0.5 gpm of water per 1000 *EDR* served. This capacity is at $5\frac{1}{2}$ in. of vacuum and with condensate at 160 F. The larger air capacity is for smaller systems and the smaller capacity for the larger systems.

Some manufacturers, however, specify more air capacity than standard where higher vacuums are desired and where air leakage is suspected.

The vacuum that can be maintained on a system depends upon the relationship of the air leakage rate into the system to the operating air capacity of the hydraulic evacuator when operating at any given return line temperature. The hotter the returns, the lower will be the possible vacuum for a given air leakage rate into the system. It is particularly essential on high vacuum installations to see that the entire system is tight in order to reduce the amount of inward air leakage and, furthermore, to see that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high pressure drips, etc. It is for this reason that the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line, but should drain to a flash tank or flash leg through a high pressure trap. The receiver should have an equalizing connection to a low pressure steam main and drain through a low pressure trap to the vacuum return main as indicated later in this chapter in section on Drips.

Vacuum Pump Controls

In the ordinary vacuum system, the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and cuts out when it has been increased to the highest point, these points being varied to suit the particular system or operating conditions. In addition to this vacuum control, a float control is included which will start the pump whenever sufficient condensate accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensate pump only, also to give manual or continuous operation when desired.

There are several variations in the control of the vacuum maintained on the system by the pump. In some sub-atmospheric systems where orifices are used, the vacuum pump control maintains a pressure difference between the supply and the return piping, which is held within relatively close limits. There are other sub-atmospheric systems which utilize special temperature-pressure actuated controls for maintaining the desired conditions in the return lines. Where various zones are connected to the same return main, the return vacuum must be controlled to meet the requirements of the zone operating at the lowest steam supply pressure.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either electric or steam driven. Their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump and at an elevation sufficiently

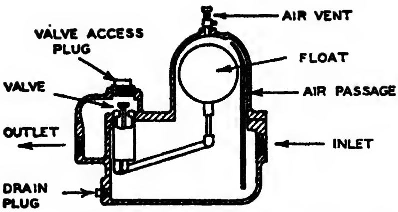


Fig. 21. Single Port Float Trap

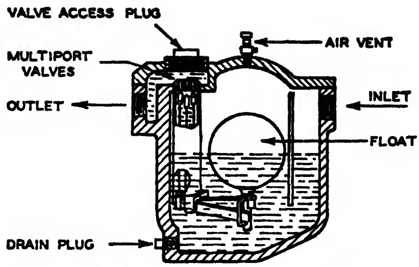


Fig. 22. Multiport Float Trap

high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed, then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensate is used for average vacuums and systems.

STEAM TRAPS

Steam traps, as the name implies, are automatic devices used to trap or hold steam in an apparatus or piping system until it has given up its latent heat, and to allow condensate and air to pass as soon as it accumulates. In general traps consist of a vessel in which to accumulate the condensate, an orifice through which the condensate is discharged, a valve to close the orifice port, mechanisms to operate the valve, and inlet and outlet openings for the entrance and discharge of the condensate from the trap vessel.

Steam traps are classified according to the type of operating device by which they function. The traps which are available on the market today may be classified as (1) float, (2) thermostatic, (3) float and thermostatic, (4) upright bucket, (5) inverted bucket, (6) flash, (7) impulse, (8) tilting, (9) lifting, and (10) boiler return trap or alternating receiver.

Float Traps. Float traps operate by the rise and fall of a float due to a change of condensate level in the trap. When the trap is empty, the float is in its lowest position and the discharge valve is closed. As condensate accumulates in the trap

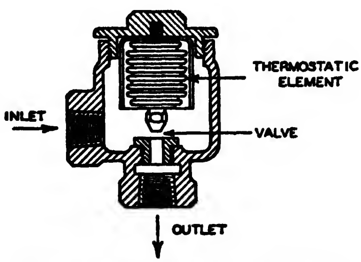


Fig. 23. Thermostatic Trap Bellows Type

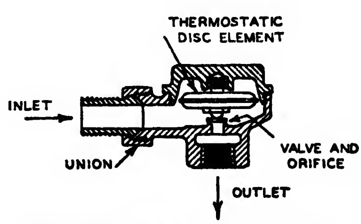


Fig. 24. Thermostatic Trap Disc Type

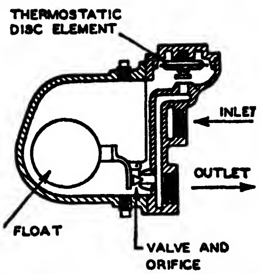


FIG. 25. TYPICAL FLOAT AND THERMOSTATIC TRAP

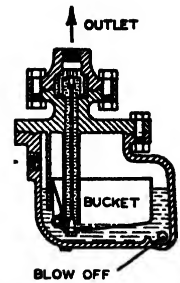


FIG. 26. UPRIGHT BUCKET TRAP

chamber, the float rises and gradually opens the valve and the pressure of the steam pushes the condensate out of the valve. The discharge from a float trap is generally continuous, since the opening of the valve is proportioned to the flow of condensate through the trap. A gage glass may be used to indicate the height of condensate in the trap chamber. Unless float traps are well made and proportioned, there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. Float traps are made in sizes from ¾ to 3 in., and for pressures varying from vacuum conditions to 200 psig. Float traps are used for draining condensate from steam separators, steam headers, blast coils, heating systems, steam water heaters, laundry equipment, sterilizers, and other equipment. When used for draining low pressure systems, float traps should be equipped with a thermostatic air vent (See Float and Thermostatic Traps). Figs. 21 and 22 illustrate types of float traps which are in use at the present time.

Thermostatic Traps. Thermostatic traps function by means of elements which expand and contract under the influence of heat and cold. Early types of thermostatic traps employed carbon posts and bi-metallic elements for expansion. In general, the modern type of thermostatic trap consists of thin corrugated metal bellows or discs enclosing a hollow chamber. The chamber is either filled with a liquid or a small amount of a volatile liquid such as alcohol is introduced. The liquid expands or becomes a gas when steam comes in contact with the expansive element. The pressure created in either case expands the element and closes the valve of the trap against the escape of steam. When condensate or air comes in contact with the element, it cools and contracts, opening the valve and allowing the escape of water and air.

The discharge from this type of trap is intermittent. Thermostatic traps find their use generally for the draining of condensate from radiators, convectors, pipe coils, drips, unit heaters, water heaters, cooking kettles, and other equipment. Except for radiators and convectors, it is recommended that a strainer be installed on the inlet connection to the trap to prevent dirt, pipe scale, and other foreign substance from entering the trap. A cooling leg of a length of pipe should also be provided ahead of the trap on unit heaters and similar apparatus to cool the condensate

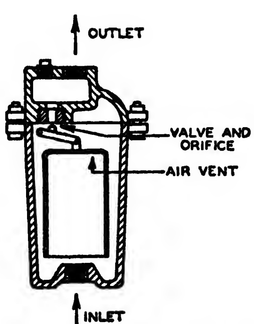


FIG. 27. INVERTED BUCKET TRAP

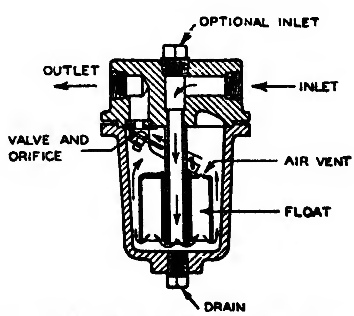


FIG. 28. INVERTED BUCKET TRAP WITH CENTRAL GUIDE

in order to help in the trap action. Thermostatic traps are made in sizes from ½ to 2 in. and for pressures ranging from vacuum conditions to 300 psig. Figs. 23 and 24 illustrate types of thermostatic traps which are available at the present time.

Float and Thermostatic Traps. This type of trap is a combination of the float trap and the thermostatic trap and finds its use in the draining of condensate from unit heaters, blast heaters, and coil heaters for water, oil or other liquids where there is apt to be a large volume of condensate which would not permit successful operation of thermostatic traps alone. The function of the float element of this trap is to handle the condensate and of the thermostatic element to permit the flow of air and to prevent the flow of steam around the float valve. Float and thermostatic traps are made in sizes from ½ to 2 in. and operate under pressures varying from vacuum conditions to 40 psig. Fig. 25 illustrates a typical float and thermostatic trap.

Upright Bucket Traps. In this type of trap, the condensate enters the trap chamber and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the condensate in the chamber rises above the edges of the bucket, it overflows into it and causes the bucket to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of condensate in the bucket to force the water to the discharge opening. When the bucket is emptied, it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent, and it requires a definite differential pressure (usually 1 psi at least) between the inlet and outlet of the trap in order to

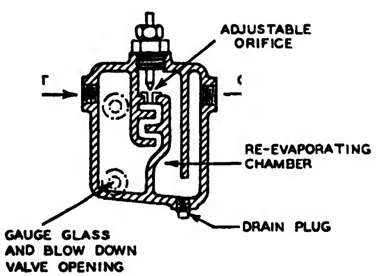


FIG. 29. FLASH TRAP

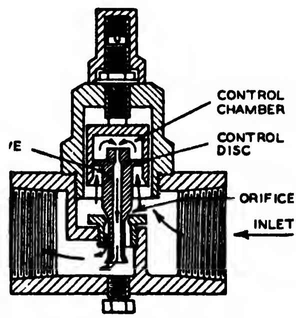


FIG. 30. IMPULSE TRAP

lift the condensate out of the bucket to the return opening. Upright bucket traps are used for draining condensate and air from blast coils, unit heaters, steam mains, laundry equipment, sterilizers, water and oil heaters and other equipment. This type of trap is particularly suited for use where there are pulsating pressures, such as draining steam lines and separators to reciprocating pumps or engines. It is not influenced by pulsations or wide fluctuations of pressure. Upright bucket traps are obtainable in sizes varying from ½ to 2½ in. and for pressures varying from vacuum to 1200 psig. Fig. 26 illustrates an upright bucket trap.

Inverted Bucket Traps. In this type of trap, steam, condensate and air enter the trap under the bell or inverted bucket. Steam floats the inverted bucket and closes the valve. Condensate entering the trap enables the inverted bucket to fall, opening the valve. The steam pressure entering through the open valve discharges the trap. Air is eliminated automatically by passing through the small vent hole located in the top of the inverted bucket. Inverted bucket traps for use on low pressure system particularly with blast coils or unit heaters are usually furnished with a large capacity opening equipped with a bi-metallic thermostatic element which closes when heated by steam and opens when cooled by air and condensate, allowing air to escape from the inverted bucket to the trap outlet. Inverted bucket traps are used for draining condensate and air from blast coils, unit heaters, steam drips, laundry equipment, sterilizers, steam water heaters and other equipment. They are particularly suited for draining condensate from steam lines or equipment where abnormal amounts of air must be discharged, and where there is also foreign matter such as dirt, sludge and oil draining to the trap. The discharge from inverted bucket traps like that of the upright bucket traps is intermittent and requires a definite differential pressure between the inlet and the outlet of the trap in order to lift the condensate from the

bottom of the trap to the outlet of same. Inverted bucket traps are made in sizes from ½ to 3 inches and for pressures varying from vacuum to 2400 psig. Figs. 27 and 28 illustrate some of the types of inverted bucket traps which are available on the market at the present time.

Flash Traps. These traps depend on the property of condensate at a high pressure and temperature to flash into steam at a lower pressure. Condensate flows freely through the orifice of the trap due to the pressure difference from inlet to outlet of trap until steam enters the inlet chamber and mixes with the remaining condensate, heating the condensate and causing it to flash thereby choking the flow through the orifice and allowing more condensate to accumulate in the trap. The discharge from flash type traps is intermittent. There are no moving parts in this type of trap. The orifice, however, is adjustable for the pressure differential required. A gage glass or float indicates whether the trap is operating. These traps can be used for draining condensate from steam water and oil heaters, blast heaters, unit heaters, dryers, vulcanizers, kitchen equipment, laundry equipment, evaporators, steam lines and other equipment, where the pressure differential between steam supply and condensate return does not drop below 5 psi. Flash type traps are made in sizes from ½ to 3 in. and for pressures varying from vacuum to 450 psig. Fig. 29 illustrates a trap of the flash type.

Impulse Traps. These traps are a modification of the flash trap, and depend on the same principle of flash for their operation. In the impulse trap the flashing action

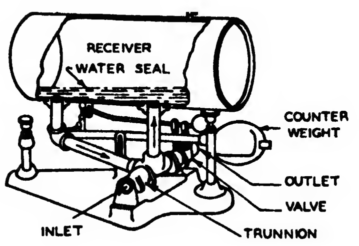


FIG. 31. TILTING TRAP

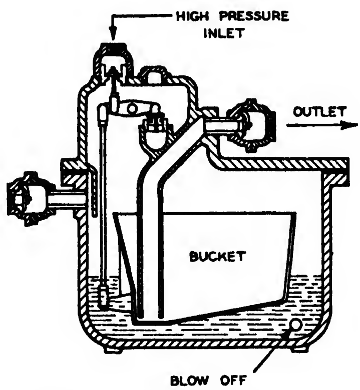


FIG. 32. LIFTING TRAP

is utilized to govern the movement of a valve by causing changes in pressure in a control chamber above the valve. When condensate, which is at a low temperature, builds up in this chamber, the flow through the center orifice does not change volume and the discharge through the orifice reduces the volume in the control chamber and the valve opens to discharge air and condensate.

When steam comes in contact with the trap, the condensate is heated and the flow in entering the control chamber flashes and increases the volume of the control flow. The discharge through the center orifice is thereby choked, pressure in the control chamber builds up, closing the valve and stopping all discharge of hot condensate except a small amount that flows through the center orifice.

The discharge from an impulse trap is pulsating or intermittent, but not as infrequent as with the bucket type of traps. It is entirely non-adjustable. Impulse traps can be used for draining condensate from steam mains, unit heaters, laundry equipment, kitchen equipment, oil and water heaters, sterilizers, and other equipment where the pressure at the trap outlet is 25 per cent or less than that of the inlet pressure. Impulse traps are made in sizes from ½ to 2 in. and for pressures ranging from one to 600 psig. Fig. 30 illustrates a trap of the impulse type.

Tilting Traps. This type of trap as the name implies depends for its operation on the tilting of the trap receiver. When the receiver is in a horizontal position condensate accumulates until the weight of condensate overbalances that of a counterweight, when the receiver tilts. The tilting action opens the discharge valve and steam pressure pushes the condensate out of the open discharge valve. When the receiver tank is emptied, except for a slight water seal, the receiver drops back to its

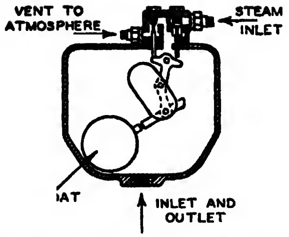


FIG. 33. BOILER RETURN TRAP OR ALTERNATING RECEIVER

FIG. 34. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

horizontal position and closes the discharge valve and is again in position to accumulate condensate.

Tilting traps are necessarily intermittent in operation, except that once the trap is in the discharge position, it will discharge condensate continuously as long as the flow of condensate is sufficient to overcome the balance of the counterweight.

This type of trap employs packing around the trunnion and valve stem in order to prevent the loss of steam and condensate. Tilting traps are used for draining laundry and dry cleaning equipment, steam cookers, drips from steam mains, steam separators and purifiers and other equipment. They are made in sizes from 1 to 3 in. and for pressures varying from 0 to 250 psig. Fig. 31 illustrates a type of tilting trap which is in use at the present time.

Lifting Traps. This type of trap is an adaptation of the upright bucket trap. It has the added feature of an auxiliary pressure inlet through which steam is introduced at a pressure higher than that of the trap inlet pressure. This high pressure steam forces the condensate to a point above the trap, and against a back pressure higher than that which is possible with normal steam pressure. Lifting traps are made in sizes from one to 3 in. and for pressures ranging from vacuum to 150 psig. Fig. 32 illustrates a trap of the lifting type.

Boiler Return Trap or Alternating Receiver. This device is not actually a steam trap in that it is not used to trap or hold steam, but is an adaptation of the lifting trap. It is used for returning condensate to a low pressure boiler when due to excess pressure the condensate cannot flow to the boiler by gravity, without flooding the return mains, and endangering the boiler by permitting it to go dry. The boiler return trap is a vessel into which condensate alternately collects and is discharged into the boiler by boiler steam pressure. These traps are available in sizes from 1½ to 2½ in. and for pressures varying from 0 to 100 psig. A typical boiler return trap is shown in Fig. 33 and a typical connection to a low pressure heating system is indicated in Fig. 13.

Steam Trap Installations

The following general rules should govern the installation of traps of all types:

1. A vertical drip as long as possible and a strainer should be installed between the trap and the apparatus it drains. Exceptions to this rule are the installation of ther-

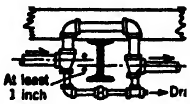


FIG. 35. LOOPING MAIN AROUND BEAM

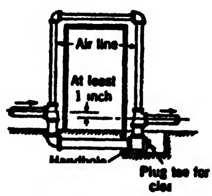


FIG. 36. LOOPING DRY RETURN MAIN AROUND OPENING

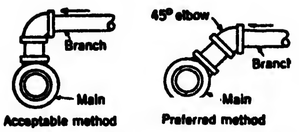


Fig. 37. Methods of Taking Branch from Main

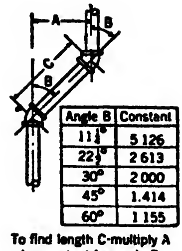


Fig. 38. Constants for Determining Length of Offset Pipe

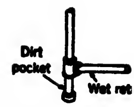


Fig. 39. Dirt Pocket Connection

mostatic traps in radiators, convectors and pipe coils. These, in general, are attached directly to the units without strainers.

2. Whenever it is necessary to maintain in continuous service, apparatus which is to be drained, it is advisable to install a gate valve on each side of the trap and a valved bypass around the trap, so that the trap may be removed and repaired and condensate drained through the throttled bypass valve.

3. Whenever it is necessary to install traps for lift service, as when the condensate must be discharged to a main located above the trap or where the trap must discharge against a definite back pressure, a check valve and a gate valve should be installed on the discharge side of the trap, the check valve to prevent continuous pressure on the discharge side of the valve and the gate valve to shut off pressure in case the trap is removed for service or repair.

DRIPS

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the direction of steam flow. Any steam main in any heating system can be elevated if dripped. Fig. 34 shows a connection where the steam main is raised and is drained to a wet return. If the elevation of the low point is above a dry return, it may be drained through a trap to the dry return in two-pipe vapor, vacuum and sub-atmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensate (Fig. 35). Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 36. It will be noted that the large pipe, in this case, runs below the obstruction and the smaller one over it.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 37, but where radiator condensate does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of

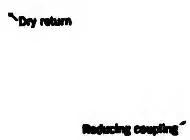


Fig. 40. Dripping End of Main into Wet Return

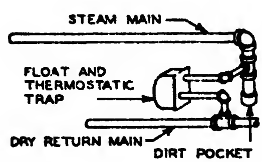


Fig. 41. Dripping End of Steam Main into Dry Return

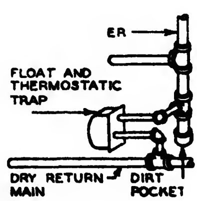


Fig. 42. Dripping Heel of Riser into Dry Return

giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 38.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which

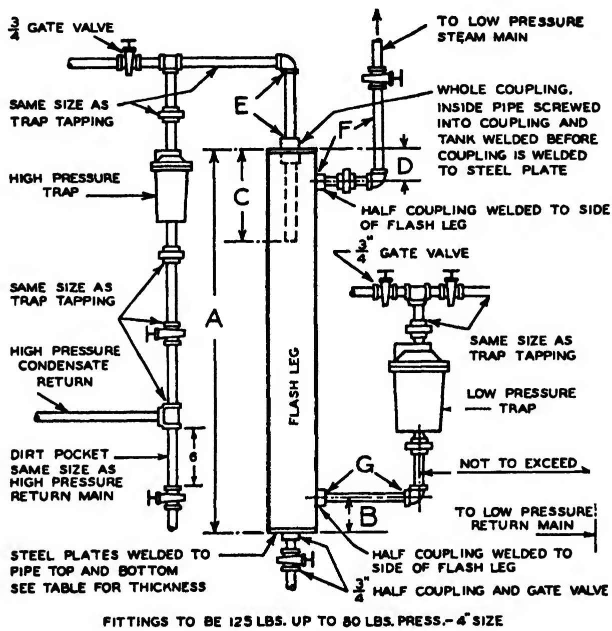


Fig. 43. Flash Leg Installation for 80 psi Maximum Steam Working Pressure

Table 13. Dimensions Applying to Fig. 43

Condensate per Hr., Lb at 70 Psi	Traps: HP Side: Pipe Size	Traps: HP Side: Orifice	Traps: HP Side: Pres. Range, psi	Traps: LP Side: Pipe Size	Traps: LP Side: Orifice	Traps: LP Side: Pres. Range, psi	Flash Leg, ips	A Ft.	B In.	C In.	D In.	E In.	F In.	G In.	Plate Thickness, In.
200	3/4	1/8	21–80	3/4	3/16	0–20	4	3	6	8	3	3/4	1 1/4	3/4	3/8
300	3/4	5/32	31–70	3/4	5/16	0–15	4	3	6	8	3	3/4	1 1/4	3/4	3/8
700	3/4	3/16	31–70	3/4	3/8	0–15	4	3	6	8	3	3/4	1 1/4	3/4	3/8
1500	3/4	9/32	61–80	1	1/2	0–15	5	4	6	10	4	3/4	1 1/2	1	3/8
2500	1	9/32	61–80	1 1/4	3/4	0–15	8	5	6	10	4	1	2	1 1/4	1/2
4000	1 1/4	11/32	61–80	2	1 1/16	0–15	10	6	6	12	5	1 1/4	2 1/2	1 1/2	5/8
8000	2	9/16	61–80	2 1/2	1 1/16	0–15	12	6	6	12	5	2	3	2	3/4
15000	2 1/2	9/16	61–80	2 1/2	1 1/16	0–15	16	6	8	14	6	3	3	3	7/8

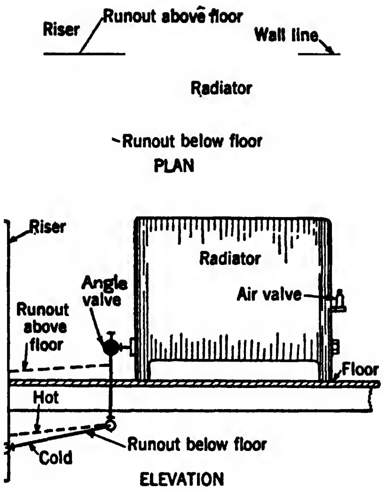


FIG. 44. ONE-PIPE RADIATOR CONNECTIONS

will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 39.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 40. On low pressure or vacuum systems, the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. A float and thermostatic type trap is recommended for dripping steam mains and risers as indicated in Figs. 41 and 42.

The dripping of high pressure mains or of equipment using high pressure steam into low pressure or vacuum returns is generally accomplished by the use of a flash tank or flash leg into which the high pressure trap is arranged to discharge. This tank provides the required space for the flashing from high temperature condensate to low pressure steam to take place. The low pressure steam therein generated is passed directly to the low pressure steam mains and the condensate is discharged through a second trap to the

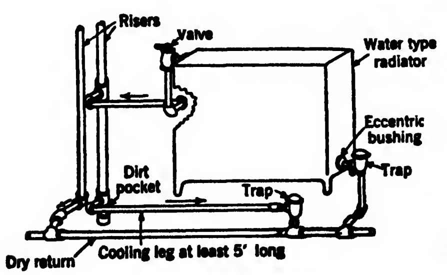


FIG. 45. TWO-PIPE TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

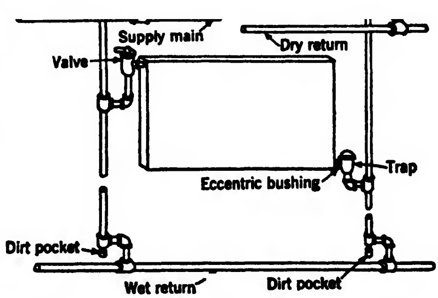


FIG. 46. TWO-PIPE CONNECTIONS TO RADIATOR HUNG ON WALL

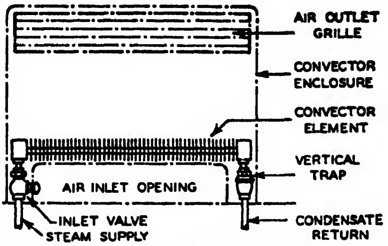


Fig. 47. Typical Convector Connections

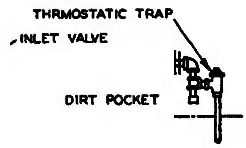


Fig. 48. Typical Connections to Finned Pipe Convector

low pressure or vacuum return. A typical arrangement of a flash leg with sizes required for varying capacities is given in Fig. 43.

CONNECTIONS TO HEATING UNITS

Riser, radiator and convector connections must not only be properly pitched at the time they are installed but must be arranged so that the pitch will be maintained under the strains of expansion and contraction. These connections may be made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. To take care of expansion in long risers, either expansion joints of commercial construction or pipe swing joints are used. Anchoring of pipes between expansion joints is desirable.

Two satisfactory methods of making runouts for one-pipe systems for either the up-feed or the down-feed type are shown in Fig. 44. Where the vertical distance is limited and the runouts must run above the floor, the radiator may be set on pedestals or raised by means of high legs. Two methods of connecting a unit heater to a one-pipe steam heating system are illustrated in Fig. 2 (and also in Fig. 5 of Chapter 26).

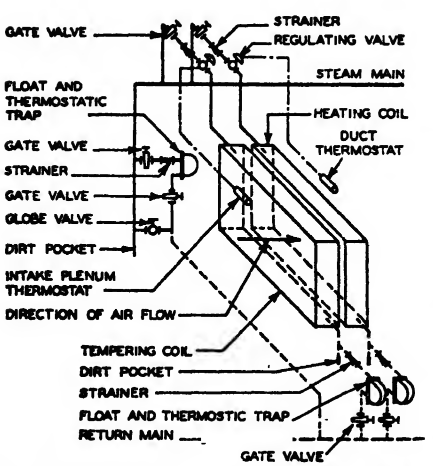


Fig. 49. Typical Connections to Finned Tube Blast Heating Coils Arranged for Series Flow of Air

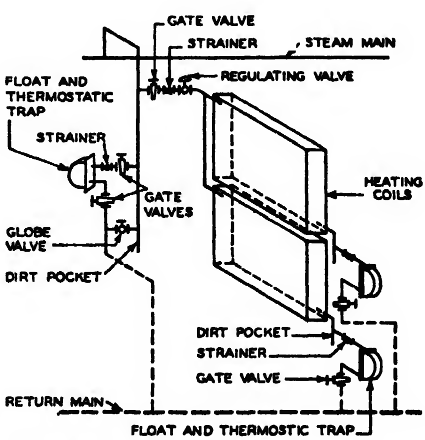


Fig. 50. Typical Connections to Finned Tube Blast Heating Coils Arranged for Parallel Flow of Air

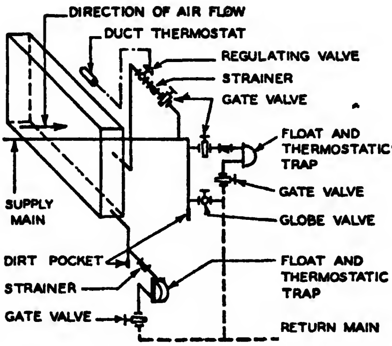


Fig. 51. Typical Connections to Finned Tube Blast Heating Coils of the Steam Distributing or Non-Freeze Type

Typical two-pipe radiator connections are shown in Fig. 45 and 46. While these show top inlet supply connections which are preferred, it is also possible to connect the supply to the bottom of the radiator. Short radiators may be connected with top supply and bottom return on the same end.

A typical method of connecting convectors is shown in Fig. 47. Sometimes the supply valve is omitted on convector connections and a damper is supplied in the outlet grille for heat control.

A typical connection for finned pipe convectors is shown in Fig. 48.

Typical connections to blast heaters are shown in Fig. 49, 50, and 51. Figure 52 shows a typical return and connection for blast heaters connected to high pressure systems.

A typical two-pipe connection to a unit heater is indicated in Figure 53.

CONTROL VALVES

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never

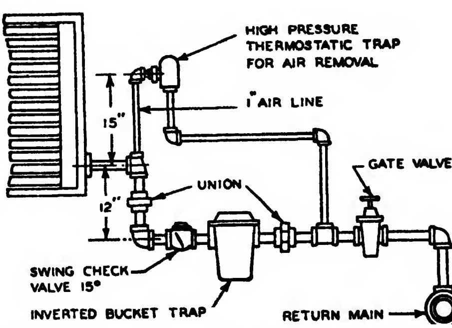


Fig. 52. Typical Return Connections to Finned Tube Blast Heaters with High Pressure Steam

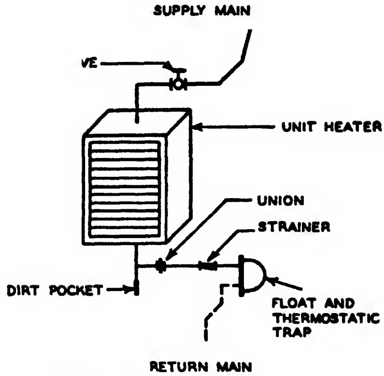


FIG. 53. TYPICAL UNIT HEATER CONNECTIONS FOR TWO-PIPE SYSTEM

be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

REFERENCES

[1] A.S.H.V.E. RESEARCH REPORT No. 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199).

[2] Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[3] A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33).

CHAPTER 24

HOT WATER HEATING SYSTEMS AND PIPING

Available Head; Friction Loss; Classification; System Design; Examples of Piping Design; One-Pipe Gravity, One-Pipe Forced Circulation, Two-Pipe Gravity, and Two-Pipe Forced Circulation Systems, Expansion Tanks, Installation Details, Zoning of Systems

A HEATING system is called a hot water system if water is used to convey heat by flowing through pipes connecting a boiler or water heater with radiators, convectors or other suitable heat dispensing means. There are two types: the gravity system in which the water flows by virtue of thermo-syphon action, and the forced system in which a pump, usually driven by an electric motor, sometimes by a steam turbine or other means, maintains the necessary flow. Most panel heating systems (see Chaper 31) fall into the category of forced hot water systems, and the design procedures pertaining to pipe sizing and friction contained in this chapter are largely applicable to such systems.

Historically, the gravity system is much the older, and many such systems have been in satisfactory operation for several decades. Operation depends on the difference in density of the water due to difference in temperature in the flow and return pipes. The available head is therefore limited and the pipes must be ample in size to permit adequate flow of water. In the forced system, the pipes, valves and fittings can be much smaller, with a resultant saving in the cost of installation, since the available head is limited only by consideration of economy in pumping the water. With the forced system, higher boiler temperatures and automatic control of the pump or circulator make possible the use of indirect water heaters with hot water systems when that is desirable. (See Chapter 50).

AVAILABLE CIRCULATION HEAD

The available head in a gravity circulation system may be found from the equation:

$$h_a = \frac{\rho_2 - \rho_1}{144} \times 2.31 \times 12{,}000 \qquad (1)$$

where

h_a = available head per foot of height, milinches (1 milinch = 1/1000 of 1 in. of water).

ρ_1 = average density of flow water, pounds per cubic foot.

ρ_2 = average density of return, pounds per cubic foot.

144 = square inches per square foot.

2.31 = height of water column equivalent to 1 psi, feet.

12,000 = milinches equivalent of 1 ft of water column.

The available head may also be found from Fig. 1. For example, at a flow temperature of 200 F and a 35 deg drop, and with the mains located 4 ft above the top of the boiler, a head of 600 milinches is obtained. This is found by following the 200 F flow riser line in Fig. 1 to its intersection with the 165 F return riser line and then reading, horizontally, a head of 150

milinches per foot or 600 milinches for 4 ft. If the first floor radiators are located 3 ft above the mains, second floor radiators 12 ft above the mains, third floor 21 ft, and fourth floor 30 ft, the heads are 450, 1800, 3150, and 4500 milinches respectively.

In *forced circulation* systems flow is produced mechanically by means of a pump driven by electricity, steam, or other source of energy. As

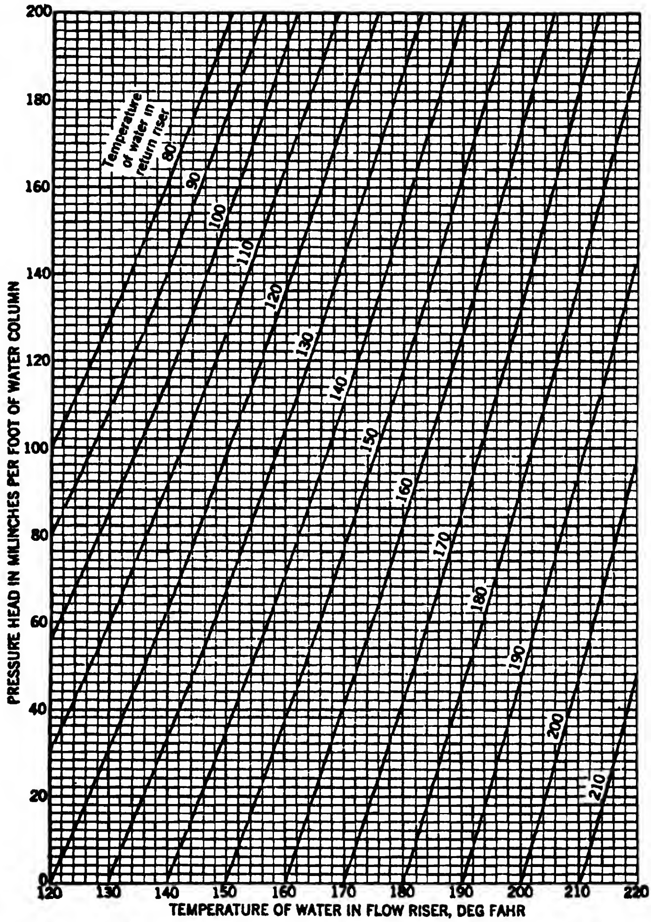


Fig. 1. Heads Resulting from Temperature Difference (Gravity Systems)

forced circulation velocities are higher than those in gravity systems, and as the friction in a heating system varies almost as the square of the velocity, a given error in the calculation or assumption of the velocity is less important in a forced circulation system than in a gravity circulation system, and, consequently, it is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FRICTION LOSS

Values of friction loss due to flow of water in the various parts of a heating system must be known in order to design either gravity or forced circulation systems. The friction loss of fittings is customarily expressed in equiv-

TABLE 1. IRON AND COPPER ELBOW EQUIVALENTS[a]

Fitting	Iron Pipe	Copper Tubing	Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0	Angle radiator valve	2.0	3.0
Elbow, 45-deg	0.7	0.7	Radiator or convector	3.0	4.0
Elbow, 90-deg long turn	0.5	0.5	Boiler or heater	3.0	4.0
Elbow, welded, 90-deg	0.5	0.5			
Reduced coupling	0.4	0.4	Tee, per cent flowing through branch:		
Open return band	1.0	1.0	100	1.8	1.2
Open gate valve	0.5	0.7	50	4.0	4.0
Open globe valve	12.0	17.0	25	16.0	20.0

[a] The friction in one 90 deg standard elbow is approximately equal to the friction of a length of straight pipe of the same nominal size and 25 diam long. Hence one elbow equivalent in feet of pipe equals 25 diam (in inches) divided by 12.

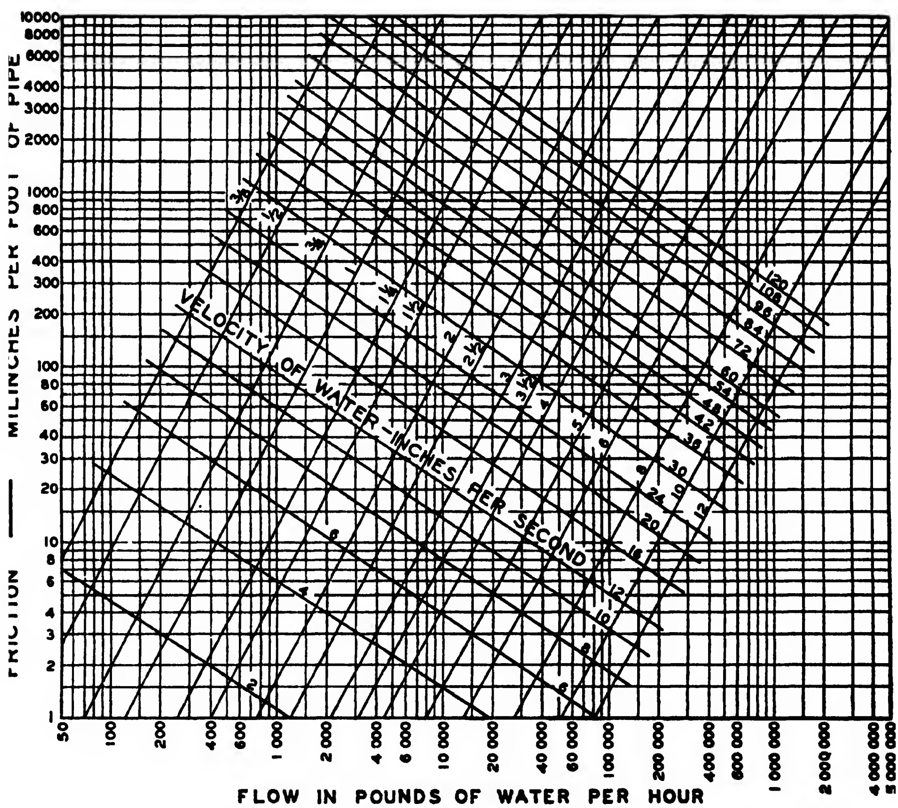


FIG. 2. FRICTION IN BLACK IRON PIPES BASED ON FLOW IN POUNDS PER HOUR

alent number of elbows of the same pipe size which would have the same friction loss. An elbow is assumed to have the same friction loss as a straight pipe having a length equal to 25 diameters (nominal) of the elbow.

The resistance of various types of fittings expressed in equivalent elbow resistance is shown in Table 1.

Table 2. Heat-carrying Capacity of Standard Black Pipes with Temperature Drop of 20 Deg[a]

Nominal Pipe Sizes ⅜ in. to 12 in., and Friction 4 to 800 milinches per foot (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Milinch Friction Loss per Foot of Pipe		Nominal Pipe Size, Inches															
		⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
4	A	0.75	1.35	2.85	5.4	11.3	17.0	33.0	53.1	95	141	197	363	596	1250	2320	3730
	B	1.5	1.7	2.1	2.4	2.9	3.2	3.8	4.3	5.0	5.5	6.0	7.0	7.9	9.6	11	12
6	A	0.9	1.7	3.6	6.75	14.0	21.2	41.3	66.4	119	176	248	456	748	1570	2920	4690
	B	1.8	2.1	2.6	3.0	3.6	4.0	4.7	5.3	6.2	6.9	7.5	8.8	10	12	14	16
8	A	1.05	2.0	4.2	7.9	16.4	24.8	48.4	77.9	140	207	291	535	879	1850	3440	5520
	B	2.1	2.5	3.0	3.5	4.2	4.7	5.6	6.3	7.3	8.0	8.8	10	12	14	17	19
10	A	1.2	2.2	4.7	8.9	18.6	28.0	54.7	88.1	153	234	329	605	997	2100	3910	6270
	B	2.4	2.8	3.4	4.0	4.8	5.3	6.3	7.1	8.2	9.1	9.9	12	13	16	19	27
12	A	1.35	2.45	5.2	9.8	20.5	31.0	60.4	97.4	175	259	364	671	1100	2320	4330	6950
	B	2.7	3.1	3.7	4.4	5.3	5.9	6.9	7.8	9.1	10	11	13	15	18	21	24
14	A	1.45	2.65	5.65	10.7	22.3	33.7	65.8	106	190	282	397	731	1200	2530	4730	7590
	B	2.9	3.4	4.1	4.8	5.7	6.4	7.6	8.5	9.9	11	12	14	16	20	23	26
16	A	1.55	2.85	6.05	11.5	24.0	36.3	70.8	114	205	303	428	787	1300	2730	5100	8190
	B	3.1	3.6	4.4	5.1	6.2	6.9	8.1	9.7	11	12	13	15	17	21	25	28
20	A	1.75	3.25	6.85	13.0	27.1	41.0	80.0	129	232	344	484	892	1470	3100	5790	9300
	B	3.5	4.1	4.9	5.8	7.0	7.7	9.2	10	12	13	15	17	20	24	28	32
25	A	2.0	3.65	7.75	14.7	30.6	46.3	90.5	146	263	389	548	1010	1670	3510	6570	10560
	B	4.0	4.6	5.6	6.5	7.9	8.8	10	12	14	15	17	19	22	27	32	36
30	A	2.2	4.0	8.55	16.2	33.8	51.2	100	162	290	430	607	1120	1850	3900	7280	11710
	B	4.4	5.1	6.1	7.2	8.7	9.7	11	13	15	17	18	22	25	30	35	40
35	A	2.35	4.4	9.3	17.6	36.8	55.7	109	176	316	469	661	1220	2010	4250	7940	12780
	B	4.7	5.5	6.7	7.9	9.5	11	13	14	16	18	20	23	27	33	39	44
40	A	2.55	4.7	10.0	18.9	39.6	59.9	117	189	341	505	712	1320	2170	4580	8570	13780
	B	5.1	5.9	7.2	8.4	10	11	13	15	18	20	22	25	29	35	42	47
50	A	2.85	5.3	11.3	21.4	44.7	67.7	133	214	386	572	807	1490	2460	5190	9720	15650
	B	5.7	6.7	8.1	9.5	12	13	15	17	20	22	24	29	33	40	47	54
60	A	3.15	5.85	12.4	23.6	49.4	74.9	147	238	427	633	893	1650	2730	5760	10780	17300
	B	6.3	7.4	8.9	11	13	14	17	19	22	25	27	32	36	44	52	60
70	A	3.45	6.35	13.5	25.7	53.8	81.4	160	258	465	690	973	1800	2970	6280	11760	18950
	B	6.9	8.0	9.7	11	14	15	18	21	24	27	29	35	40	48	57	65
80	A	3.7	6.8	14.5	27.6	57.9	87.6	172	278	500	743	1050	1940	3200	6770	12690	20440
	B	7.4	8.6	10	12	15	17	20	22	26	29	32	37	43	52	62	70
100	A	4.15	7.7	16.4	31.1	65.4	99.0	194	314	566	840	1190	2200	3630	7680	14400	23200
	B	8.3	9.7	12	14	17	19	22	25	30	33	36	42	48	59	70	80
150	A	5.2	9.6	20.4	38.8	81.6	124	243	393	709	1050	1490	2760	4560	9650	18120	29220
	B	10	12	15	17	21	23	28	32	37	41	45	53	61	74	88	101
200	A	6.05	11.2	23.9	45.4	95.5	145	285	461	832	1240	1750	3240	5360	11350	21320	34400
	B	12	14	17	20	25	27	33	37	43	48	53	62	71	87	104	118
300	A	7.5	13.9	29.7	56.6	119	181	356	577	1040	1550	2190	4060	6730	14270	26830	43300
	B	15	18	21	25	31	34	41	46	54	60	66	78	90	110	131	149
400	A	8.75	16.2	34.7	66.2	140	212	417	676	1220	1820	2570	4780	7910	16790	31580	51000
	B	18	21	26	30	36	40	48	54	64	71	78	92	105	129	154	175
500	A	9.85	18.3	39.2	74.8	158	239	471	765	1380	2060	2910	5410	8970	19040	35840	57880
	B	20	23	29	33	41	45	54	62	72	80	88	104	119	147	174	199
600	A	10.9	20.2	43.2	82.5	174	264	521	846	1530	2280	3220	5990	9930	21100	39740	64210
	B	22	26	32	37	45	50	60	68	80	89	97	115	132	162	193	221
800	A	12.7	23.6	50.5	96.5	204	310	610	992	1790	2670	3780	7030	11670	24820	46780	75620
	B	25	30	37	43	52	59	70	80	94	104	114	135	155	191	228	260

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 deg the capacities shown in this table are to be multiplied by 1.5.

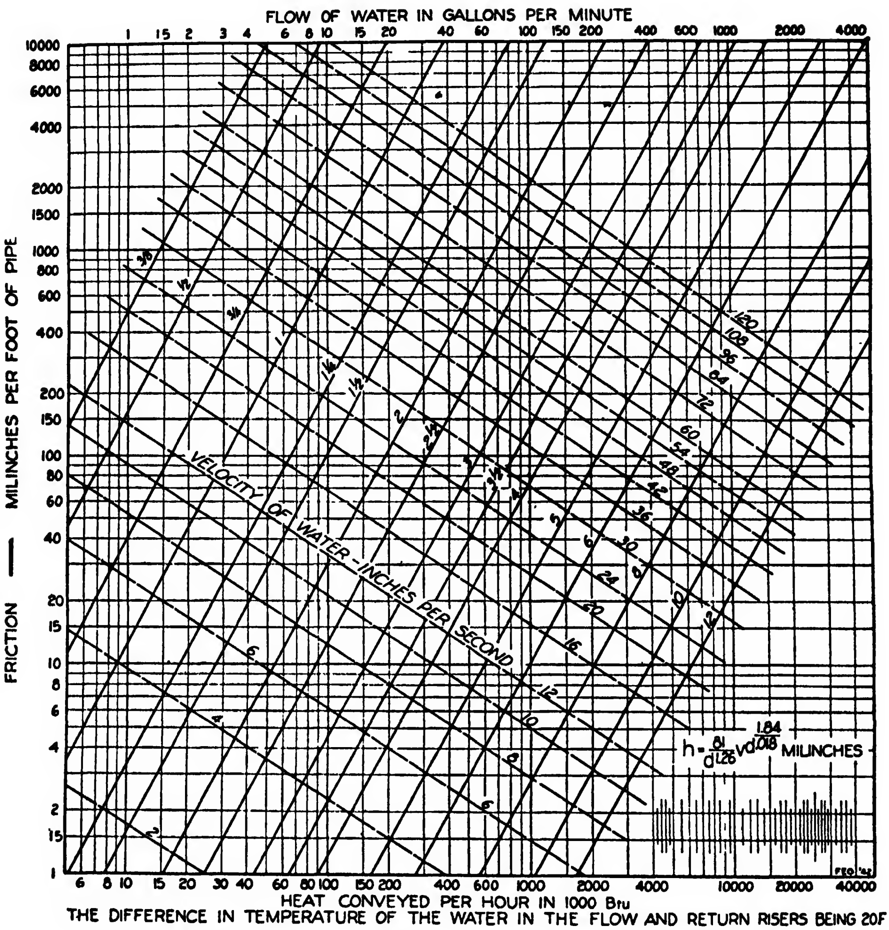


FIG. 3. FRICTION IN BLACK IRON PIPES

To find friction when temperature drop is other than 20 deg, multiply the actual heat conveyed by $\left(\frac{20}{\text{actual temp drop}}\right)$ and read the corresponding friction.

The friction loss in black iron pipes for a determined weight of water may be obtained from Fig. 2. For the frequently used temperature drop of 20 deg the friction loss may be determined directly from the heat requirement by means of Table 2 or Fig. 3 for black iron pipes, or by means of Table 3 for copper tubing.

Orifices drilled in plates inserted in pipe unions are convenient means for introducing friction, where required to balance various circuits. The friction losses caused by various sizes of orifices are given in Table 4.

CLASSIFICATION OF SYSTEMS

Gravity or *forced* systems of piping may be classified according to piping arrangement and type of circulation as shown in Table 5. Flow and return main piping (gravity systems) for one-pipe, two-pipe direct return, and two-pipe reversed return systems are shown in Figs. 4, 5, and 6 re-

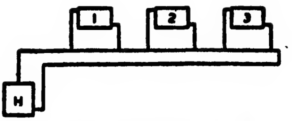


Fig. 4. One-Pipe System

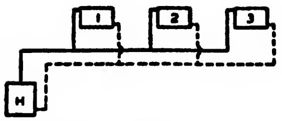


Fig 5. A Two-Pipe Direct Return System

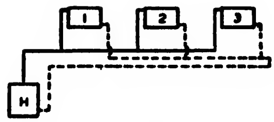


Fig. 6. A Two-Pipe Reversed Return System

spectively. These figures would also illustrate forced circulation if a pump or circulator were shown in the return line at the boiler.

One-pipe gravity systems require very precise design owing to the small circulating head available. Also, circulation in them is slow, and temperature drop is large toward the end of the main, and consequently these systems are usually considered impractical.

One-pipe forced systems compared with gravity systems provide more rapid circulation with consequent smaller temperature drop in mains and more uniform water temperature in all radiators and are therefore preferred. Special flow and return fittings are available for improving the circulation to risers.

Two-pipe systems have separate flow and return mains. If the return main is *direct* as shown in Fig. 5 the radiator at the end of the system has

Table 3. Heat-carrying Capacity of Type L Copper Tubing with Temperature Drop of 20 Deg[a]

Nominal Tube Sizes ⅜ in. to 4 in., and Friction 60 to 720 milinches per foot. (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Nominal Tube Size, In.		Milinch Friction Loss per Foot of Tube											
		720	600	480	360	300	240	180	150	120	90	75	60
⅜	A	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
	B	27	24	21	18	16.5	14	13	11	10	8.5	8	7
½	A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
	B	35	30	25	21	19	17	15	13	12	10	9	8
⅝	A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	0	7.9
	B	37	34	30	24	21	19	17	15	13	11	10	9
¾	A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
	B	42	38	33	27	24	21	19	17	14	12	11.5	10
1	A	104	94	82	70	63	56	47	42	37	32	28	25
	B	48	45	39	34	30	25	22	19	17	14.5	13	12
1¼	A	185	169	149	125	112	100	84	75	66	56	50	4[illegible]
	B	55	51	45	39	35	30	25	22	19	17	15	1[illegible]
1½	A	300	270	235	200	180	160	134	120	105	90	81	71
	B	62	57	51	43	39	35	30	25	22	19	17	15
2	A	625	560	495	420	375	335	280	250	200	188	170	150
	B	76	68	59	51	47	42	36	32	27	22	20	18
2½	A	1130	1010	890	750	680	600	500	450	395	335	305	270
	B	90	80	69	58	49	47	42	37	33	26	23	21
3	A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
	B	98	90	80	66	59	52	47	42	36	30	27	23
3½	A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
	B	110	100	89	75	66	57	51	45	40	35	30	26
4	A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
	B	120	108	96	83	73	63	55	49	44	37	34	29

[a] For other temperature drops the pipe capacities may be changed correspondingly. For example, with temperature drop of 30 deg the capacities shown in this table are to be multiplied by 1.5.

TABLE 4. FRICTION (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

(One milinch equals 0.001 in.)

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Inches per Second									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

the longest supply and longest return piping. The lengths of circuits to the various radiators may be equalized by using a *reversed return* main, (see Fig. 6). In some cases *reversed return* mains require no more piping than *direct return* systems.

With *gravity* circulation and *direct* return piping it is necessary to design the longest circuit for the available circulating head and to obtain the same resistance in all other circuits by proper selection of pipe sizes, by addition of fittings, or by use of orifices. When a *reversed return* system is used, it is usually found that but little adjustment is required to attain uniform distribution to all radiators.

Forced circulation in two-pipe systems, because of increased available circulating head, permits design for higher velocities with a consequent reduction in pipe sizes. The increased velocity also shortens the heating-up period and facilitates control of circulation. Reversed return mains are also advantageous in forced circulation systems in equalizing piping resistance to all heating units.

TABLE 5. CLASSIFICATION OF HOT WATER HEATING SYSTEMS

PIPING ARRANGEMENT	TYPE OF CIRCULATION	EXPANSION TANK	
One-Pipe	Gravity	Open	Closed
	Forced	Open	Closed
Two-Pipe Direct Return	Gravity	Open	Closed
	Forced	Open	Closed
Two-Pipe Reversed Return	Gravity	Open	Closed
	Forced	Open	Closed

PIPING SYSTEM DESIGN

In designing hot water heating systems certain assumptions are usually made for the purpose of simplification as follows:

1. Water temperature drop is assumed to be 30 to 35 deg for gravity systems and 20 deg for forced circulation systems. These values usually result in economical design but, particularly in large forced circulation systems, it is necessary to take into account the cost of pumping the water required at various velocities in relation to the annual charges in the capital cost of the system.

2. Water velocities in forced systems in excess of 4 fps are likely to cause disturbing noises in buildings other than factories.

3. Design outlet water temperatures in gravity systems are generally selected between 140 and 200 F (with the average approximately 180 F); while forced circulation design temperatures vary from 170 to 220 F, although higher temperatures can be used if the pressure in the system corresponds.

4. For forced circulation systems, the allowable friction loss, which is based upon the available circulating head, is determined partially by the characteristics of the pumps available.

5. Forced hot water system friction should usually be held between 600 and 250 milinches per foot. Above 600 milinches high velocities would be encountered and below 250 milinches circulation would become too slow so that much of the rapid response expected from forced circulation would be lost.

The water to be circulated is

$$W = H/(C \Delta t) \tag{2}$$

where

W = weight of water, pounds per hour [gallons per minute = $W/(8 \times 60)$].

H = heat required, Btu per hour.

C = specific heat of water (= 1).

Δt = drop in temperature between supply and return, Fahrenheit degrees.

The following graded series of examples of the design of hot water piping systems will illustrate the fundamental principles and methods. The differences between reversed return and direct return systems are shown, and the methods of balancing the several radiators or circuits are illustrated. A simple gravity system is shown in Fig. 7 and an elementary forced circulation system is diagrammed in Fig. 8.

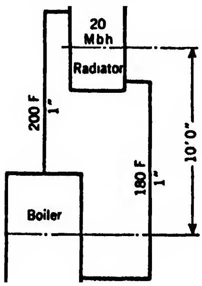


FIG. 7. GRAVITY SYSTEM

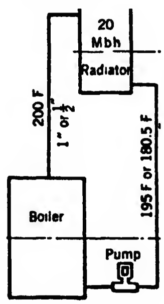


FIG. 8. FORCED CIRCULATION SYSTEM

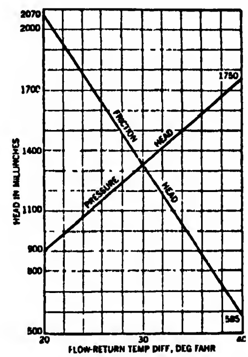


FIG. 9. DETERMINATION OF REQUIRED TEMPERATURE DIFFERENCE

Elementary Gravity System

Example 1. A simple gravity-circulation system is illustrated in Fig. 7 with one radiator that is giving off heat at the rate of 20,000 Btu per hr or 20 Mbh. The boiler imparts heat to the water at the same rate, and the water circulates at a uniform velocity. This uniform velocity is such that the friction of the circuit is equal to the head developed by the difference in density between the supply and return water and the height of the system. The circuit consists of 1 boiler, 1 radiator, 2 ells, 1 radiator valve and a total of 24 ft of pipe.

Solution. With the average water temperatures of 200 and 180 F in the supply and return risers, respectively, the head will be 90 milinches per foot of water column. This head may be found from Fig. 1. Since the center of the radiator is 10 ft above the center of the boiler, the total head of the circuit is 10 x 90, or 900 milinches, or 0.9 in. of 190 F water. The friction of the circuit must then also be 900 milinches. The friction of 1 ft of 1 in. pipe is found from Fig. 3 to be about 46 milinches at 20 Mbh, and the corresponding velocity 9 in. per second. (Note that all values in Fig. 3 are based on a temperature difference of 20 deg.)

Similarly, if a 1¼ in. pipe were to be used, the friction head would be about 12 milinches per foot and the corresponding velocity about 5 in. per second, from Fig. 3.

To find the friction in the elbows, boiler, radiator, and valve, Table 1 is used, and the entire circuit is found to be equal to 10 elbow-equivalents plus 24 ft of pipe. Each elbow-equivalent is equal to a pipe length of 25 times the nominal diameter. Then the equivalent lengths of straight pipe are 45 ft of 1 in. pipe or 50 ft of 1¼ in. pipe. In many cases, it is sufficiently accurate to add 50 per cent to the total pipe length to correct for resistance of fittings.

Hence, if 1 in. pipe is used, the friction of the circuit will be 45 × 46, or 2070 milinches, and if 1¼ in. pipe is used, the friction will be 50 × 12, or 600 milinches. A 1 in. pipe would, therefore, be too small and a 1¼ in. pipe too large to permit the desired circulation with a flow-return temperature difference of 20 deg.

If the circuit is of 1 in. pipe, the circulation will take place with a temperature difference greater than 20 deg, and if the circuit is of 1¼ in. pipe, the circulation will take place with a temperature difference smaller than 20 deg. To find, for example, the temperature difference at which a circuit of 1 in. pipe would transmit the required 20 Mbh, assume the difference to be 40 deg.

From Fig. 1, the head available for producing circulation would be 175 milinches per foot or 1750 for the system for a temperature drop from 200 to 160 F. The friction of the system may be found from Fig. 3; the chart of this figure is based on a temperature difference of 20 deg; if the temperature difference were 40 deg, the heat conveyed would be twice that shown in the chart. Hence, find 10 Mbh on the lower scale, proceed vertically upward to the intersection with the 1 in. line, and from there to the left scale read 13 milinches per foot. Note that the velocity would then be only about 5 in. per second. The total friction would then be 45 × 13 or 585 milinches. Since the head would be 1750, circulation would take place with a temperature difference less than 40 deg. The required temperature difference may be determined by constructing the diagram of Fig. 9, from which it appears that the temperature difference with which the 1 in. pipe circuit would function is about 30 deg. Hence, if the

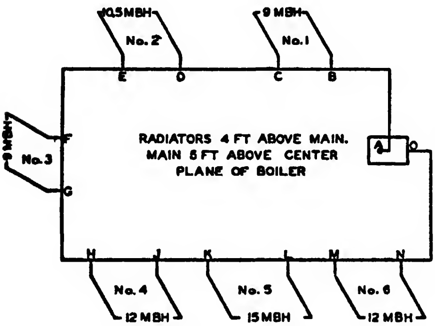


FIG. 10. ONE-PIPE GRAVITY CIRCULATION SYSTEM (EXAMPLE 3)

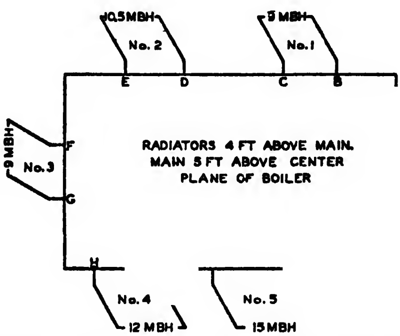


FIG. 11. ONE-PIPE FORCED CIRCULATION SYSTEM (EXAMPLE 4)

flow riser temperature is 200, the return riser temperature will be 170, and the average water temperature in the radiator about 185 F.

Elementary Forced Circulation System

Example 2. Design a system for the piping arrangement shown in Fig. 8, according to one of the outlined procedures. The procedure may be as follows: Assume the head developed by the circulating pump and the pipe size and find the flow-return temperature difference; or, assume the head developed by the pump and the flow-return temperature difference and find the pipe size; or, assume the pipe size and the flow-return temperature difference and find the head which the circulating pump must develop.

Solution. Assume that the circulating pump will develop a head of 2 ft or 24,000 milinches and that a 1 in. pipe is to be used. The equivalent length of the circuit will then be 45 ft, as in Fig. 7, and the available head will be 24,000/45, or 533 milinches per foot. In Fig. 3, find 533 on the left scale, move horizontally to the intersection with the 1 in. pipe line, and read about 77 Mbh delivered by the pipe (with a velocity of about 35 in. per second) for a temperature difference of 20 deg. Since the circuit is to deliver only 20 Mbh, the temperature difference will be 20 divided by 77 and multiplied by 20, or 5.2 deg. Hence, if the flow riser temperature is 200, the return riser temperature will be about 195, and the average water temperature in the radiator about 197.5 F.

If a ¾ in. pipe were used instead of a 1 in., the equivalent length of circuit would be 35 ft instead of 45; the unit head, 686 milinches instead of 533; the velocity, 27 in. per

second instead of 35; the temperature difference, 19.5 instead of 5.2; and the average water temperature in the radiator, about 190.5 instead of 197.5 F.

If the 1 in. pipe is used for the circuit, the gravity head will be 22 milinches per foot, or 220 for the circuit (Fig. 1, 200 to 195). Since this is only 1 per cent of the pump head (24,000 milinches), it may be neglected in the calculation, as was done previously. However, there are cases in which the gravity head is so large compared with the pump head that it should be included in the calculation.

The methods just described for the design of the two elementary systems are fundamental and apply to the design of all hot water heating systems. In every system, however large and complicated, the pipe system must be such that the head forcing the water from the boiler to any one radiator is equal to the friction in that radiator's circuit when the radiator is receiving its proper quantity of hot water and the system is functioning at a steady rate.

Other examples illustrating design of various systems follow.

One-Pipe Gravity Circulation System

Example 3. Select pipe sizes for the one-pipe gravity system having a total load of 67,500 Btu, shown in Fig. 10. Assume: flow temperature 190 F, return temperature 160 F, mains 5 ft above datum plane of boiler, center plane of radiators 4 ft above the mains, length of main 100 ft.

Solution: From Fig. 1 the available circulating head for 190 F flow and 160 F return temperature is 126 milinches per foot of height. The available circulating head for design of the main is therefore $5 \times 126 = 630$ milinches. The measured length of main plus 50 per cent added for resistance of fittings equals 150 ft equivalent length.

The main can then be designed for a friction loss of $630 \div 150 = 4$ milinches per foot. From Table 2 at 4 milinch friction loss, a 2 in. pipe will supply 33 Mbh and a 2½ in. pipe will supply 53.1 Mbh at 20 deg drop. This is equivalent at 30 deg drop to 49.5 Mbh for 2 in. and 79.6 Mbh for 2½ in. pipe. A 2½ in. main will therefore be selected and the pressure drop will be somewhat less than 4 milinches per foot.

The piping from main to radiators is sized in a similar manner. Assume that water reaches point B, Fig. 10, at 190 F and has a 30 deg drop in the radiator circuit. From Fig. 1 the available head is 126 milinches per foot of height or a total of $4 \times 126 = 504$ milinches for the circuit (with the radiator 4 ft above the main).

The measured length of piping is 11 ft and the fittings add 14 elbow equivalents (which would be equivalent to 22 ft if the pipe size is assumed to be ¾ in.); the equivalent length is therefore 33 ft. The circuit can therefore be designed for a friction loss of $504 \div 33 = 15$ milinches per foot.

From Table 2 by interpolation a ¾ in. pipe would supply 5.85 Mbh at 20 deg drop or 8.78 Mbh at 30 deg drop. Since the load is 9 Mbh the ¾ in. size will be satisfactory.

The remaining radiator circuits may be sized in a similar manner. Allowance should be made in one-pipe gravity systems for the drop in temperature which occurs in the supply main as the cooler water returns from the radiators. The drop will be in the same proportion to the total drop of 30 deg which the load supplied to any point in the main bears to the total system load; *e.g.* the temperature at D will be $190 - \left(\frac{9000}{67,500} \times 30\right) = 186$ F. At point F the temperature will be $190 - \left(\frac{9000 + 10,500}{67,400} \times 30\right) = 181$ F.

One-Pipe Forced Circulation System

Example 4. Select pipe sizes for the one-pipe forced circulation system having a load of 67,500 Btu shown in Fig. 11. Assume a water temperature drop of 20 deg. The water temperature does not affect the size of piping but does affect the radiator sizes required.

Solution: The water to be circulated at 20 deg drop will be $67,500 \div 20 = 3375$ lb per hour or $\frac{3375}{8 \times 60} = 7$ gpm.

By reference to manufacturers' pump capacity charts (typical example, Fig. 12),

it will be found that a 1 in. pump will deliver 7 gpm against a head of 4½ ft (54,000 milinches).

Since the main from A to O has an equivalent length of 150 ft (100 ft actual length plus 50 per cent added for friction loss in fittings), the main may be sized for 54,000/150 = 360 milinches per foot.

From Table 2 by interpolation at 360 milinches friction loss and at 20 deg drop a 1 in. pipe would supply 62,600 Btu per hour and a 1¼ in. pipe would supply 131,600. Since the 1 in. pipe is too small, a 1¼ in. pipe will be used.

Since the 1¼ in. pipe offers less than 360 milinches resistance per foot, the velocity of water will increase until the output of the pump and the friction loss are in equilibrium at some point on the pump performance curve, for instance, at 10 gpm and a head of 4 ft or 48,000 ÷ 150 = 320 milinches per foot of pipe. The friction loss in the main between flow and return connections to radiators will be assumed to be 320 milinches per foot.

In determining sizes for the piping from the main to any radiator, the resistance in the radiator circuit such as B-C (which has a load of 9 Mbh) is made equal to the

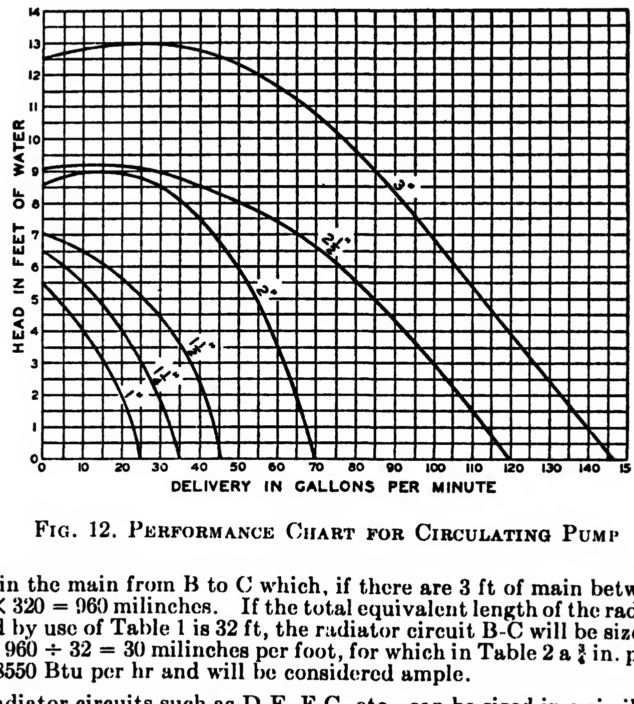


Fig. 12. Performance Chart for Circulating Pump

resistance in the main from B to C which, if there are 3 ft of main between connections, is 3 × 320 = 960 milinches. If the total equivalent length of the radiator circuit determined by use of Table 1 is 32 ft, the radiator circuit B-C will be sized for a friction loss o 960 ÷ 32 = 30 milinches per foot, for which in Table 2 a ¾ in. pipe is found to supply 8550 Btu per hr and will be considered ample.

Other radiator circuits such as D-E, F-G, etc., can be sized in a similar manner.

Two-Pipe Gravity System (with Reversed Return)

Example 5. Select pipe sizes for the two-pipe gravity system shown in Fig. 13. The center plane of the highest radiator is 8 ft above the center plane of the boiler. Assume a 180 F flow temperature and a 150 F return temperature.

Solution. The piping should be sized so that the frictional resistance at the desired rate of flow is equal to the available circulating head.

From Fig. 1 at 180 F flow and 150 F return temperature the available head is 118 milinches per foot of height or 8 × 118 = 944 milinches total for the highest radiator. The longest circuit from boiler to radiator and back to boiler must therefore have a resistance of 944 milinches. The longest circuit (see Fig. 13) is A-D + D-H + H-N containing 38 ft of pipe and, if 50 per cent is added for equivalent length of fittings, the equivalent length is 57 ft.

The circuit should then be designed for a friction loss of $944 \div 57 = 16$ milinches per foot (approximately).

The pipe size may be found from Table 2 at 16 milinches per foot as follows:

Section	Load Mbh	Size of Pipe for 16 Milinches per Foot	Section	Load Mbh	Size of Pipe for 16 Milinches per Foot
A-B	58	2	G-H	11	1
B-C	31	1½	H-K	19	1¼
C-D	20	1¼	K-L	25	1¼
D-E	16	1¼	L-M	31	1½
E-F	6	¾	M-N	58	2

Pipe sizes to the radiators may be sized for the same resistance per foot. From Table 2 at 16 milinch per foot the sizes will be selected as follows:

Radiator	#1	#2	#3	#4	#5
Load, Mbh	11	8	6	6	24
Pipe size, In.	1	1		¾	1¼

A hot water heating system will adjust its rate of flow until the friction loss balances the available head. It is therefore self-correcting in regard to small errors made in selection of pipe sizes.

Two-Pipe Forced Circulation System

Example 6. Select pipe sizes for the two-pipe forced circulation reversed return system having a total load of 159 Mbh shown in Fig. 14. Assume a difference of 20 deg in supply and return water temperature. The total equivalent length of the longest circuit is 180 ft. The gravity circulating head due to difference in temperature may be disregarded in design.

Solution. The water to be circulated is $159{,}000 \div 20 = 7950$ lb per hr or $\frac{7950}{60 \times 8} = 16.5$ gpm. From a pump performance chart such as Fig. 12 it is found that 16.5 gpm will be delivered by a 1 in. pump against a 3 ft head (36,000 milinches) or a 1¼ in. pump against a 4.5 head (54,000 milinches).

The longest circuit including the supply and return main and the longest radiator circuit is 120 ft and, if 50 per cent is added for friction loss in fittings, the equivalent length is 180 ft. If the 1 in. pump is used, the piping will be sized for $36{,}000/180 = 200$ milinches per foot, resulting in selection from Table 2 of a 2 in. main for the Section A-B which supplies 159 Mbh. The large difference in pump and main size, as well as the low velocity resulting from the 200 milinch per foot friction loss, indicates that the 1¼ in. pump should be considered. The design friction loss, if the 1¼ in. pump is used, can be $54{,}000/180 = 300$ milinches per foot and at this friction loss Table 2 will indicate the pipe sizes for the various sections in Fig. 14 as follows:

Supply			Return		
Section	Mbh	Pipe Size, In.	Section	Mbh	Pipe Size, In.
A-B	159	1½	J-K	16	¾
B-C	91	1¼	K-L	28	¾
C-D	75	1¼	L-M	42	1
D-E	63	1¼	M-N	54	1
E-F	49	1	N-O	75	1¼
F-G	37	1	O-P	91	1¼
G-H	16	¾	P-Q	159	1½

The radiator circuits may also be sized for the same friction loss, 300 milinches per foot, using Table 2 as follows:

Radiator	#1	#2	#3	#5 and #6	#7
Load, Mbh	16	12	14	21	16
Pipe size, In.	¾	½	½	¾[a]	¾

[a] Where circuit divides, use ¾ in. branch to #5 and ½ in to #6 radiator.

EXPANSION TANKS

Water heated from 40 F to 200 F expands about 0.04 of the original volume. The expansion tank permits the change in volume of the water

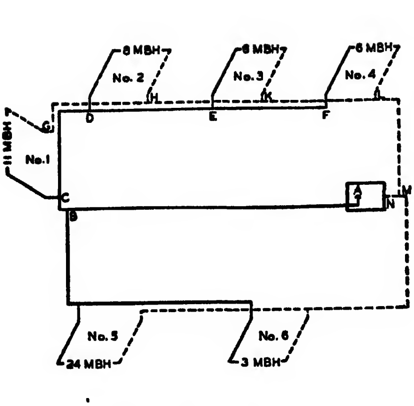


FIG. 13. TWO-PIPE REVERSED RETURN GRAVITY SYSTEM (EXAMPLE 5)

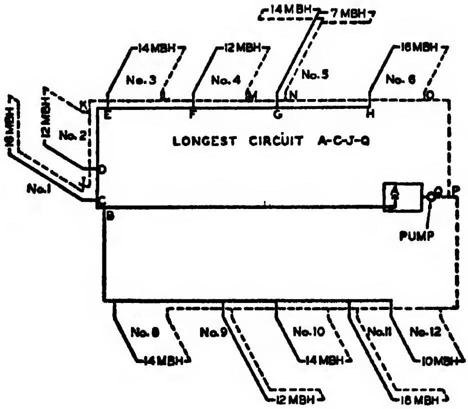


FIG. 14. FORCED CIRCULATION TWO-PIPE REVERSED RETURN SYSTEM

in the heating system to take place without producing undesirable stresses due to pressure in any part of the system. Expansion tanks may be open, as illustrated in Fig. 15, or closed as shown in Fig. 16. An open expansion tank has free vent to the atmosphere and consequently the pressure on the surface of the water is always that of one atmosphere. The minimum contents of an open tank should be 0.06 of the volume of the water in the system including that in the boiler, heat transmitters, pipes, etc. This capacity is 50 per cent in excess of the actual increase in volume of water due to increase in temperature from 40 F to 200 F. The tank should be located at least 3 ft above the highest radiator. Provision must be made

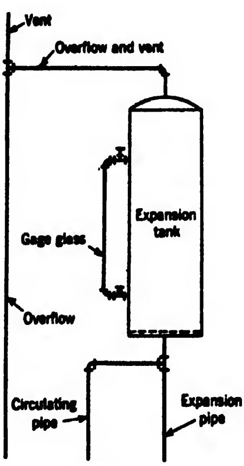


FIG. 15. AN OPEN EXPANSION TANK

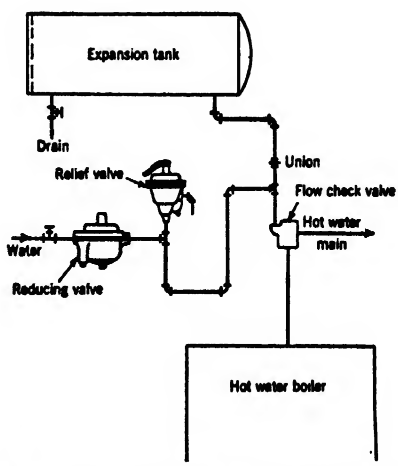


FIG. 16. A CLOSED EXPANSION TANK

to prevent freezing of the water in the tank as well as in the pipe leading to the tank.

When the vent from an open expansion tank is extended through the roof, it should be not less than 4 in. in diameter from a point below the roof, through and beyond the roof line. This will prevent vapor, which sometimes rises from an expansion tank, from closing the vent during outside freezing temperatures.

In a gravity circulation system, the pipe to the open expansion tank should be connected to the supply riser from the boiler, so that the air liberated from the water in the boiler will enter the expansion tank.

In a forced circulation system, the pipe to an open expansion tank should be connected on the suction side of the circulating pump.

A closed expansion tank is sealed against free venting to the atmosphere. The tank may be above the highest radiator or heat transmitter, or may be below the lowest one. The minimum contents of a closed expansion tank must be such that the expansion of the water due to increase in temperature will be cushioned against a reservoir of compressed air above the water level in the expansion tank. The tank must provide space not only for the change in water volume, but also for variations in air volume within the tank due to changes in air pressure. If the closed expansion tank is below the heat transmitters, the tank should be larger than if it is above them, and the higher the building, under such circumstances, the larger should be the air capacity in excess of that required for increase in water volume due to temperature rise.

The size of an expansion tank for installation in a closed system may be determined by the following formula:

$$V = \frac{E}{\dfrac{P_1}{P_1 + 0.434H} - \dfrac{P_1}{P_2}} \tag{3}$$

where

V = required tank capacity, gallons.

E = expansion of water from cold system to flow riser temperature, gallons.

P_1 = atmospheric pressure, psia.

P_2 = maximum tank pressure specified for heated system, psia.

H = height of top of filled system above tank, feet. (NOTE: Top of system open to atmosphere when system is filled.)

Example 7. Select a closed expansion tank for basement installation on a system containing 5000 gal and operating at 200 F flow temperature. The static head due to the height of the system is 70 ft. The maximum pressure should not exceed 100 psig.

Solution. Assume that the system is filled at 40 F. Then

$$E = 0.04 \times 5000 = 200 \text{ gal}$$
$$P_1 = 14.7 \text{ psia}$$
$$P_2 = 100 + 14.7 = 114.7 \text{ psia.}$$

Substituting these values in Equation 3:

$$V = \frac{200}{\dfrac{14.7}{14.7 + 0.434\,(70)} - \dfrac{14.7}{114.7}} = \frac{200}{0.327 - 0.128} = 1000 \text{ gal.}$$

The size of a basement-located closed expansion tank should be at least equal to the following:

One story buildings: $x = 0.10\ V$ Three story buildings: $x = 0.17\ V$

Two story buildings: $x = 0.13\ V$ Four story buildings: $x = 0.23\ V$

where

x = expansion tank size in gallons.

V = water volume in heating system in gallons.

This condition favors, especially in tall buildings, the placing of the closed expansion tank above the highest heat transmitter.

It is common practice to use multiple tank installations on large systems in lieu of one tank the required capacity of which is beyond commercially available sizes.

Any closed expansion tank located above the heat transmitters of a hot water heating system should be connected by a direct pipe with the flow main leaving the boiler, in order to enable the air to pass easily to the expansion tank. In a closed hot water heating system the water under pressure tends to absorb air at a rate increasing with pressure increase and decreasing with temperature increase.

Table 6. Volume of Water in Standard Pipe

Pipe Size, In.	Lineal Ft of Pipe Containing 1 Gal	Pipe Size, In.	Lineal Ft of Pipe Containing 1 Gal
½	63.1	2	5.75
¾	36.1	2½	4.02
1	22.2	3	2.60
1¼	12.8	4	1.52
1½	9.47	5	0.96
		6	0.67

Means must be provided to adjust and to observe the proportion of air within any closed expansion tank. This involves the provision of an air inlet valve, a water gage, and a relief valve. A source of supply of compressed air for renewing the air cushion is highly desirable, especially in large, high pressure, hot water heating systems where it is inconvenient, if not impracticable, to drain down the water in the system so as to permit introduction of atmospheric pressure air.

For every hot water heating system the designer should calculate the volume of water contained in the radiators, piping system, boiler, etc., in order to select the proper size of expansion tank. The water content of the piping can be obtained from Table 6. For a rough selection of size, however, it is sometimes assumed that 50 per cent of the volume of water is contained in the radiators, and that the water content per square foot of radiator heating surface is 0.2 gal for column radiators and 0.13 gal for tube type radiators.

Another rough method for determining the size of an expansion tank to be located above the highest radiator is to divide the square feet of radiation by the factor 40 to obtain the required capacity in gallons of the tank.

INSTALLATION DETAILS

Items that should be considered in the design of this type of system are:

All piping must be so pitched that all air in the system can be vented either through an open expansion tank, radiators or automatic relief valves. When piping must be run around an obstacle such as a beam, it is advisable to drop the piping below the beam. If looped over the beam, it becomes necessary to provide for venting of air from the high point of the pipe.

When changing the size of horizontal runs of pipe, eccentric fittings should be used to keep the tops of the pipes in line to permit free passage of air along the pipe.

All piping must be arranged so that the entire system can be drained. Sections of piping individually valved shall have corresponding drain valves.

In large buildings, the piping may be zoned according to exposure of building, usage of building, or method of control.

All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

The pipe system should be designed so that each circuit has its correct friction for balanced water distribution. This may be done by change of pipe size or change in piping detail.

The connections from the boiler to the mains should be short and direct, to reduce the friction and should allow for expansion.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

Generally, connections to risers or radiators are taken out of the top of mains either 45 or 90 deg.

Supply connections are usually made at the bottom of radiators so that circulation will not be stopped by accumulation of air, as would be the case with a top supply connection. Short radiators are sometimes connected for top supply and bottom return on the same end. When so connected, attention must be given to venting of air from the top of the radiator somewhat oftener than when bottom connections are used.

Unless used as heating surface, all piping, both flow and return, should be insulated.

All large systems should be provided with extra stop and drain valves, suitably located so that parts of the system may be isolated for repairs without making it necessary to drain the water from the entire system.

ZONING

In large hot water systems improved control and economy can be achieved by separating the systems into sections or zones (vertical or horizontal) which can be operated independently of each other. Variations in heat requirement of the different zones as influenced by the exposure of

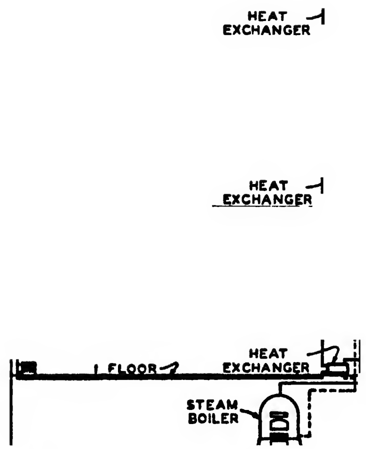


FIG. 17. VERTICAL ZONING OF HOT WATER HEATING SYSTEM IN A 12-STORY BUILDING

the building, solar heat, weather conditions, heat from processes, type of occupancy, building chimney effect, etc., can readily be compensated for when heat can be supplied only where needed.

In tall buildings, vertical zoning such as shown in Fig. 17 not only provides the advantages of control and economy, but also reduces the water pressure in the system to that caused only by the number of floors served by each section. As shown in Fig. 17 a steam boiler can conveniently be used to supply steam to the heat exchangers supplying heated water to each zone.

CHAPTER 25

RADIATORS, CONVECTORS, COILS

Heat Emission of Radiators and Convectors, Types of Radiators, Convectors, Radiator and Convector Ratings, Effect of Operating Conditions, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators, Coils, Coil Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangement, Applications, Heat Transfer and Air Flow Resistance, Coil Selection

RADIATORS and convectors are primarily used for heating and since their performance is affected by many of the same conditions, they will be considered together. Coils for heating and cooling when used in a fan system will be treated separately later in this chapter.

HEAT EMISSION OF RADIATORS AND CONVECTORS

Most heating units emit heat by *radiation* and *convection*. An exposed *radiator* emits roughly half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, the proportion of heat emitted by radiation is reduced. The balance of the emission occurs by conduction to the air in contact with the heating surface, and this heated air rises and causes circulation by convection transmitting this warm air to the space which is to be heated.

The output of a radiator can be measured only by the heat it emits and is generally expressed in units of: Btu per hr; *Mbh* (1000 Btu per hour); or in equivalent direct radiation (240 Btu per hour for steam or 150 for water radiation).

TYPES OF RADIATORS

Present day radiators are usually of tubular type and are generally made of cast-iron. The small-tube type of tubular radiators with a spacing of $1\frac{3}{4}$ in. per section are about the only available radiating surface for homes and office buildings. Small-tube radiators occupy less space and are particularly suited for installation in recesses.

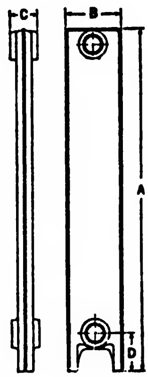
Baseboard radiation consists of long, low units which are made to resemble conventional baseboards and are installed along the outside walls of rooms in place of the usual wooden baseboard. Units are made either of hollow cast-iron panels (with, or without fins on the back) or of ferrous or nonferrous finned tubing installed behind a metal enclosure. They are primarily used in hot water systems, but may also be used on two-pipe steam systems.

Advantages claimed for baseboard radiators are: They are inconspicuous; they are clean in operation; they offer a minimum of interference with furniture placement, and, they distribute the heat near the floor. This reduces the floor to ceiling temperature gradient to about 2 deg F and tends to produce uniform temperatures throughout the room.

After a study of the demand for various sizes of radiators, the *Institute of Boiler and Radiator Manufacturers*, in cooperation with the Division of Simplified Practice, *National Bureau of Standards*, established Simplified

Table 1. Small-Tube Cast-Iron Radiators

Number of Tubes per Section	Catalog Rating per Section[a]	Section Dimensions				
		A Height[c]	B Width Minimum	B Width Maximum	C Spacing[b]	D Leg Height[c]
	Sq Ft	In.	In.	In.	In.	In.
3[d]	1.6	25	3¼	3½	1¾	2½
4[d]	1.6	19	4 7/16	4 13/16	1¾	2½
	1.8	22	4 7/16	4 13/16	1¾	2½
	2.0	25	4 7/16	4 13/16	1¾	2½
5[d]	2.1	22	5⅝	6 5/16	1¾	2½
	2.4	25	5⅝	6 5/16	1¾	2½
6[d]	1.6	14	6 13/16	8	1¾	2½
	2.3	19	6 13/16	8	1¾	2½
	3.0	25	6 13/16	8	1¾	2½
	3.7	32	6 13/16	8	1¾	2½



[a] The square foot of equivalent direct steam radiation is defined as the ability to emit 240 Btu per hour with steam at 215 F, in air of 70 F. These ratings apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See A.S.H.V.E. Code for Testing Radiators adopted January, 1927.)
[b] Length equals number of sections times 1¾ in.
[c] Over-all height and leg height, as produced by some manufacturers, are one inch (1 in.) greater than shown in Columns A and D. Radiators may be furnished without legs. Where greater than standard leg heights are required this dimension shall be 4½ in.
[d] Or equal.

Practice Recommendation R174-43 for small-tube cast-iron radiators. Table 1 shows the size and dimensions now being manufactured.

Wall radiators are now rated in terms of equivalent square feet, the same as small-tube radiators. Tests have shown that the heat emitted from a wall-type radiator may be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually such a large difference in the temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of the rooms satisfactorily.

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but are not often found in this service today. When

Table 2. Heat Emission of Pipe Coils Placed Vertically on a Wall (Pipes Horizontal) Containing Steam at 215 F and Surrounded with Air at 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

Size of Pipe	1 In.	1¼ In.	1½ In.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

coils are used, the miter type assembly is preferable as it readily permits expansion in the pipe.

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2 which has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

CONVECTORS

Cast-iron radiators may be concealed in a cabinet or other enclosure for appearance. In such cases a greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector. A

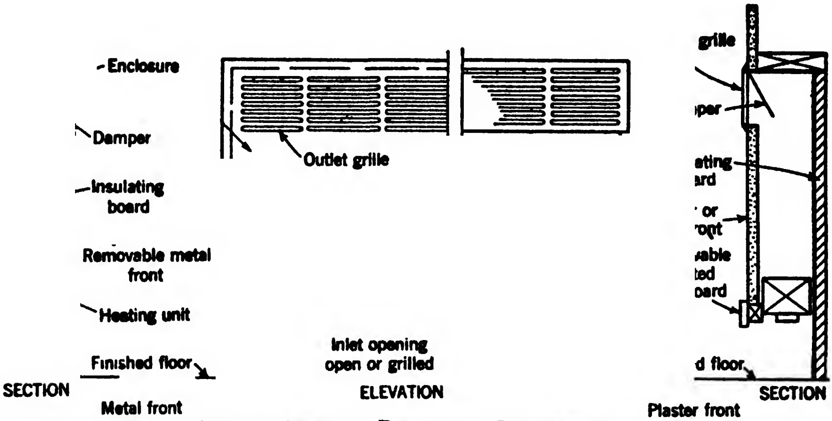


Fig. 1. Typical Recessed Convector

typical recessed convector is shown in Fig. 1. The heating element consisting of a large percentage of fin surface is usually shallow in depth and placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is heated moderately in passing through the core and is delivered to the room through an opening near the top of the enclosure. This air movement accomplishes a reduction in temperature differentials and tends to assure maximum comfort in the living zone.

Concealed heaters or convectors are generally available as completely built-in units. Combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is desirable that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated cannot by-pass the heating element in passing through the enclosure.

The output of a convector, for any given length and depth, is a function

of the height of the discharge grille above the heating element. Therefore the published ratings are generally given in terms of square feet of *Equivalent Direct Radiation*, EDR. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit.

RADIATOR AND CONVECTOR RATINGS

A standard method of testing radiators was adopted by the A.S.H.V.E. in 1927[1]. This Code provides for a standard test room, the temperature of which is to be maintained at 70 F, measured in the center of the room at an elevation of 5 ft above the floor. The steam temperature in the radiator is to be 215 F, which corresponds to 15.6 lb per square inch absolute.

TABLE 3. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTORS

Steam Press. Approx.		Heating Medium Temp F Steam or Water	Factors for Direct Cast-Iron Radiators — Room Temperature F							Factors for Convectors — Inlet Air Temperature F						
Gage Vacuum In. Hg.	Abs. Lb per Sq In.		80	75	70	65	60	55	50	80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1 62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1 52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10 9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
LbperSqIn.																
1	15.6	215	1.10	1.05	1 00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0 95	0.91	0.87	0.83	0.79	0.76
15	30	250	0 81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0 79	0.76	0 73	0.70	0 68	0.65
27	42	270	0.70	0 68	0.66	0.64	0.62	0 60	0.58	0.70	0.68	0.65	0 63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0 49	0 56	0 54	0.53	0.51	0 49	0.48	0 47

[a] To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacities at the basic conditions by the proper factor from the above table.

The weight of condensate per hour, under these standard conditions, multiplied by the difference in the enthalpy of the steam entering the radiator and that of the condensate leaving the radiator, gives the radiator output in Btu per hour. This output divided by 240 gives the steam rating of the radiator in equivalent square feet, EDR.

Similar test methods for convectors are the A.S.H.V.E. Codes for Testing and Rating Concealed Gravity Type Radiation[2], (Steam Code 1932 and Hot Water Code 1933). These Codes recognize a different type of test booth, and the air temperature used is that of the air entering the convector casing instead of the temperature in the center of the room. The entering air temperature for standard test conditions is 65 F. For hot water the standard test conditions call for a mean temperature of the water in the convector of 170 F.

The method of testing and rating both ferrous and non-ferrous convectors, which is now generally accepted, is given in Commercial Standard CS140-47, Testing and Rating Convectors, which has been developed cooperatively by the *Convector Manufacturers Association*, the *Institute of*

Boiler and Radiator Manufacturers, other members of the trade, and the *National Bureau of Standards*.

The rating of a top outlet convector is established at a value not in excess of the *condensation capacity* (which is the heat extracted from the steam or water in the convector, under standard test conditions). The rating of a front outlet convector includes the *condensation capacity* plus an allowance for heating effect in the occupied zone, based on convector enclosure height from bottom of the enclosure to top of the outlet. A table of heights and heating effect allowances is given in the Commercial Standard CS140-47, and lists allowances from zero per cent for a 38-in. height to 15 per cent for a 20-in. height.

For an inclined outlet convector the rating includes the *condensation capacity* plus a heating effect allowance obtained by multiplying the allowance for a front outlet convector by a factor (angle of outlet to horizontal ÷90).

Approval of convector ratings may be obtained by the manufacturer by submitting test data to a *Convector Rating Committee* composed of two members appointed by the *Convector Manufacturers Association* and the *Institute of Boiler and Radiator Manufacturers*, and one appointed by the Division of Trade Standards of the *National Bureau of Standards*. Requests should be addressed to the Division of Trade Standards.

Effect of Operating Conditions

The heat output of a radiator is proportional to the 1.3 power of the temperature difference between the air in the room at the 60 in. level and the heating medium in the radiator. The heat output of a convector is proportional to the 1.5 power of the temperature difference between the air entering the convector and the heating medium, steam or hot water, within the convector[3]. For hot water the arithmetical average between entering and leaving water temperatures is used. These laws may be expressed as correction factors to change from output under standard rating-test conditions to output under other operating conditions. Such factors are given in Table 3.

When it is desired to change the output under any test conditions to the corresponding output under standard Code test conditions, the reciprocal form of correction factor may be derived. The equations for steam units are:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3} \tag{1}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5} \tag{2}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{3}$$

where

C_s = correction factor.

t_s = steam temperature during test, Fahrenheit degrees.

t_r = room temperature during test, Fahrenheit degrees.

t_i = inlet air temperature during test, Fahrenheit degrees.

H_s = heat emission rating under standard conditions, Btu per hour.

H_t = heat output under test conditions, Btu per hour.

The relation between the size of the radiator or convector and the size of the test room will affect the results obtained in a capacity-rating test.[4] The height and location of the radiator and the insulation of the test room are other important factors that are not specifically regulated by the Code.

For a radiator, the finish coat of paint affects the heat output. Oil paints of any color will give about the same results as unpainted black or rusty surfaces, but an aluminum or a bronze paint will reduce the heat emitted by radiation. The net effect may be a reduction of 10 per cent or more in the total heat output of the radiator[5,6,7].

Radiator enclosures and convector casings affect the heat distribution within the room as well as the total amount of heat supplied by the steam or hot water[8].

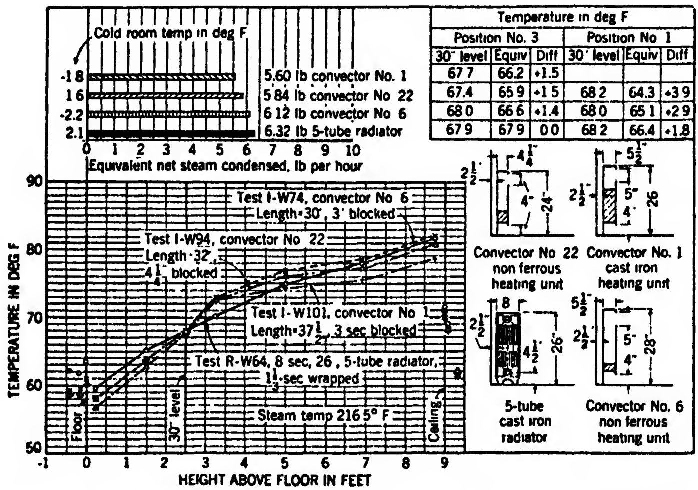


FIG. 2. TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR RADIATOR AND CONVECTORS WITH COMMON 30 IN. LEVEL TEMPERATURE

Heating Effect

For several years the term *heating effect* has been used to designate the relation between the *useful output* of a radiator, in the comfort zone of a room, and the total input as measured by steam condensation or water temperatures[9,10]. The application of such a heating effect factor is a recognition that some radiators and convectors use less steam than others for producing equal comfort heating results in the room.

No standard method for evaluating the heating effect of radiators and convectors and correlating it with comfort has yet been accepted. One method, with test data[11] on radiators and convectors, and making use of the eupatheoscope for evaluating the environment produced, has been suggested by the University of Illinois. The principle underlying the eupatheoscope involves the measurement of the heat loss from a sizable body by radiation and convection, when the surface is maintained at some constant temperature. Through the use of this instrument and its calibration curve, non-uniform environments may be referred to uniform environments in which the air and all surrounding surfaces are at the same temperature.

The temperatures of the uniform environments are referred to as equivalent temperatures.

The Kata thermometer[12], the thermo-integrator[13,14], and the globe[15] thermometer are other instruments which have been used to measure the influence of air temperature, air movement and radiation in an environment.

Data given in Fig. 2 show that while the air temperature at the 30-in. level is the same for the three convectors and the one large-tube cast-iron radiator, in position No. 3 in the test room, the equivalent temperature is 1.5 F lower than the air temperature in the case of the three convectors, and the same as the air temperature in the case of the radiator. The difference between the minimum and the maximum amount of heat re-

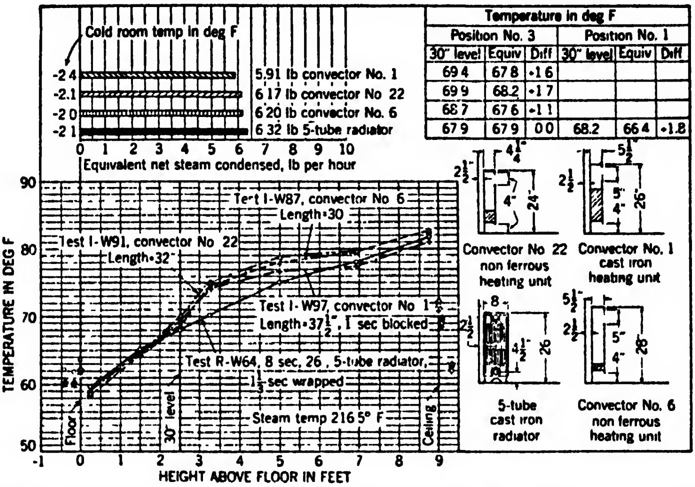


FIG. 3. TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR RADIATOR AND CONVECTORS WITH COMMON EQUIVALENT TEMPERATURE

quired to maintain the common air temperature at the 30-in. level is of the order of 13 per cent.

In Fig. 3 are shown the results of tests made with the same three convectors and the one large-tube cast-iron radiator, so adjusted in size that each gave approximately the same equivalent temperature in the No. 3 position in the test room. The difference between the minimum and the maximum amount of heat required to maintain the common equivalent temperature is of the order of 7 per cent.

The following statements applying to the use of radiators are based on experience and test results:

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.
2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.
3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.
4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing

line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath windows.

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Tests[16] on an old-style column-type cast-iron radiator indicated that in the first 10 min the condensation rate reached a peak of 0.95 lb per square foot of radiator per hour and 10 to 15 min later dropped to a rate of 0.24 lb. In practice the rate of steam supply to the heating unit, while heating up, is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard

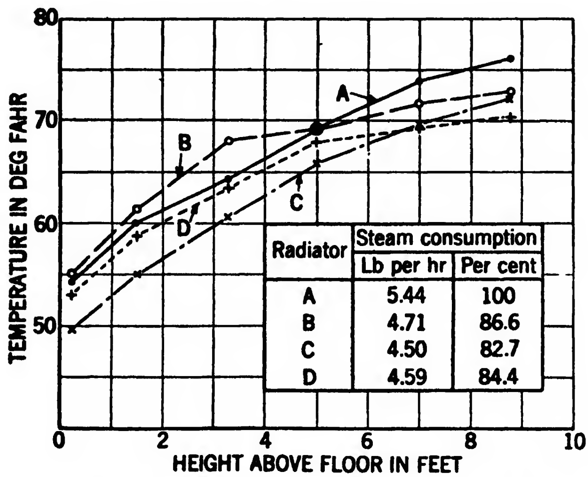


FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

the supply of steam. Vacuum types of air venting valves may be used to reduce the length of the venting periods.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Investigations[10] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (A) and the same radiator with a well-designed enclosure (B), with a poorly-designed enclosure (C), and with a cloth cover (D) will illustrate the relative heating effects. In Fig. 4 the curve (B) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating

effect. Curve (C) shows the unsatisfactory effects produced by improperly-designed enclosures. Curve (D) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Some commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[17] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when a radiator is steam heated and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

COILS

Coils described in this chapter are used for heating or cooling an air stream under forced convection. Surface coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the fluids used and the condition of the air handled, or by an economic analysis of the possible alternates on each installation.

For heating service, coils are used as tempering coils, preheaters, reheaters or booster heaters. The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases, such as reheating by means of discharge gas from a refrigerating system.

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water cooled coils to remove sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together as explained later. The assembly usually includes air cleaning means to protect the coil from accumulation of dirt and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, there are cases of cooling coils purposely wetted to aid in air cleaning and odor absorption.

The usual cooling media used in surface coils are cold water or Group I (*ASA* Classification) refrigerants, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering air temperatures with large latent heat loads require a refrigerant temperature so low that use of water becomes impracticable. Sometimes, also, brine from an industrial system already installed is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of

each should be considered. The fact that a spray dehumidifier is usually designed to deliver nearly saturated air tends to simplify the control problem. In this case the dry-bulb temperature is also the dew-point, and hence a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have an advantage over unwetted coils of obtaining some air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling water circuit, obviating the unbalanced pumping head, the complication of water level control, and danger from possible floods incidental to multiple spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in the one season and hot water in the other, with consequent saving in apparatus and piping. Another advantage is that where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. The safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case and no general rule can be given. There are many installations in which either may be used.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of plain tubes or pipe and those having *extended* surfaces. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling within spray dehumidifiers.

The heat transmission from air passing over a tube to a fluid flowing within it is impeded by three resistances. The first resistance is from the air to the surface of the tube, usually called the outside surface resistance or air-film resistance. Second is the resistance to the flow of heat by conduction through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction, and the inside surface or film resistance are usually low as compared with the air-side surface resistance. Economy in space, weight and cost makes it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from the tube to refrigerant. This may be accomplished by increasing the external surface by means of fins. Sometimes water spray is applied to the same type surface as would have been used without it. The over-all heat transfer is not necessarily increased much by such an arrangement, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning. (See Chapter 7.)

In fin or *extended* surface coils the external surface of the tubes is known as *primary* and the fin surface is called *secondary*. The primary surface consists generally of round tubes or pipes. In some cases these are staggered and in others in line with respect to the air flow. The staggered arrangement is usually preferred because it obtains a somewhat higher heat transfer value. Numerous types of fin arrangement are used, the most common of which are spiral, flat and flat-crinkled or corrugated, all as shown in Fig. 5. While the spiral fin surrounds each tube individually in all cases, the flat

types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types, are in use, the selection for a particular installation being based on economic considerations, space requirements and resistances of individual designs of coils. A most important factor in the performance of extended surface coils is the bond between the fin and the tube. An intimate contact between the tube and the fin must be maintained permanently in order to assure a continuing rated performance after the heating units have been in service for a period of time. In some coils fins are wound on the tubes under pressure in order to upset the metal slightly at the fin root and then are given a coating of solder while the fin and tube are still revolving for the purpose of assuring a uniform coating of solder. In other types the spiral fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube hole flanges of a flat or corrugated fin may be made

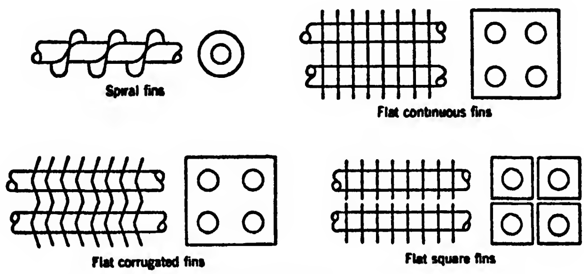


FIG. 5. TYPES OF FIN COIL ARRANGEMENT

to override those in the preceding fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself.

For *heating coils* materials most generally used are copper and aluminum. Steel is occasionally used where sodium or calcium chloride brine is circulated in the tubes. Aluminum fins with copper tubes is a common construction. Generally speaking, brass does not serve as a satisfactory fin material because of corrosion difficulties. *Cooling coils* for water or for volatile refrigerants most frequently have copper fins and tubes, although aluminum fins on copper tubes are also used. There are many makes of heating and cooling coils of the light weight extended surface type for both heating and cooling with tubes commonly $\frac{1}{2}''$, $\frac{5}{8}''$, $\frac{3}{4}''$, and $1''$ outside diameter and with fins spaced three per inch up to eight per inch. The tube spacing generally varies from about $1\frac{1}{8}''$ to $2\frac{1}{2}''$ on centers depending upon the width of individual fins and on other considerations of performance. Fin spacing should be chosen for the duty to be performed with special attention being paid to lint accumulation and especially in dehumidifying, the consideration of frost accumulation.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continuously eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifices in the steam

header, or by perforated internal steam-distributing pipes extending into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform temperature of delivered air. The tendency of condensate to freeze at the bottom of the coil with cold entering air and light heating loads is also minimized. This is especially valuable for outside air preheaters.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air

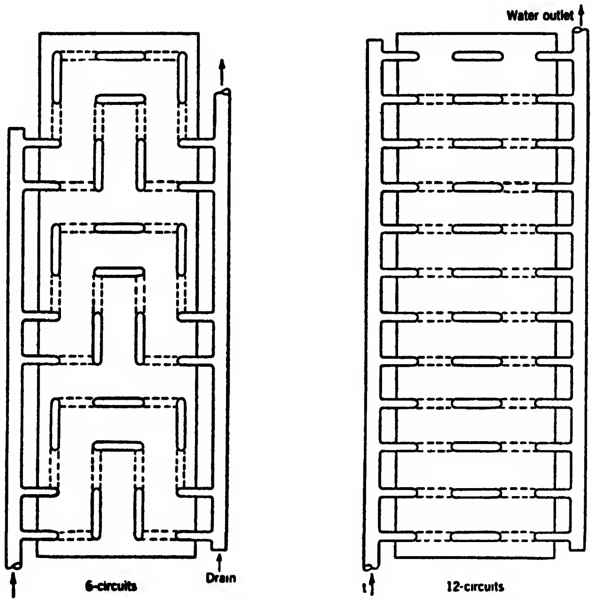


Fig. 6. Various Water Circuit Arrangements

elimination is taken care of in the system piping as described in Chapter 24. To assure a pressure drop sufficient for adequate distribution but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18, 24, or 36 parallel water circuits as conditions may require. Orifices in individual tubes are occasionally employed but are usually unnecessary as the resistance of individual water circuits is generally sufficient to effect a satisfactory distribution. In precooling coils using well water where there may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for complete drainage (see Fig. 6). The drains are usually provided in the water piping at the coil header.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. It is desirable that the coil

be effectively and uniformly cooled throughout, and necessary that the compressor be protected from entrained, unevaporated refrigerant. There are two types; namely, flooded systems, and thermal expansion valve systems, as shown in Figs. 7 and 8. In a flooded coil, the circulation is similar to that in a water tube boiler. The liquid is maintained at the proper level by the action of a float regulator as shown in Fig. 7. The thermal expansion valve system depends upon the thermal valve automatically feeding just as much liquid to the coils as is required to maintain the superheat at the coil suction outlet within predetermined limits which vary from about 6 to 10 deg. The thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is rarely used.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds and the head of liquid imposed upon the liquid inlets. No auxiliary distributing devices are required. With the thermal valve system there are two factors to consider.

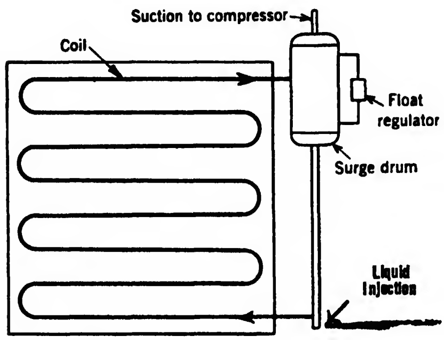


FIG. 7. DIRECT-EXPANSION COIL WITH FLOODED SYSTEM

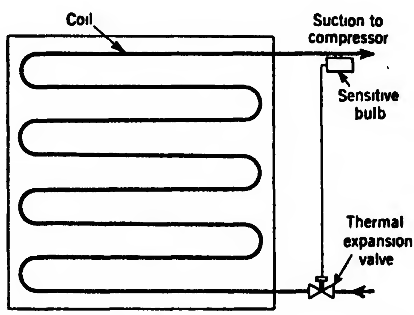


FIG. 8. DIRECT-EXPANSION COIL WITH THERMAL VALVE SYSTEM

There must be, generally, more than one refrigerant feed through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits and to reduce the corresponding penalty in increased evaporating temperature. At the same time the coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it handles the same refrigerating load. Here, a distributing means is imposed between valve and coil liquid inlets to divide the refrigerant equally among the feeds. Such a distributor must be effective for distributing both liquid and vapor, because the entering refrigerant is a mixture of the two. Fig. 9 shows three typical types of distributors. In distributor A the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor B the refrigerant is discharged at a high velocity through a central jet against the end plate, forming a uniform mixture of gas and liquid within the distributor, from which individual connections

are led as shown. In type C the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. There are also other types of headers such as the centrifugal and weir type. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing and are all of the same length and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Sometimes a single thermal valve is used per coil. In other cases multiple valves are used, with the coil divided across the air flow or parallel to the air flow as shown in Fig. 10. The arrangement of Fig. 11

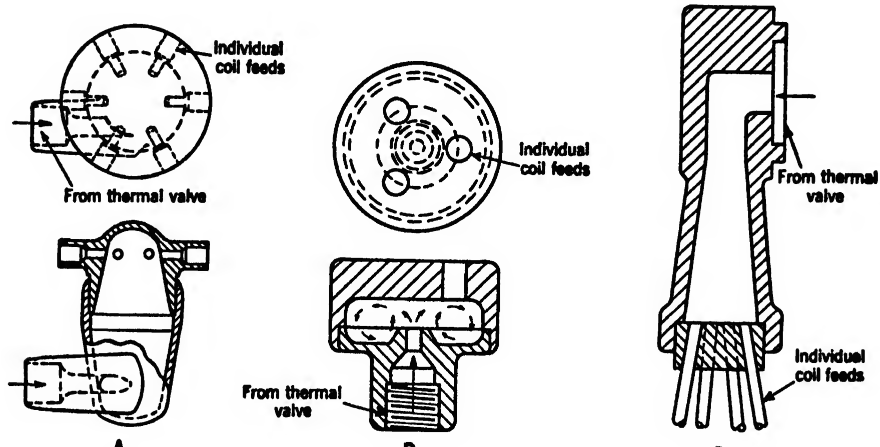


Fig. 9. Types of Refrigerant Feed Distributing Heads

should be avoided since it offers the disadvantage of unequal load on the two parallel circuits.

Flow Arrangement

The relative direction of flow of the air outside the tubes and the medium within them influence the performance of the surface. There are three types of relative flow in common use. Fig. 12A shows parallel-flow in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 12B shows counter-flow in which the medium in the tubes proceeds in a direction opposite to the flow of air. Fig. 12C shows cross-flow in which the air and the medium in the tubes pass at right angles to each other. The counter-flow arrangement is almost universally used in brine or water coils to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. It is also commonly used in coils fed with volatile refrigerant to take advantage of the higher air temperature for superheating the leaving gas. In deep coils, however, it is sometimes advantageous to use parallel flow from second row to last row and then to complete the circuit by passing through the first row to take advantage of the higher air temperature for superheating.

Complete evaporation and superheating of the refrigerant are essential to proper operation of the thermal expansion valve. Cross-flow is common in steam heating coils, the temperature within the tubes being substantially uniform and the mean temperature difference the same whatever the direction of flow, relative to the air. Cross-flow is to be avoided in coils with volatile refrigerants on account of unequal loading of parallel circuits and the danger of short circuiting of liquid refrigerant which disturbs proper functioning of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 43. They may be arranged with the air flow

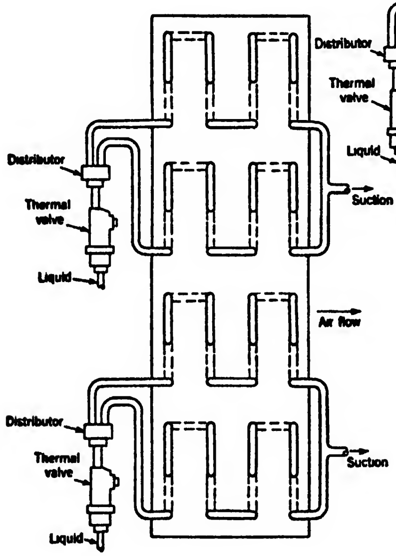


FIG. 10. ARRANGEMENT FOR FACE CONTROL

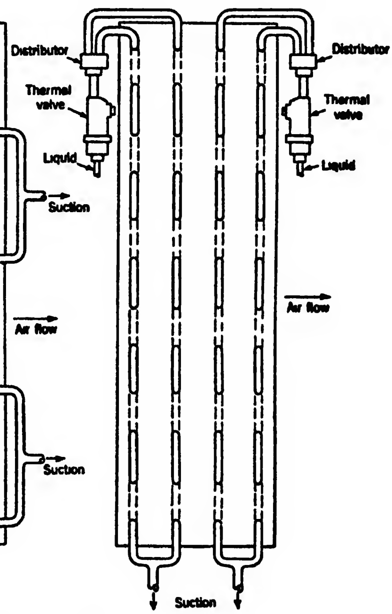


FIG. 11. ARRANGEMENT FOR DEPTH CONTROL

vertical or horizontal, although the latter is more common. For steam heating, the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken against freezing. Where steam coils are used with entering air below freezing temperature, throttling the steam supply may result in freezing the condensate in the bottom of the coil if the tubes are of the variety not provided with internal distributing pipes, or an equivalent arrangement.

There are coils available having inner distributing tubes and having the supply and return headers cast in one piece. In this type of coil the condensate that forms in the outer tube has resulted from steam fed from the inner tube orifices. This condensation flowing back along the warm inner tube is prevented from freezing. A wide range of modulation at very

low temperatures without danger of freezing is therefore obtained. As an added precaution with both steam and water coils the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of cooling coils is shown in Fig. 13. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty and to collect the non-evaporated water from the humidifying sprays in winter. The drip connection should be made ample in size and liberally provided with cleanout fittings. It should not be exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings and for cleaning the coils. When coils are used for dehumidifying, eliminators must be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm. Where a number of coil

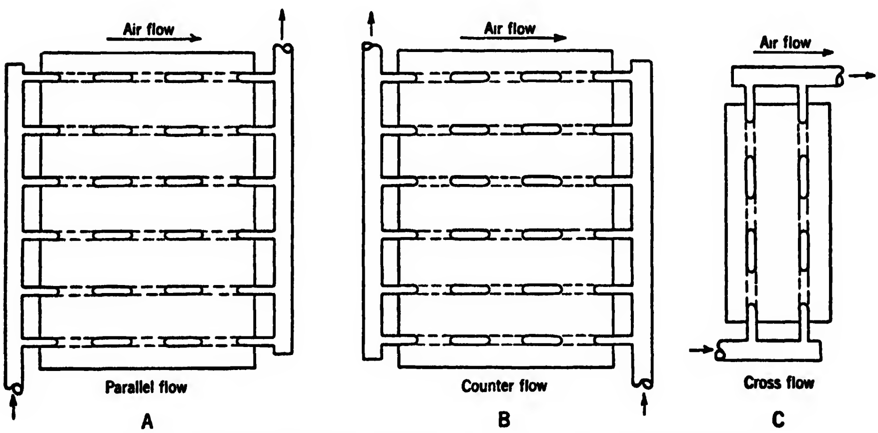


Fig. 12. Flow of Media in Tubes in Relation to Air Flow

sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 14 are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying duty are provided with water sprays. These sprays are of two types. In the first type a set of spray nozzles is arranged for intermittent cleaning. These sprays are not operative when the system is in use and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 15 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is advantageous to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably con-

stant, and to assure the required design quantity of by-passed air when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 13.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on to the coil connections. (See Chapter 23.)

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The temperature difference.
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes.

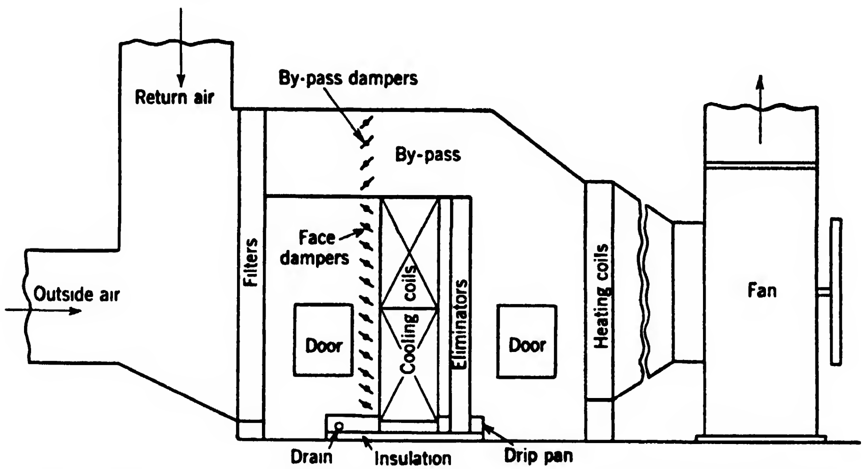


Fig. 13. Typical Arrangement of Cooling Coils in a Central System

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. For combined cooling and dehumidification, the logarithmic difference does not apply strictly and such problems should be handled as described in Chapter 7. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit and superheating in the remainder. In spite of this, heat transfer and ratings for coils using volatile refrigerants are usually based on a refrigerant temperature corresponding to the average pressure in the coil.

The design and surface arrangement of the coil include such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins may be more

effective than flat. This design and surface arrangement has a large effect on the air film heat transfer resistance.

The velocity of the air usually considered is the coil face velocity. This bears a *varied relation to the actual velocity* over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the *actual surface velocity that is significant.* The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil *rating* information may be misleading. *The significant value is mass velocity in pounds per (minute) (square foot of face area) and not cubic feet per minute,*

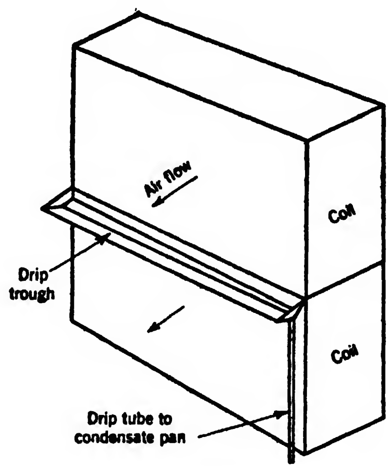


Fig. 14. Coil Arranged with Drip Trough

because for a fixed volume the corresponding weight may vary widely, depending upon the temperature and barometric pressure.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings and in realizing rated performance in practical installations. The resistance through the coils will assist in distributing the air properly, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through the coils. This reduces the capacity, but can be avoided by proper layout or by the use of directing baffles.

Heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. Heat transfer rates expressed as Btu per (square foot of internal surface) (degree logarithmic mean effective temperature difference between the fluid and tube wall) are, for example: about 150 to

300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about 1200 for condensing steam. The influence of the medium in the tubes on the over-all heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions. For details on coil calculation and performance see Chapter 7.

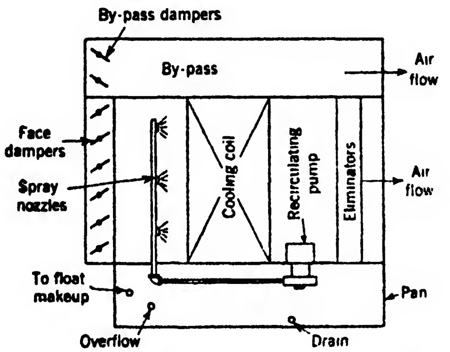


FIG. 15. RECIRCULATING SPRAY SYSTEM FOR CLEANING COILS

COIL SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying.
2. Temperature of entering air—dry-bulb only if there is no dehumidification, dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media.
4. Space and dimensional limitations.
5. Air quantity limitations.
6. Allowable resistances in air circuit and through tubes.
7. Peculiarities of individual designs of coils.
8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapters 6, 8, 14 and 15. There may or may not be a choice of cooling and heating media, as well as temperatures available, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The

air quantity through heating coils is often made the same as that necessary to handle the summer cooling load. The air handled may be fixed by the use of old ventilating ducts as an air distribution system for new air conditioning apparatus, or may be dictated by requirements of satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited by the maximum fan peripheral velocity which requirement of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment and upon certain other factors. The usual coil ratings are based on a uniform face velocity of air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the manufacturer's ratings cannot usually be obtained. To obtain this performance it is necessary also that the air quantity be adjusted on the job to that used in determining the coil selection, and must also be kept at this value. The most common causes for a reduction of air quantity are the fouling of the filters and collection of dirt in the coils. These difficulties can be avoided by proper design and proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash off the accumulation with steam, compressed air and water, or hot water. The most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the necessities of each case and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and *hot water heating coils* are usually *rated* within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.
Steam Pressure—2 to 200 lb, sometimes up to 350 lb per square inch.
Hot Water Temperature—150 to 225 F.
Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual *heating installation practice:*

Air Face Velocity—500 to 800 fpm face, 500 being a common figure.

Delivered Air Temperature—varies from about 72 F for ventilation only to about 150 F for complete heating.

Steam Pressure—2 to 10 lb, 5 lb being common.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Water Quantity—Based on about 20 F temperature drop through a hot-water coil.

Air Resistance—The total resistance through heating coils is usually limited to from $\frac{3}{8}$ to $\frac{5}{8}$ in. of water gage for public buildings, to about 1 in. for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure, it is possible to select several arrangements of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance.

Cooling Coils

Cooling and *dehumidifying coils* are usually *rated* within these limits:

Entering Air Dry-Bulb—60 to 100 F.
Entering Air Wet-Bulb—50 to 80 F.
Air Face Velocities—300 to 800 fpm, (sometimes as low as 200 and as high as 1200).
Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.
Water Temperatures—40 to 65 F.
Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature rise of from 4 to 12 deg.
Water Velocity—2 to 6 fps.

TABLE. 4. VARIOUS COOLING COIL ARRANGEMENTS

Selection	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1.45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0.11	0.27	0.51	0.37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp. F deg	45	45	45	38

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, *i.e.*, sensible heat is from 60 to 100 per cent of total, depending on the application. (See Chapter 43.) Since required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, general rules as to their values may be misleading. On usual comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures ordinarily vary between 40 and 50 F where cooling is accompanied by dehumidification. Water velocities range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew-point of the entering air is equal to or lower than the cooling coil surface temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible heat transfer only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc., for the same duty.

Dehumidifying Coils

The selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling and requires consideration of both dry- and wet-bulb air temperatures. (See Chapter 7.) It is further complicated by the fact that the proportional amount of dehumidi-

fication required is also highly variable. The methods outlined previously under Heat Transfer and Resistance may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving air conditions are arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 4.

It is possible as shown in Table 4 to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil gives low air velocity and resistance but high air quantities per ton. The coil of small face area and great depth requires small air quantities per ton of refrigeration, high resistance and high air velocities. As shown also in Table 4 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible to keep

Table 5. Capacity Balances for Maximum and Minimum Load Conditions

Conditions	Capacity in Tons			Ratio Total/Sensible
	Total	Sensible	Latent	
Required at peak load conditions	10.90	7.90	3.00	1.38
Required at minimum load conditions	6.62	3.36	3.26	1.98
Peak load equipment balance	10.90	7.90	3.00	1.38
Same equipment balanced at minimum load conditions	9.85	6.58	3.26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8.38	5.05	3.33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6.62	3.36	3.26	1.98

the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with correspondingly small compressor operating expense can be attained but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air the evaporating temperature of a volatile refrigerant coil is determined by a balance between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information. If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease is generally in the sensible capacity. This is the effect also when the air by-pass or volume control is used.

It is necessary that not only the total capacity but also the sensible and latent cooling requirements both be met. The installation of an excess of coil results in an increase in total capacity, but not a proportional gain in latent heat capacity. On installations controlled from dry-bulb temperature the operating time is shortened because of the added sensible cooling capacity. This results in less moisture pick-up and higher relative humidity than calculated. If an oversize condensing unit is installed, the opposite situation occurs. Generally, this is not a disadvantage except that it

results in a load from outside air greater than calculated, as well as in increased power consumption. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

Sometimes arbitrary air quantities are specified for ventilation or other reasons independent of the selection of the cooling coil. As shown in Table 4, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil rebalance. This may result in more sensible and less latent capacity than required at the light load condition, with an increased relative humidity in the conditioned space. Such a condition is shown in Table 5. If approximately 40 per cent of the total air is by-passed, the condition is improved as indicated. The situation may be entirely avoided by using reheat, where it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity[18].

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With usual coils for comfort installations, this does not occur unless the evaporating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of the coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature and checking the balance at the minimum load.

Care should be exercised in the design of humidity control to minimize the cycling of the refrigerating compressor because of re-evaporation of moisture from the fins. It is sometimes necessary to by-pass air around a coil when the compressor is not operating.

REFERENCES

[1] A.S.H.V.E. Code for Testing Radiators (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18).

[2] A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367); (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237). (See also A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 38, and Vol. 42, 1936, p. 29).

[3] A.S.H.V.E. RESEARCH REPORT No. 998—Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 443).

[4] Factors Influencing the Heat Output of Radiators, by A. C. Davis, W. M. Sawdon and David Dropkin (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1942, p. 185) and Cornell University, *Engineering Experiment Station Bulletin* No. 29, April, 1942.

[5] Heat Emission from Radiators, by K. F. Rubert (Cornell University, *Engineering Experiment Station Bulletin* No. 24, 1937).

[6] Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 41).

[7] Heat Loss from Direct Radiation, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 11).

[8] Heat Output of Concealed Radiators, by E. A. Allcut (University of Toronto, *School of Engineering, Research Bulletin* No. 140, 1933).

[9] The Heating Effect of Radiators, by Charles Brabbée (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33).

[10] University of Illinois, *Engineering Experiment Station Bulletins* Nos. 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 77).

[11] A.S.H.V.E. RESEARCH REPORT No. 962—The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

[12] The Kata Thermometer—Its Value and Defects, by W. J. McConnell and C. P Yagloglou. (Reprint No. 953 from *U. S. Public Health Service Report*, pp. 2293–2337, September 5, 1924).

[13] The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

[14] The Calibration of the Thermo-Integrator, by C.-E. A. Winslow, A. P. Gagge, Leonard Greenburg, I. M. Moriyama and E. J. Rodee. (*The American Journal of Hygiene*, Vol. 22, No. 1, July, 1935, pp. 137–156.)

[15] The Globe Thermometer in Studies of Heating and Ventilation, by T. Bedford and C. G. Warner. (*The Journal of Hygiene*, Vol. 34, No. 4.)

[16] A.S.H.V.E. RESEARCH REPORT No. 1067—The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 389).

[17] University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p. 20.

[18] Reheating by Means of Refrigerant Compressor Discharge Gas, by S. F. Nicoll (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 239).

CHAPTER 26

UNIT HEATERS, UNIT VENTILATORS, UNIT HUMIDIFIERS

Definitions; Unit Heaters: Classification, Application, Outlet Velocities, Ratings for Various Types, Temperature, Automatic Control, Location, Piping Connections; Boiler Capacity for Steam Unit Heaters; Unit Ventilators: Ratings, Applications, Location, Exhaust Vents; Window Ventilators; Unit Humidifiers

DESCRIPTIONS of heating, cooling, ventilating, humidifying, and dehumidifying systems are given in other chapters. This chapter deals with unit heaters, unit ventilators, and unit humidifiers. Cooling units, unit air conditioners, and attic fans are described in Chapter 36.

Definitions

The generally accepted meaning of the word *unit* in the terms unit heaters, unit ventilators, and unit humidifiers is that of a factory made, encased assembly of the functional elements indicated by its name. Such units can be shipped complete or in sections so that the only field work necessary is the assembling of the sections, providing proper supports and connecting the unit to sources of heat (or fuel), power and water supply, and, if necessary, to vent pipes for combustion gases.

The term *unit heater* denotes an assembly of elements the principal function of which is heating. The essential elements of a unit heater are a fan and motor, a heater, a housing, and outlet vanes or diffusers.

The term *unit ventilator* denotes an assembly the principal function of which is to ventilate. It may serve to circulate air within the space, or to introduce air from without the space, or may accomplish both purposes. The essential elements of a unit ventilator are a fan and motor, a heater, a set of dampers, a housing and outlet vanes or diffusers.

The term *unit humidifier* denotes an assembly of elements the principal function of which is to humidify. The essential element of a unit humidifier is an atomizer or evaporator. To this may be added a fan, a heater, outlet vanes or diffusers, and a housing to enclose the various parts.

UNIT HEATERS

Classification

The various types of unit heaters which are at present available can usually be classified according to one of the three following methods:

1. *By type of heater.* Under this classification there are three types of heating elements to be considered: (*a*) *the steam or hot water type*, (*b*) *the electric type*, and (*c*) *the direct fired type* which may be *gas*, *oil*, or *coal fired*.

2. *By type of fan.* Under this classification there are two types of fans to be considered: (*a*) the *propeller type*[1] which may be equipped with a horizontal or vertical shaft, and (*b*) the *centrifugal type* which may be designed for *horizontal* or *vertical blow*.

3. *By arrangement of elements.* Under this classification there are two types of heaters to be considered: (*a*) the *draw-through type*, in which the

fan draws air through, and (*b*) the *blow-through type*, in which the fan blows air through the heater.

Unit heaters are available in any combination of the three preceding general classifications. For example, the steam or hot water type may be secured with either the propeller or centrifugal type of fans, and in either the draw-through or blow-through type.

Unit heaters also vary in other minor respects. For example, steam and return inlets and outlets may be located on the top and bottom respectively or on the same side of the unit. Some units are supported by the piping and some have independent supports. The heating surface of steam or water type units is generally made up of a non-ferrous tube-and-fin assembly; or it may be fabricated of steel, or cast in steel.

Application of Unit Heaters

Unit heaters are used principally for heating commercial and industrial structures such as garages, factories, laboratories, and stores. They may also be used for heating finished rooms, if properly applied and concealed and if some consideration is given to the problem of noise.

Unit heaters may also be adapted to a number of industrial processes, such as drying and curing, in which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is released. When such conditions are severe, it is necessary that the unit heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. See discussion of condensation in Chapter 6.

There are three major factors to consider in the application of unit heaters, namely: (1) location of unit, (2) air distribution, and (3) heating medium.

There are a variety of applications which are favorable to the use of *electric unit heaters*. For supplemental heat in residence bath rooms, for the heating of ticket booths, watchmen's offices, factory offices, locker rooms and other isolated rooms scattered over large areas, their use is peculiarly adaptable. They are particularly useful in isolated and untended pumping stations or pits where they may be thermostatically controlled to prevent freezing temperatures.

Gas fired unit heaters find application in industrial plants, offices, stores, garages, in fact in almost every location where steam type units are used. The installation cost of gas fired units is usually less than that of a type requiring that a new boiler be installed.

Oil fired unit heaters are used in industrial plants, garages and commercial buildings.

Coal fired unit heaters are of finned, welded steel, or cast-iron construction and equipped with centrifugal blowers. They are usually stoker-fired to insure proper firing of fuel. They are used principally in large industrial plants such as foundries or assembly plants, and provide a convenient source of heat, readily installed and economical in operation since all heat given off by the surface of the heater remains within the space being heated.

Outlet Velocities

Outlet velocities of unit heaters vary from about 400 to 2500 fpm depending upon the type of unit and the distance to which the air is to be projected.

Noise and drafts must be considered in the choice of air velocities, since both increase with increase of air velocity.

Velocities and decibel ratings for the various types of unit heaters illustrated in Figs. 1, 2, 3 and 4 are given in Table 1. (See Chapter 42 for a discussion of decibel ratings.)

In the selection of unit heaters it is important to ascertain that the throw is sufficient. The throw is dependent to a marked degree on the temperature of air leaving the heater as well as upon its velocity. See discussion under heading of Inlet, Outlet, and Space Temperatures with Unit Heaters.

Air Outlets

In order to direct the air to points desired and to diffuse the air to avoid drafts, unit heaters are commonly equipped with directional outlets, adjustable louvers or fixed types of diffusers.

TABLE 1. OUTLET VELOCITIES, DISTANCE OF BLOW AND DECIBEL RATINGS FOR VARIOUS TYPES OF UNIT HEATERS

Type of Unit Heater	Outlet Velocities FPM	Distances of Blow—Ft[a]	Decibel Rating
Centrifugal Fan	1500–2500	20–200	34–90
Horizontal Propeller Fan	400–1000	30–100	26–84
Vertical Propeller Fan	1200–2200	70	50–82

[a] Refer to manufacturers' tables for limits of blow and decibel ratings. Values in this table are maxima and do not apply to all makes of units.

RATINGS OF UNIT HEATERS

It is standard practice to rate unit heaters on the basis of the amount of heat delivered by the air in Btu per hour above an entering air temperature of 60 F. This applies to all types of unit heaters, the steam or hot water type, the electric type and the direct fired type. There are, however, other factors which must be taken into account, especially when an attempt is made to compare one type of heater with another. These are the temperature of the heating element and the velocity of air through it. Consideration is given to these factors in the discussion of ratings for each type of unit heater in the following paragraphs.

Ratings for Steam Type

The rating of steam type unit heaters has been standardized by a code[2] in which the following items are the basis of rating: dry saturated steam at 2 psig pressure (29.92 in. Hg barometric pressure) at the heater coil; air at 60 F entering the heater; and heater operating free of external resistance to air flow.

The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heating capacity for any condition of steam pressure and entering air temperature other than standard may be calculated approximately from any given rating by the use of factors in Table 2 for the blow-through or draw-through types.

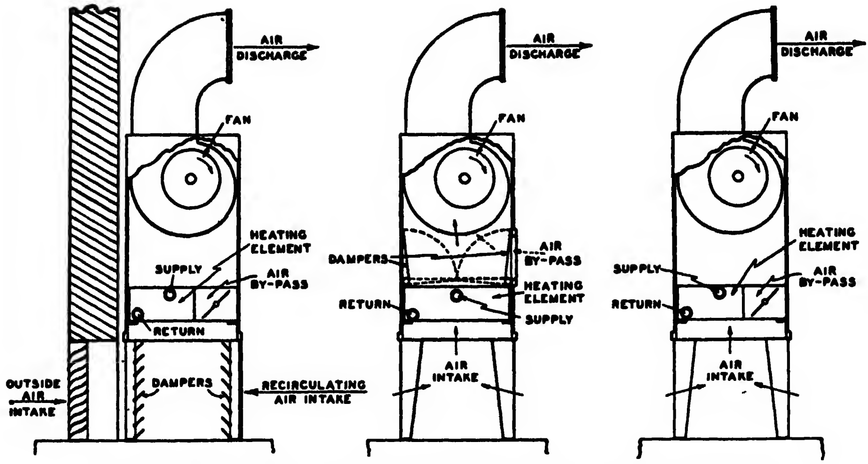


Fig 1. Centrifugal Fan Type Unit Heater—Floor Mounted

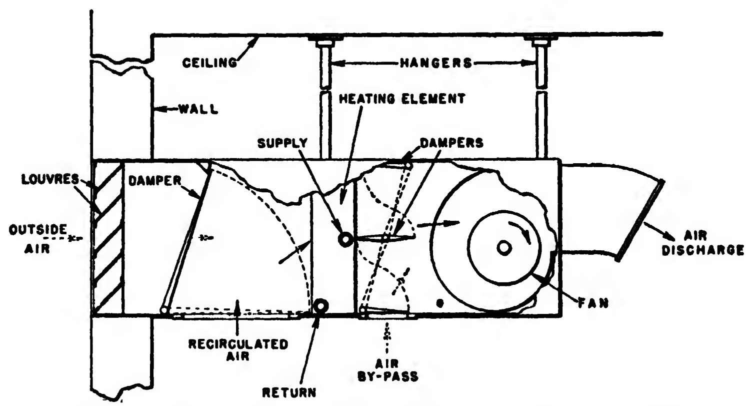


Fig. 2. Centrifugal Fan Type Unit Heater—Suspended

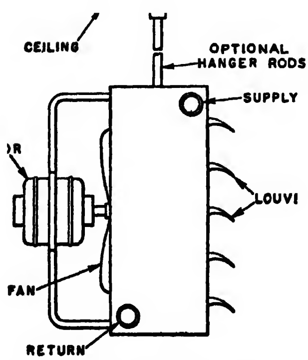


Fig. 3. Propeller Fan Type Unit Heater—Horizontal Blow

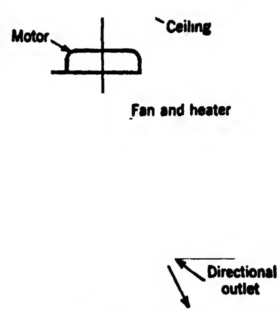


Fig. 4. Propeller Fan Type Unit Heater—Vertical Blow

Table 2. Constants for Determining the Capacity of Unit Heaters for Various Steam Pressures and Temperatures of Entering Air

(Based on Steam Pressure of 2 psig and Entering Air Temperature of 60 F)

	Steam Pressure psi	Temperature of Entering Air											
		−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
Blow Through Type	0	1.54	1.45	1.37	1.27	1.19	1.11	1.03	0.96	0.88	0.81	0.74	0.67
	2	1.59	1.50	1.41	1.32	1.24	1.16	1.08	1.00	0.93	0.85	0.78	0.71
	5	1.64	1.55	1.46	1.37	1.29	1.21	1.13	1.05	0.97	0.90	0.83	0.76
	10	1.73	1.64	1.55	1.46	1.38	1.29	1.21	1.13	1.06	0.98	0.91	0.84
	15	1.80	1.71	1.61	1.53	1.44	1.34	1.28	1.19	1.12	1.04	0.97	0.90
	20	1.86	1.77	1.68	1.58	1.50	1.42	1.33	1.25	1.17	1.10	1.02	0.95
	30	1.97	1.87	1.78	1.68	1.60	1.51	1.43	1.35	1.27	1.19	1.12	1.04
	40	2.06	1.96	1.86	1.77	1.68	1.60	1.51	1.43	1.35	1.27	1.19	1.12
	50	2.13	2.04	1.94	1.85	1.76	1.67	1.58	1.50	1.42	1.34	1.26	1.19
	60	2.20	2.09	2.00	1.90	1.81	1.73	1.64	1.56	1.47	1.39	1.31	1.24
	70	2.26	2.16	2.06	1.96	1.87	1.78	1.70	1.61	1.53	1.45	1.37	1.29
	75	2.28	2.18	2.09	1.99	1.90	1.81	1.72	1.64	1.55	1.47	1.40	1.32
	80	2.31	2.21	2.11	2.02	1.93	1.84	1.75	1.66	1.58	1.50	1.42	1.34
	90	2.36	2.26	2.16	2.06	1.97	1.88	1.79	1.71	1.62	1.54	1.46	1.38
	100	2.41	2.31	2.20	2.11	2.02	1.93	1.84	1.75	1.66	1.58	1.50	1.42
Draw Through Type	0	1.48	1.41	1.33	1.25	1.18	1.11	1.03	0.96	0.89	0.82	0.75	0.69
	2	1.52	1.44	1.36	1.29	1.22	1.14	1.07	1.00	0.93	0.86	0.79	0.73
	5	1.57	1.49	1.41	1.33	1.26	1.19	1.11	1.05	0.98	0.91	0.84	0.77
	10	1.64	1.56	1.48	1.40	1.33	1.25	1.18	1.11	1.04	0.97	0.90	0.84
	15	1.69	1.61	1.53	1.46	1.38	1.31	1.24	1.17	1.10	1.03	0.96	0.90
	20	1.73	1.65	1.57	1.50	1.42	1.35	1.28	1.21	1.14	1.07	1.00	0.94
	30	1.80	1.73	1.65	1.57	1.50	1.42	1.35	1.28	1.21	1.15	1.08	1.01
	40	1.86	1.79	1.71	1.64	1.56	1.49	1.42	1.35	1.28	1.22	1.15	1.08
	50	1.93	1.85	1.77	1.70	1.63	1.55	1.48	1.42	1.35	1.28	1.21	1.15
	60	1.97	1.90	1.82	1.75	1.67	1.60	1.53	1.46	1.40	1.33	1.26	1.19
	70	2.02	1.94	1.87	1.80	1.72	1.65	1.58	1.51	1.44	1.38	1.31	1.24
	75	2.04	1.97	1.90	1.82	1.75	1.68	1.61	1.54	1.47	1.40	1.33	1.27
	80	2.06	1.99	1.91	1.84	1.77	1.70	1.63	1.56	1.49	1.42	1.35	1.29
	90	2.10	2.03	1.95	1.88	1.80	1.73	1.66	1.59	1.52	1.46	1.39	1.32
	100	2.15	2.07	1.99	1.92	1.85	1.77	1.70	1.63	1.56	1.49	1.43	1.36

Note: **To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering air and 2 psi.**

When increasing steam pressure it is important to determine whether the heater is suitable for the increased pressure application and whether the resulting increased outlet temperature is satisfactory.

Ratings for Hot Water Type

A standard for the rating of hot water type unit heaters has also been established by code[8] in which the following items are the basis of rating: entering water at 200 F; entering air at 60 F (29.92 in. Hg barometric pressure); and heater operating free of external resistance to air flow. This code also prescribes a method of translating the output in Btu and the temperature rise as obtained under test conditions to standard conditions of air and water temperature.

Ratings for Electric Type

Electric type unit heaters are available in sizes up to at least 60 kw capacity. They consist of resistance type heating elements combined with fan and motor, together with a suitable casing. Electric unit heaters are made in the built-in-wall model, suspension model, and free-standing or portable model.

Electric unit heaters are rated on the energy input to the heater, expressed in terms of kilowatts, Btu or EDR. Quite often all three ratings are given in parallel columns in the catalogs.

Ratings for Gas Fired Type

Gas fired unit heaters are built in both suspended and floor models with either propeller or centrifugal type fans. They are available in a wide range of sizes from about 24,000 to over 4,000,000 Btu per hr capacity and are usually rated in terms of both input and output according to the approval requirements of the *American Standards Association*. Any gas fired unit which is thermostatically controlled and has a pilot must have an element in the pilot flame which will automatically close the gas valve on pilot failure.

Ratings for Oil Fired Type

The oil fired type of unit heater is usually equipped with the centrifugal type of fan only and can be obtained in sizes ranging from 125,000 to 1,650,000 Btu per hour output capacity in standard units. It is furnished in either the floor mounted or in smaller sizes in the suspended type.

Ratings for Coal Fired Type

The stoker fired type of unit heater can be obtained in ranges of from 300,000 to 6,000,000 (or more) Btu per hour output capacity. Ratings are based upon the delivered output at the heater outlet.

Effect of Resistance Upon Capacity

Unit heaters are customarily rated as free delivery type units. If outside air intakes, air filters, or ducts on the discharge side are used with the unit, a reduction in air and heating capacity will result because of this added resistance. The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design and speed of the fans, so that no specific percentage reduction can be assigned for all heaters at a given added resistance. In general, however, propeller fan type units will experience a larger reduction in capacity than housed centrifugal fan units for a given added resistance and a given heater will have a larger reduction in capacity as the fan speed is lowered. The heat output to be expected under other than free delivery conditions should be secured from the manufacturer.

Inlet, Outlet and Space Temperatures with Unit Heaters

In the selection of unit heaters for any particular design, consideration should be given to the temperature of air entering the heaters as well as the temperature to be maintained in the working zone of the space. In general, the temperature differences per foot of elevation, when using unit heaters, are less than corresponding variations when using direct radiation[4]. High velocity units will maintain slightly lower temperature differences than low discharge velocity units. Correspondingly, units with lower discharge air temperature will maintain lower temperature differences than units with higher discharge temperatures. Directional control of the discharged air from a unit heater can be an important factor, added to qualities of reasonably good outlet velocity and outlet temperature, in effecting satisfactory distribution of heat and reducing floor-to-ceiling temperature difference.

Since the outlet temperature of air from a unit heater increases with the temperature of the heating medium, such as high pressure steam, heaters can be obtained with heating elements having less than the regular amount of heating surface in order to obtain, with the high temperature heating medium, approximately the same leaving air temperature as would be obtained from a lower temperature heating medium.

When some *outside air* is introduced, the temperature of the mixture of outside and recirculating air must be calculated and used as the entering air temperature at the heater. Unit heaters connected in this manner perform the function of unit ventilators. For a discussion of this function see the section of this chapter entitled *Unit Ventilators*.

For recirculating heaters located *at the floor* or with *intakes at the floor*, the temperature of air entering the heater should be assumed to be the same as that to be maintained in the room itself.

Automatic Control of Unit Heaters

Thermostatic control of unit heaters may be accomplished either by starting and stopping the fan or by controlling the flow of the heating medium to the heating element. If the fan is controlled, it is advisable to provide a temperature-operated switch to prevent the fan from starting until the heating element is heated throughout.

Unit heaters may be used in summer as a means of circulating air to give some measure of comfort due to air motion. In such cases the heating element should be shut off from the source of heat. The thermostat which prevents the fan from starting until the heating element is heated, should be provided with a by-pass switch, which, upon being closed, will permit the fan to be operated independently of the heating element.

Location of Unit Heaters

Care should be taken in the location of unit heaters to insure free air circulation to the intake. The best arrangement is to locate units so that they discharge air nearly parallel to exterior walls, and in a direction which will produce a rotational circulation around the room. This is preferable to directing the discharge against the outside walls.

Various types and makes of unit heaters are illustrated in the *Catalog Data Section* of this edition. As hot blasts of air in working zones are usually objectionable, heaters mounted on the floor should have their discharge outlets above the head line, and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will

not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air is brought as close above the head line as possible, yet not into the working zone.

In connection with the use of vertical type unit heaters, care must be exercised in the selection of the heater. It has been found that the higher the unit is placed above the floor, the lower must be the outlet temperature of the air leaving the heater in order that the heated air may be forced into the occupied zone.

Determining Unit Heater Requirements

The formulas given in the section on Unit Ventilators may be used to determine unit heater capacity requirements.

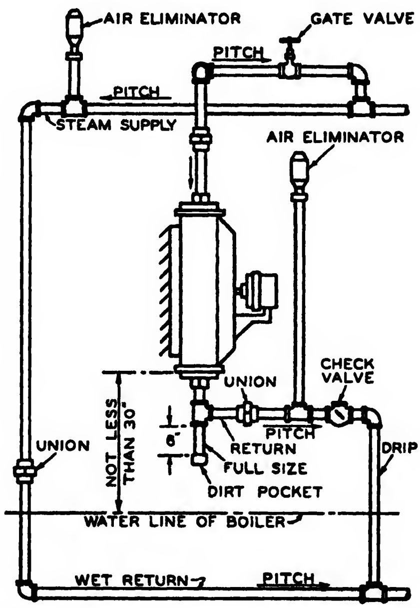


Fig. 5. Unit Heater Connection to One-Pipe Gravity Steam System

PIPING CONNECTIONS FOR STEAM UNIT HEATERS

Piping connections for steam unit heaters are similar to those for other types of fan blast heaters. The piping of unit heaters must conform strictly to the system requirements while at the same time permitting the heaters themselves to function as intended. The basic piping principles for steam systems are discussed in Chapter 23.

Rapid condensation of steam, especially during heating-up periods, is characteristic of this type of equipment. The return piping must be planned to keep the heating coil free of rapid condensation, while the steam piping must be ample to carry a full supply of steam to the unit to take the place of that condensed. Adequate sizes of piping are especially important where a unit heater fan is operated under start-and-stop control and where all or part of the air is taken from the outside. In such installations the

condensation rate may vary rapidly and the necessity for ample pipe capacity is particularly important.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 5. In cases where the return main is located above the boiler water line, an artificial water line must be created by providing an equalizing loop to prevent steam passing into the return and thus into other units.

A piping arrangement where both the air and condensate pass through a common return to a boiler, with vent trap or condensate pump and receiver, is shown in Fig. 6. The traps must pass air and condensate rapidly to keep the return piping partially full of water.

Since unit heaters are often constructed with sufficient strength, the use of high pressure steam in them is a common practice. As shown in Fig. 7 the condensate and air reach the return overhead through traps, and check

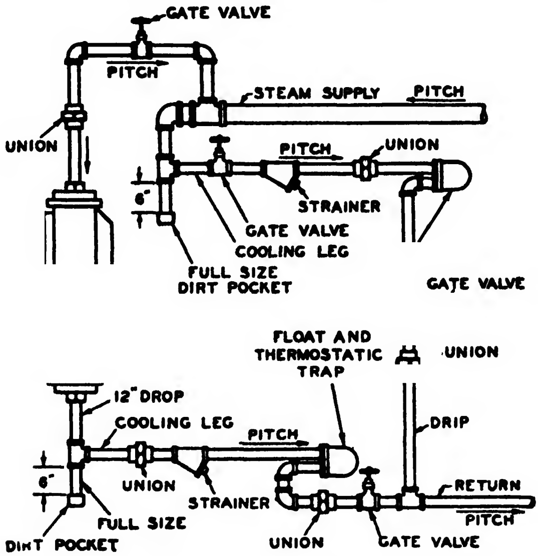


Fig. 6. Unit Heater Connection for Vacuum or Vapor System Discharging Condensation into Dry Return

valves are located in the return piping. It is, however, preferable to locate the high pressure return below the heater.

For two-pipe closed gravity return systems, the return from each unit should be fitted with a heavy duty or blast trap, and an automatic air valve should be connected into the return header of each unit heater. Provisions must be made to compensate for the pressure drop by elevating the unit heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns as in vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vent systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty

trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, and then eliminated at some other point in the system.

Fig. 8 represents the connections to a hot water heating system. The air vent is not required if the main is above the heater and air can be eliminated through the piping system.

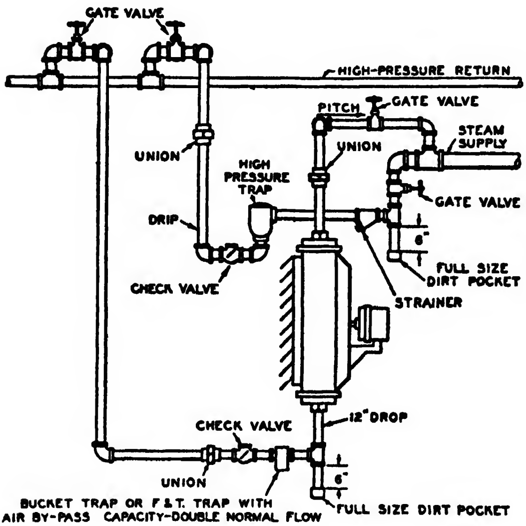


Fig. 7. Method of Connecting Unit Heater to High Pressure System

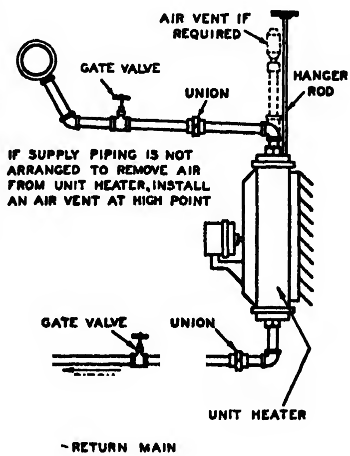


Fig. 8. Method of Connecting Unit Heater to Hot Water System

BOILER CAPACITY FOR STEAM UNIT HEATERS

The capacity of the boiler should be based on the rated capacity of the unit heaters at the lowest entering air temperature and highest fan speed that will occur, plus an allowance for pipe line losses. It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attention to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller unit heaters instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their heating surfaces are designed for those pressures, and when proper provision is made for returning the condensate. If units admit air that may be at a temperature below freezing, a steam pressure of

not less than 5 lb should be maintained on the heating element, or a corresponding differential in pressure between the supply and returns should be maintained by means of a vacuum.

UNIT VENTILATORS

A unit ventilator is essentially a unit heater equipped with dampers for introducing outdoor and recirculating air in varying quantities, and arranged with a system of control that permits the heating effect to be varied with the fans operating continuously. Either steam or forced hot water may be employed as the heating medium, although steam is the more common. Unit ventilators are intended primarily for use in schools, meeting rooms, offices or other applications where the density of occupancy indicates the need for ventilation. In normal operation, the discharge air temperature from a unit ventilator is varied in accordance with the room

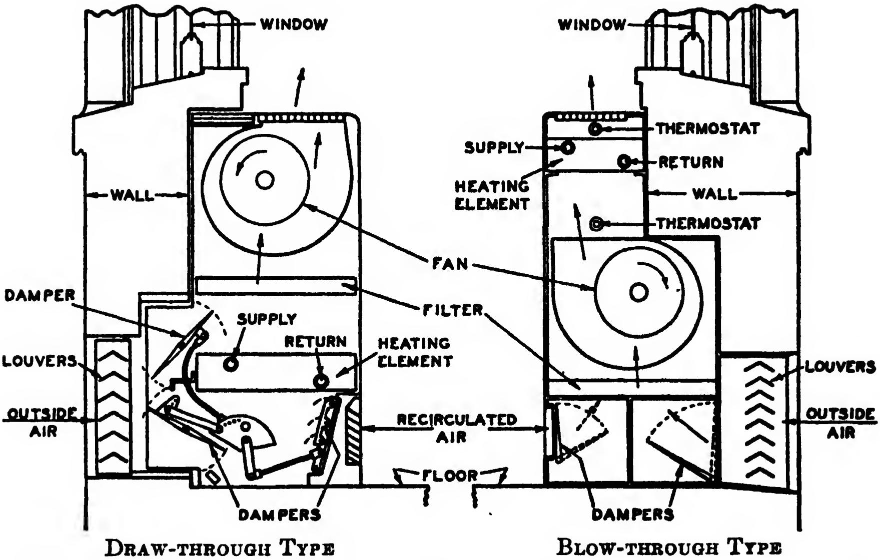


FIG. 9. TYPICAL UNIT VENTILATORS SHOWING TWO OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

demands. Where a heating effect is required, the air delivered is above room temperature. Where the heat generated within the room by occupants, sun, etc., is sufficient to cause overheating, then the air delivery temperature must be below that of the rooms. It is customary to equip unit ventilators with control devices that prevent the delivery of air at a temperature that will cause cold drafts. Unit ventilators may be of the radiator or damper controlled type constructed on the blow-through or draw-through principle as illustrated in Fig. 9.

Ratings of Unit Ventilators

Unit ventilators of the steam type are customarily cataloged with two ratings, one the input and the other the output capacity. The first is the heat input to the unit which is determined by measuring the temperature and quantity of condensate and the pressure and quality of the steam. The second is the heat output of the unit which is determined by measuring

Table 3. Typical Capacities of Unit Ventilators for an Entering Air Temperature of Zero

Cubic Feet of Air per Minute		Total Capacity in Square Feet, Equivalent Direct Radiation	Capacity Available for Heating the Room, Square Feet Equivalent Direct Radiation	Final Air Temperature F Deg
Anemometer Rating	Condensate Rating			
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

the quantity of air delivered and the temperature of the air to and from the unit. Table 3 shows the air handling capacities by the two methods of rating[5] and also the approximate heating data. In accordance with the A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators the information to be supplied regarding ratings and the basis of rating is the following:

Rating Factors to Be Specified. The rating of the unit ventilator shall specify the following:

a. Final temperature at different entering air temperatures.

b. Total EDR at different entering air temperatures.

c. Air delivered by the unit in cubic feet per minute at the *standard basis of rating* with the fans operated at rated speed, with all air being blown through the heating unit and with the standard louver and grille on the outlet.

The Standard Basis of Rating shall be as follows:

a. Dry saturated steam at a temperature at the unit corresponding to an absolute pressure of 16.7 psi (218.5 F).

b. Entering air temperature of zero Fahrenheit degrees.

c. Volume delivered in cubic feet per minute converted to standard air at 70 F.

Rating Tables for unit ventilators shall contain the following data in addition to the standard rating, for entering air temperatures from −30 F to +60 F:

a. Inlet temperature, Fahrenheit degrees.

b. Final temperature, Fahrenheit degrees.

c. Total EDR at the specified entering temperature.

d. Surplus or heating EDR at the specified entering temperature.

Surplus or Heating Equivalent Direct Radiation for the purposes of this code shall be construed to mean difference between the total EDR at a specified inlet temperature and the EDR required to heat the air from that temperature to 70 F.

If no direct heating surface (radiation) is installed to take care of the normal heat transfer losses, and the unit ventilator is to be used for both heating and ventilation, then the combined requirements must be taken care of by the unit ventilator.

Heat Required for Ventilating Only

When all of the air handled by the unit is taken from the outside, the total heat to be supplied is obtained by means of Equations 1, 2, 3, and 4.

$$H = 0.24\ W\ (t_y - t) \qquad (1)$$

$$H_v = 0.24\ W\ (t - t_o) \qquad (2)$$

$$H_t = 0.24\ W\ (t_y - t_o) = H + H_v \qquad (3)$$

$$W = d\ 60\ Q \qquad (4)$$

From Equations 2, 3, and 4:

$$(t - \qquad (5)$$

where

d = density of air, pounds per cubic foot (0.075 lb per cu ft for Standard Air by definition).
H = heat loss of room, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour.
Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room, Fahrenheit degrees.
t_o = outside temperature, Fahrenheit degrees.
t_y = temperature of the air leaving the unit, Fahrenheit degrees.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure (approximate value).

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. Substituting in Equation 5:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70 - 0) = 99{,}600 \text{ Btu per hour}$$

$$\frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} \quad 70 = 92.2$$

Heat Required for Ventilating and Recirculating

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$(t_y - t_o) + 0.24\ W_i\ (t_y - t) \qquad (7)$$

$$W_o = d_o\, 60\, Q_o \qquad (8)$$

$$W_i = d_i\, 60\, Q_i \qquad (9)$$

$$H_t = H + 0.24\ d_o\, 60\, Q_o\ (t - t_o) \qquad (11)$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.
d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cubic feet per minute.
Q_i = volume of air taken in from the room, cubic feet per minute.

Applications of Unit Ventilators

Items to be considered in the application of unit ventilators are: (1) combination with other means of heating, (2) location of units, and (3) method of venting or exhausting.

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at or slightly above room temperature, and enough radiation is installed in the room to take care of the normal heat transfer losses. Where the unit ventilator selected has a capacity more than suffi-

cient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct radiation installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

The *combined* system employs a unit ventilator with sufficient capacity for both ventilation and the normal heat transfer losses. In such a case no direct radiation is required. The necessary operation of the fan when the room is being heated also gives assurance that some ventilation is being provided, especially if automatic dampers are used in the air intake and in the recirculating intake. These dampers are arranged to provide a certain quantity of outside air, depending upon the weather conditions. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be available only the heating effect of the units acting as convectors.

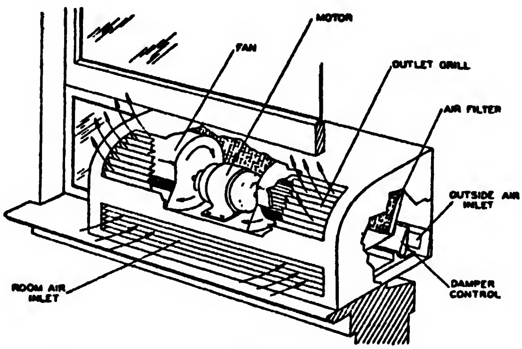


FIG. 10. TYPICAL WINDOW VENTILATOR

Location of Unit Ventilator

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall and on the center line of the room. It is difficult to obtain proper air distribution if the unit is installed either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom.

Air Exhaust Vents and Flues

The size and location of the air exhaust vent[6] outlet is important. In many cases the sizes for public buildings are regulated by law.

In cases where no codes govern, the location and size of vents are left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the

vent openings of from 600 to 800 fpm produce the best diffusion results from this system. Many states, however, have regulations that will not permit velocities as high as 800 fpm. If a vent opening at or near the floor is near a desk or place where a person is seated, a velocity of 800 fpm in the vent opening will produce an objectionable draft. In such a case the velocity in the vent opening should not exceed 400 to 450 fpm, although duct velocities are maintained at 600 to 800 fpm if codes permit.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

WINDOW VENTILATORS

A window ventilator illustrated in Fig. 10 consists of filter and switch controlled motor driven fans enclosed in a cabinet to be mounted on the window sill. Such units accomplish ventilation, air cleaning, and air cir-

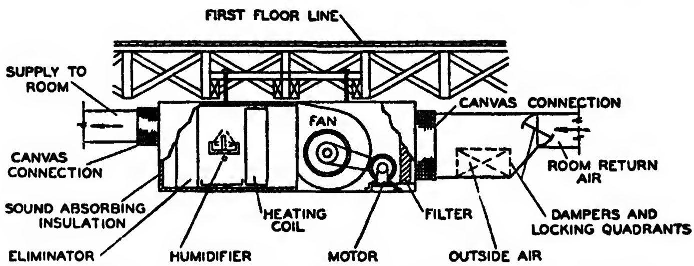


Fig. 11. Typical Unit Humidifier of the Spray Type with Steam Coil to Preheat the Air for Residences

culation, but have no means of heating the air. The direction of air discharge is manually adjustable for seasonal operation.

UNIT HUMIDIFIERS

Unit humidifiers fall into four general classifications, depending on the method of causing evaporation. These are as follows: (1) *Nozzle Type*, (2) *Rotary Type*, (3) *Cascade Type*, and (4) *Heater Type*.

In the *nozzle type* of humidifier water is sprayed into the air and evaporation is effected by adiabatic exchange of energy. Units of this type in simplest form spray a fine mist of water directly into the air in a space. They are used to a great extent in the textile industry.

To the simple type of nozzle unit may be added the following accessories: a water heater whose function is to increase the vapor pressure, an air heater to heat the air either before or after humidification takes place, a screen or air filter over which the water is sprayed and through which the air is drawn for intimate contact with the water, a fan to create an air stream and to deliver air to the space to be humidified, and a housing to enclose all the elements. This type of unit has been made in practically all of the variations mentioned. It has the disadvantage of clogging the nozzles and must be serviced continually.

Fig. 11 illustrates a nozzle unit having a humidifying capacity sufficient for a residence or small building. These units usually include air filters and in some cases provide ventilation air by means of an outside air duct

connection to the unit. The units are constructed for either floor or ceiling mounting and are usually placed in a central location in the basement with short supply and return duct connections from the first floor. Room air is brought into the unit through the return duct connection, is passed over a tempering coil heated by steam or hot water, and is then humidified by passing through some type of spray humidifier. Surplus moisture is removed by an eliminator and the humidified air is delivered to the room through a duct connection. Since a large percentage of the tempering coil capacity is transformed into latent heat during the humidifying process, the unit does not generally eliminate any existing steam radiation but does tend to improve comfort conditions by supplying heat during the off-period of furnace operation[7].

In the *rotary type* of humidifier the spray is created by rotating vanes or discs which throw the water by centrifugal force and in so doing break it up into a fine mist. In all other respects, this type of humidifier is similar to the nozzle type. It has the advantage over the nozzle type of being less liable to become clogged.

In the *cascade type* the humidification takes place by water falling in sheets over a series of baffles or trays. This type is usually furnished with a fan, air heater, and air filter all enclosed in a housing.

In the *heater type* of humidifier the water is heated either to the boiling point or to a temperature at which the water vapor readily passes into the air stream. There are many variations of this type of humidifier. In the simplest form the heating element using steam, hot water, gas, oil, or electricity is placed in a pan or vessel of water and the vapor passes from the surface of the water to the stream of air.

A modification of this type of humidifier is the combination of spray nozzle and heater type in which the water is sprayed over a hot surface and evaporated. It has the disadvantage of accumulating scale on the surfaces of the vessel or heating surface.

For a discussion of various methods of humidification see Chapter 37.

REFERENCES

[1] See *National Fan Manufacturers Association* standard definitions in Chapter 32.

[2] Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165), prepared by a Joint Code Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Industrial Unit Heater Association* and adopted 1930.

[3] Standard Code for Testing Hot Water Unit Heaters prepared by Engineering Committee of *Industrial Unit Heater Association*. Adopted by *Industrial Unit Heater Association* August 1942 and published September 1942.

[4] A.S.H.V.E. RESEARCH REPORT No. 958—Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243). A.S.H.V.E. RESEARCH REPORT No. 1011—Tests of Three Heating Systems in an Industrial Type of Building by G. L. Larson, D. W. Nelson and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

[5] A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

[6] A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463). A.S.H.V.E. RESEARCH REPORT No. 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 279).

[7] Estimating the Humidification Requirements of Residences, by W. H. Severns (Papers Presented at the First Annual Conference on Air Conditioning, University of Illinois, *Engineering Experiment Station Circular, No. 26*, October, 1936).

CHAPTER 27

PIPE, FITTINGS, WELDING

Pipe Materials, Types of Pipe, Commercial Pipe Dimensions, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Flange Facings and Gaskets, Welding in Erection of Piping, Valves

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IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper, and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installations. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe*.

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld, resistance-weld, or butt-weld process. While the lap-weld and resistance-weld processes produce a better weld than the butt type, lap-weld and resistance-weld pipe are seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operations. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is claimed to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher first cost is said to be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast-Ferrous Pipe. There are now available several types of cast-ferrous

metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft, with external and internal diameters closely approximating those of extra strong wrought pipe. Cast-ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small amount of copper and molyb-

TABLE 1. DIMENSIONS OF SCHEDULES 30 AND 40 AND STANDARD WEIGHT PIPE[a]

Size	Diameter In.		Thickness[b], In.	Weight per Ft, Lb		Threads per In.	Circumference, In.		Transverse Area, Sq In.			Length of Pipe, Ft per Sq Ft		Length of Pipe, Ft Containing 1 Cu Ft	Weight of Water, Lb per Ft
	External	Internal		Plain Ends	Threads and Couplings		External	Internal	External	Internal	Metal	External Surface	Internal Surface		
1/8	0.405	0.269	**0.068**	0.244	0.245	27	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2533.775	0.025
1/4	0.540	0 364	**0.088**	0.424	0.425	18	1.696	1.144	0.229	0.104	0.125	7.073	10.493	1383.789	0.045
3/8	0.675	0.493	**0.091**	0.567	0.568	18	2.121	1.549	0.358	0.191	0 167	5.658	7.748	754.360	0.083
1/2	0.840	0.622	**0.109**	0.850	0.852	14	2.639	1.054	0.554	0.304	0.250	4.547	6.141	473.906	0.132
3/4	1 050	0.824	**0.113**	1.130	1.134	14	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034	0.231
1	1.315	1.049	**0.133**	1.678	1.684	11½	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.618	0.375
1¼	1.660	1.380	**0.140**	2.272	2.281	11½	5.215	4.335	2.164	1.495	0.669	2.301	2.768	96.275	0.65
1½	1.900	1.610	**0.145**	2.717	2.731	11½	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	0.88
2	2.375	2.067	**0.154**	3.652	3.678	11½	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	1.45
2½	2.875	2.469	**0.203**	5.793	5.819	8	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	2.07
3	3.500	3.068	**0.216**	7.575	7.616	8	10.996	9 638	9.621	7.393	2.228	1.091	1.245	19.479	3.20
3½	4.000	3.548	**0.226**	9.109	9.202	8	12.566	11 146	12.566	9.886	2.680	0.954	1.076	14.565	4.29
4	4.500	4.026	**0.237**	10.790	10.889	8	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312	5.50
5	5.563	5.047	**0.258**	14.617	14.810	8	17.477	15.856	24 306	20.006	4.300	0.686	0.756	7.198	8.67
6	6.625	6.065	**0.280**	18.974	19.185	8	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984	12.51
8c	8.625	8.071	0 277	24.696	25.000	8	27 096	25.356	58.426	51.161	7.265	0.443	0.473	2 815	22.18
8	8.625	7.981	**0.322**	28.554	28.809	8	27.096	25.073	58.426	50.027	8.399	0.443	0.478	2.878	21.70
10c	10.750	10.136	0.307	34.240	35.000	8	33.772	31.843	90.763	80.691	10.072	0.355	0.376	1.785	34.95
10	10.750	10.020	**0.365**	40.483	41.132	8	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826	34.20
12c	12.750	12.090	0.330	43.773	45.000	8	40.055	37.982	127.676	114.800	12 876	0.299	0.315	1.254	49.70
12	12.750	12.000	0.375	49.562	50.706	8	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273	49.00

[a] Standard-weight wrought-iron pipe has approximately the same wall thicknesses and weights as contained herein for steel pipe. For exact dimensions, see *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10.

[b] Thicknesses shown in bold face type are identical with thicknesses for Schedule 40 pipe of *A.S.A.* B36.10.

[c] Same as Schedule 30, *A.S.A.* B36.10.

denum have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin wall pipe, but the corrosive nature of the fluid

conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The two weights of steel and wrought-iron pipe commonly used are known as *standard weight* and *extra strong*, which correspond to Schedules 40 and 80, respectively, of the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10. The same external diameter is used for both

TABLE 2. STANDARD WEIGHTS AND DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE[a]

Size	Outside Diameter, In.	No. of Threads per In.	Standard-Weight Pipe				Extra-Strong Pipe				Double Extra-Strong Pipe[b]	
			Schedule 30		Schedule 40		Schedule 60		Schedule 80			
			Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness In.	Weight per Ft, Lb Plain Ends
1/8	0 405	27	....	....	0.068	0.25	....	....	0.095	0.31	....	....
1/4	0.540	18	....	....	0.088	0.43	....	....	0.119	0.54	....	....
3/8	0.675	18	....	....	0.091	0 57	....	....	0.126	0.74	....	....
1/2	0.840	14	....	....	0.109	0.85	....	....	0.147	1.09	0.294	1.71
3/4	1.050	14	....	....	0.113	1.13	....	....	0.154	1.47	0.308	2.44
1	1.315	11 1/2	....	....	0.133	1.68	....	....	0.179	2.17	0.358	3.66
1 1/4	1.660	11 1/2	....	....	0.140	2.28	....	....	0.191	3.00	0.382	5.21
1 1/2	1.900	11 1/2	....	....	0.145	2.73	....	....	0.200	3.63	0.400	6.41
2	2.375	11 1/2	....	....	0.154	3.68	....	....	0.218	5.02	0.436	9.03
2 1/2	2.875	8	....	....	0.203	5.82	....	....	0.276	7.66	0.552	13.70
3	3.500	8	....	....	0.216	7.62	....	....	0.300	10.25	0.600	18.58
3 1/2	4.000	8	....	....	0.226	9.20	....	....	0.318	12.51	0.636	22.85
4	4.500	8	....	....	0.237	10.89	....	....	0.337	14.98	0.674	27.54
5	5.563	8	....	....	0.258	14.81	....	....	0.375	20.78	0.750	38.55
6	6.625	8	....	....	0.280	19.19	....	....	0.432	28.57	0.864	53.16
8	8.625	8	0.277	25.00	0.322	28.81	....	....	0.500	43.39	0.875	72.42
10[c]	10.750	8	0.307	35.00	0.365	41.13	0.500	54.74	....	....	....	....
12[d]	12.750	8	0.330	45.00	0.375	50.71	0.500[d]	65.41	....	....	....	....

From Standard Specifications for Welded and Seamless Steel Pipe of the *American Society for Testing Materials, A.S.T.M.* Designation A120.

[a] Sizes larger than those shown in the table are measured by their outside diameter, such as 14 in. outside diameter, etc. These larger sizes will be furnished with plain ends, unless otherwise specified. The weights will correspond to the manufacturers' published standards although it is possible to calculate the theoretical weights for any given size and wall thickness on the basis of 1 cu in. of steel weighing 0.2833 lb.

[b] The American Standard for Wrought-Iron and Wrought-Steel Pipe *A.S.A.* B36.10-1939 has assigned no schedule number to *Double Extra-Strong* pipe.

[c] A 10 in. Standard Weight pipe is also available with 0.279 in. wall thickness, but this wall is not covered by a Schedule Number.

[d] Owing to a departure from the *Standard-Weight* and *Extra-Strong* wall thicknesses for the 12 in. nominal size, Schedules 40 and 60, Table 2 of the *A.S.A.* B36.10-1939, Standard for Wrought-Iron and Wrought-Steel Pipe, the regular *Standard* and *Extra-Strong* wall thicknesses (0.375 in. and 0.500 in.) have been substituted.

weights of each nominal size for manufacturing reasons as well as to afford interchangeability in threading, and other elements associated with fabrication and erection. Hence the difference in wall thickness is accompanied by a corresponding change in inside diameter. In sizes up to 14 in., pipe is designated by its nominal size which corresponds roughly to the inside diameter of Schedule 40 pipe. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.), and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by Schedule 40 (standard weight) pipe, the erection

of pipe by welding sometimes warrants using lighter wall thicknesses. The considerations governing pipe wall thickness and its relation to joint design are covered in the *American Standard Code for Pressure Piping, A.S.A.* B31.1-1942, see Section 122. Standard schedules of pipe thicknesses are contained in the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10, which includes standard-weight and

Table 3. Standard Dimensions and Weights, and Tolerances in Diameter and Wall Thickness for Copper Water Tubes[a]

(All tolerances in this table are plus and minus except as otherwise indicated)

Standard Water Tube Size, In.	Actual Outside Diameter In.	Average Outside Diameter Tolerance, In.		Wall Thickness In. Type K		Type L		Type M		Theoretical Weight, Lb per Ft		
		Annealed	Drawn Temper	Nominal	Tolerance	Nominal	Tolerance	Nominal	Tolerance	Type K	Type L	Type M
1/8	0.250	0.002	0.001	0.032	0.003	0.025	0.0025	0.025	0.0025	0.085	0.068	0.068
1/4	0.375	0.002	0.001	0.032	0.004	0.030	0.0035	0.025	0.0025	0.134	0.126	0.107
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.145
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.204
5/8	0.750	0.0025	0.001	0.049	0.004	0.042	0.0035	0.030	0.0025	0.418	0.362	0.263
3/4	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.465
1¼	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.682
1½	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.940
2	2.125	0.005	0.002	0.083	0.007	0.070	0.006	0.058	0.006	2.06	1.75	1.46
2½	2.625	0.005	0.002	0.095	0.007	0.080	0.006	0.065	0.006	2.93	2.48	2.03
3	3.125	0.005	0.002	0.109	0.007	0.090	0.007	0.072	0.006	4.00	3.33	2.68
3½	3.625	0.005	0.002	0.120	0.008	0.100	0.007	0.083	0.007	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.010	0.110	0.009	0.095	0.009	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.010	0.125	0.010	0.109	0.009	9.67	7.61	6.66
6	6.125	0.005	0.002	0.192	0.012	0.140	0.010	0.122	0.010	13.9	10.2	8.92
8	8.125	0.006	+0.002 −0.004	0.271	0.016	0.200	0.014	0.170	0.014	25.9	19.3	16.5
10	10.125	0.008	+0.002 −0.006	0.338	0.018	0.250	0.016	0.212	0.015	40.3	30.1	25.6
12	12.125	0.008	+0.002 −0.006	0.405	0.020	0.280	0.018	0.254	0.016	57.8	40.4	36.7

[a] From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials A.S.T.M.* Designation B88-41.

Note 1:—For copper gas and oil burner tubes, the tolerances shown above for various wall thicknesses (type K) apply irrespective of diameter.

Note 2:—For tubes other than round no standard tolerances are established. These tolerances do not apply to condenser and heat exchanger tubes.

extra-strong thicknesses in Schedules 40 and 80, respectively, and eight other schedules of varying wall thickness to provide for different service conditions. Dimensions and other useful data for Schedules 30 and 40 pipe are given in Table 1. Table 2 from *A.S.T.M.* Specifications A53 and A120 combines the schedule thicknesses of *A.S.A.* B36.10 and the old series designations.

Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together.

Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials.* There are

TABLE 4. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Psig	Temperature Fahrenheit Degrees	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe	Pressure Psig	Temperature Fahrenheit Degrees	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe
		−20	0	0	0	0	2.5	220	1.634	1.852	1.936	2.720
		0	0.127	0.145	0.152	0.204	10.3	240	1.780	2.020	2.110	2.960
		20	0.255	0.293	0.306	0.442	20.7	260	1.931	2.183	2.279	3.189
		40	0.390	0.430	0.465	0.655	34.5	280	2.085	2.350	2.465	3.422
29.39	....	60	0.518	0.593	0.620	0.888	52.3	300	2.233	2.519	2.630	3.665
28.89	....	80	0.649	0.725	0.780	1.100	74.9	320	2.395	2.690	2.800	3.900
27.99	....	100	0.787	0.898	0.939	1.338	103.3	340	2.543	2.862	2.988	4.145
26.48	....	120	0.926	1.055	1.110	1.570	138.3	360	2.700	3.029	3.175	4.380
24.04	....	140	1.051	1.209	1.265	1.794	180.9	380	2.859	3.211	3.350	4.628
20.27	....	160	1.200	1.368	1.427	2.008	232.4	400	3.008	3.375	3.521	4.870
14.63	....	180	1.345	1.528	1.597	2.255	293.7	420	3.182	3.566	3.720	5.118
6.45	....	200	1.495	1.691	1.778	2.500	366.1	440	3.345	3.740	3.900	5.358

temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Type *K*—Designed for underground services and general plumbing service.
Type *L*—Designed for general plumbing purposes.
Type *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work, or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. In new or exposed work copper pipe of a hard temper is generally used. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall-thickness tolerances for these classes of copper tubing are given in Table 3. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper

tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 deg or more above room temperature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion*, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast-iron, steel, wrought-iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 4.

The three methods by which the elongation due to thermal expansion may be taken care of are: (1) Expansion joints; (2) Swivel joints; (3) Inherent

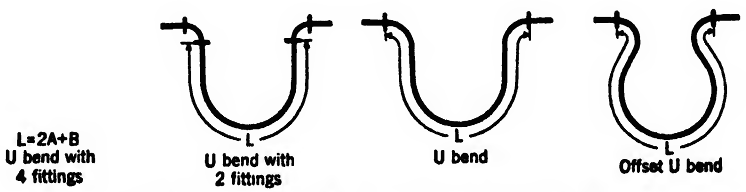


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used to some extent in low-pressure steam and hot-water heating systems, and in hot-water supply lines. Since swivel joints permit the expansive movement of the pipe by turning of threaded joints, which may ultimately result in a leak, it is preferable to provide sufficient flexibility without resorting to swiveling in the threads.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating installations.

Fig. 1 shows several types of expansion bends commonly used for taking

up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from Equation 1.

$$L = 6.16 \sqrt{D\Delta} \tag{1}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 psi. When square type bends are used, the width of the bend should not exceed about twice the height, since for a given total length of pipe in the bend, the height of the bend becomes progressively less with increase in width until the height approaches zero and no flexibility exists. Actually, wide bends utilize to best advantage the inherent flexibility of the line, but such bends cannot be proportioned on the basis of Equation 1. For such applications, more accurate methods[1] should be employed. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter. Use of welding elbows with radii of 1½ times the pipe diameter will decrease the end thrusts somewhat but will raise the fiber stress correspondingly.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

Proper anchoring of piping is especially necessary with light-weight radiators, to allow for freedom of expansion in order that no pipe strain will distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters, they usually emit disturbing sounds.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support due to expansion, or the bottom support should be designed to withstand the entire load.

THREADING PRACTICE

In all threaded pipe for heating and ventilating installations the American Standard taper pipe thread, *A.S.A.* B2.1-1942 is used. This thread is cut

with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. The number of threads per inch varies with the pipe size. Threads for fittings are the same, except that it is regular practice to furnish straight tapped couplings for Schedule 40 pipe 2 in. and smaller. For steam pressures in excess of 25 psi, it is recommended that taper-tapped couplings be used to obtain a tight joint. These may be secured by ordering *line* pipe[2] which is used for oil piping, the couplings of which are provided with taper-tapped threads and may be used with regular mill-threaded standard weight pipe. Thread lengths should be in accordance with *A.S.A.* B2.1. Right-hand threads are used unless otherwise ordered. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are

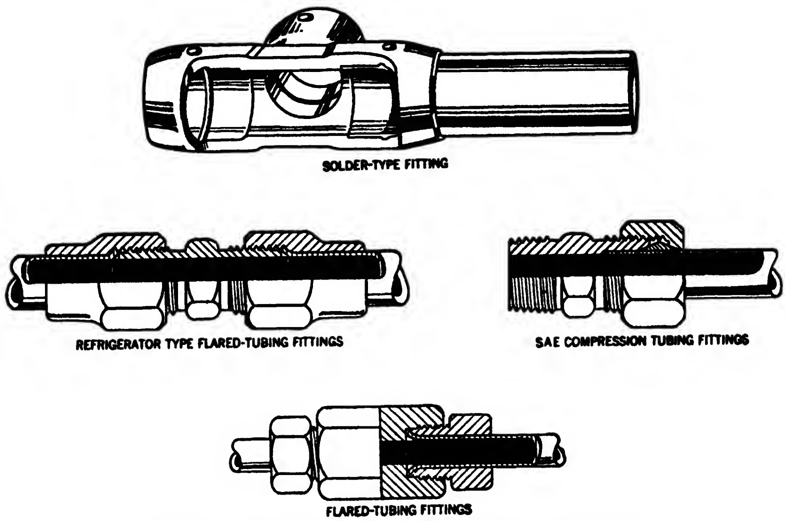


Fig. 2. Copper or Brass Tubing Fittings

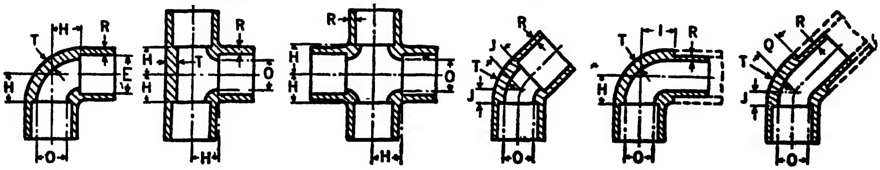
known to the trade as pitch elbows and are commercially available. All threaded pipe joints should be made up with a thread paste suitable for the service for which the pipe is to be used.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this, malleable-iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively. Screwed galvanized fittings are made according to the 150 lb American Standard.

TABLE 5. AMERICAN STANDARD DIMENSIONS OF ELBOWS, TEES, CROSSES, AND 45 DEG ELBOWS, SOLDERED-JOINT FITTINGS, *A.S.A.* A40.3-1941



Nominal Size[a]	Cast Brass[b]							Wrought Metal	Bore of Fittings
	Laying Length, Tee, Ell, and Cross[b]	Laying Length, Ell With External Shoulder	Laying Length, 45 Deg Ell	Laying Length, 45 Deg Ell External Shoulder	Inside Diameter of Fittings,[c] Min.	Metal Thickness[d]		Metal Thickness Min.[f]	
	H	I	J	Q	O	T	R	T and R	E, Min.
1/4	1/4	3/8	3/16	1/4	0.31	0.08	0.048	0.030	0.378
3/8	5/16	7/16	3/16	5/16	0.43	0.08	0.048	0.035	0.503
1/2	7/16	9/16	3/16	5/16	0.54	0.09	0.054	0.040	0.628
3/4	9/16	11/16	1/4	3/8	0.78	0.10	0.060	0.045	0.878
1	3/4	7/8	5/16	7/16	1.02	0.11	0.066	0.050	1.1285
1 1/4	7/8	1	7/16	9/16	1.26	0.12	0.072	0.055	1.3785
1 1/2	1	1 1/8	1/2	5/8	1.50	0.13	0.078	0.060	1.629
2	1 1/4	1 3/8	9/16	3/4	1.98	0.15	0.090	0.070	2.129
2 1/2	1 1/2	1 5/8	5/8	7/8	2.46	0.17	0.102	0.080	2.629
3	1 3/4	1 7/8	3/4	1	2.94	0.19	0.114	0.090	3.129
3 1/2	2	2 1/8	7/8	1 1/8	3.42	0.20	0.120	0.100	3.629
4	2 1/4	2 3/8	15/16	1 1/4	3.90	0.22	0.132	0.110	4.129
5	3 1/8	......	1 7/16	......	4.87	0.28	0.168	0.125	5.129
6	3 5/8	......	1 5/8	......	5.84	0.34	0.204	0.140	6.129

All dimensions given in inches.

[a] This size is the nominal bore of the tube.

[b] These dimensions may be used for wrought-metal fittings as well as for cast-brass fittings at manufacturer's option.

[c] This dimension is the same as the inside diameter Class L tubing (American Standard Specifications for Copper Water Tube, *A.S.A.* H23.1-1939 (*A.S.T.M.* B88).

[d] Patterns shall be designed to produce body thicknesses given in the table. Metal thickness at no point shall be less than 90 per cent of the thicknesses given in the table.

[e] This dimension has the same thickness as Type L tubing.

[f] These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used.

NOTE 1:—Wrought fittings, as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket.

NOTE 2:—Street fittings with male ends are for use in connection with other fittings illustrated.

As in the case of pipe, several weights of fittings are manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure, *A.S.A.* B16b2, B16a, and B16b respectively.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure, *A.S.A.* B16c.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure, *A.S.A.* B16d.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure, *A.S.A.* B16e.

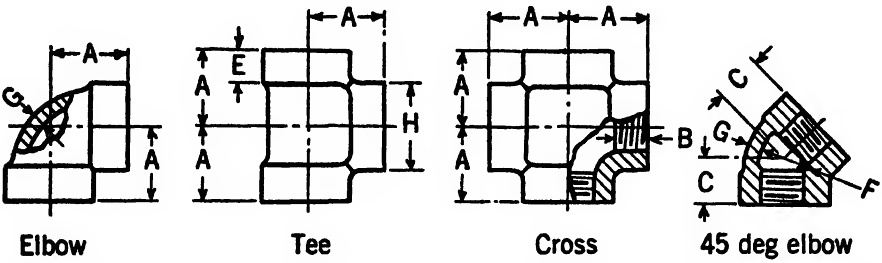
The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

War standard ratings in effect for the duration of the emergency permitted higher ratings for certain sizes of the 125 lb cast-iron flanged

fitting standard, and for 300 lb steel flanges and flanged fittings than those shown in the regular American Standards mentioned previously.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings of steel or wrought-iron; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees;

TABLE 6. AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45-DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR CLASS 125 CAST-IRON SCREWED FITTINGS, *A.S.A.* B16a-1939



Nominal Pipe Size	A	C	B	E	F		G	H
	Center to End, Elbows, Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread, Min.	Width of Band, Min.	Inside Diameter of Fitting		Metal Thickness,[a] Min.	Outside Diameter of Band, Min.
					Min.	Max.		
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08[b]	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50[b]	5.97	1.88	1.88	12.750	12.850	0.800	15.47

All dimensions given in inches.

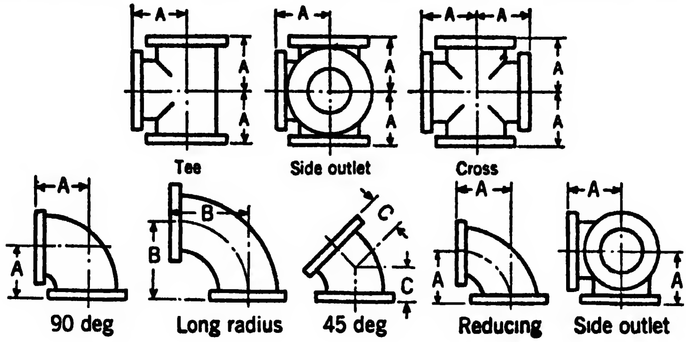
[a] Patterns shall be designed to produce castings of metal thickness given in the table. Metal thickness at no point shall be less than 90 per cent of the thickness given in the table.

[b] Applies to elbows and tees only.

crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types are shown in Fig. 2.

TABLE 7. AMERICAN STANDARD DIMENSIONS OF TEES, CROSSES[a] (STRAIGHT SIZES), AND ELBOWS FOR CLASS 125 CAST-IRON FLANGED FITTINGS, *A.S.A.* B16a-1939



NOMINAL PIPE SIZE[b-e]	A: CENTER TO FACE TEES, CROSSES[c-d] AND ELBOWS	AA: FACE TO FACE TEES AND CROSSES[c-d]	B: CENTER TO FACE LONG RADIUS ELBOWS[e,f,g]	C: CENTER TO FACE 45 DEG ELBOWS	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN.	METAL[h] THICKNESS OF BODY
1	3½	7	5	1¾	4¼	7/16	5/16
1¼	3¾	7½	5½	2	4⅝	½	5/16
1½	4	8	6	2¼	5	9/16	5/16
2	4½	9	6½	2½	6	⅝	5/16
2½	5	10	7	3	7	11/16	5/16
3	5½	11	7¾	3	7½	¾	⅜
3½	6	12	8½	3½	8½	13/16	7/16
4	6½	13	9	4	9	15/16	½
5	7½	15	10¼	4½	10	15/16	½
6	8	16	11½	5	11	1	9/16
8	9	18	14	5½	13½	1⅛	⅝
10	11	22	16½	6½	16	1 3/16	¾
12	12	24	19	7½	19	1¼	13/16
14 O.D.	14	28	21½	7½	21	1⅜	⅞
16 O.D.	15	30	24	8	23½	1 7/16	1
18 O.D.	16½	33	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	36	29	9½	27½	1 11/16	1⅛
24 O.D.	22	44	34	11	32	1⅞	1¼
30 O.D.	25	50	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	56	49	18	46	2⅜	1⅝
42 O.D.	31	62	56½	21	53	2⅝	1 13/16
48 O.D.	34	68	64	24	59½	2¾	2

All dimensions given in inches.

[a] Crosses both straight and reducing sizes 18 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.

[b] Size of all fittings listed indicates nominal inside diameter of port.

[c] Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[d] Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[e] Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[f] Special degree elbows, ranging from 1 to 45 deg, inclusive, shall have the same center to face dimensions as given for 45-deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

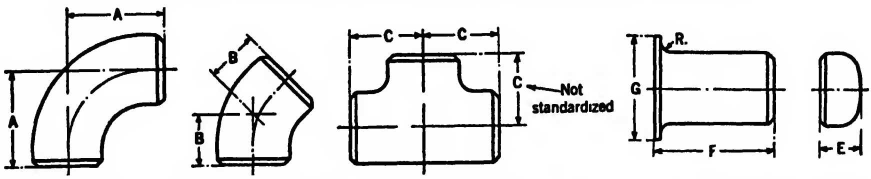
[g] Side outlet elbows shall have all openings on intersecting center-lines.

[h] Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection. Table 5 from *A.S.A.* Standard A40.3-1941 contains dimensions for soldered joint elbows, tees, crosses, and 45 deg elbows.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in both large and small sizes. An American Standard, *A.S.A.* A40.2-1936 has been prepared to standardize dimensions for brass fittings for flared copper water tubes. Flared tube fittings are widely used in refrigerating work where *S.A.E.* dimensions

TABLE 8. AMERICAN STANDARD DIMENSIONS FOR BUTT-WELDING ELBOWS, TEES, CAPS, AND LAPPED-JOINT STUB ENDS, *A.S.A.* B16.9-1940



Nominal Pipe Size	Outside Diameter	Center-to-End			Caps *E* [b-c]	Lapped-Joint Stub Ends		
		90-Deg Elbows *A*	45-Deg Elbows *B*	Of Run Tee *C* [a]		Length *F* [b]	Radius of Fillet *R*	Diam. of Lap *G* [d]
1	1.315	1 1/2	7/8	1 1/2	1 1/2	4	1/8	2
1 1/4	1.660	1 7/8	1	1 7/8	1 1/2	4	3/16	2 1/2
1 1/2	1.900	2 1/4	1 1/8	2 1/4	1 1/2	4	1/4	2 7/8
2	2.375	3	1 3/8	2 1/2	1 1/2	6	5/16	3 5/8
2 1/2	2.875	3 3/4	1 3/4	3	1 1/2	6	5/16	4 1/8
3	3.500	4 1/2	2	3 3/8	2	6	3/8	5
3 1/2	4.000	5 1/4	2 1/4	3 3/4	2 1/2	6	3/8	5 1/2
4	4.500	6	2 1/2	4 1/8	2 1/2	6	7/16	6 3/16
5	5.563	7 1/2	3 1/8	4 7/8	3	8	7/16	7 5/16
6	6.625	9	3 3/4	5 5/8	3 1/2	8	1/2	8 1/2
8	8.625	12	5	7	4	8	1/2	10 5/8
10	10.750	15	6 1/4	8 1/2	5	10	1/2	12 3/4
12	12.750	18	7 1/2	10	6	10	1/2	15

All dimensions given in inches.

[a] The dimensions of welding tees cover those which have side outlets from one size less than half the size of the run-way opening of the tees to full size.

[b] Dimensions *E* and *F* are applicable only to these fittings in schedules up to and including Schedule 80, *A.S.A.* Standard B36.10-1939.

[c] The shape of these caps shall be ellipsoidal and shall conform to the requirements of the *A.S.M.E.* Boiler Construction Code.

[d] This dimension is for standard machined facings in accordance with American Standard for Steel Pipe Flanges and Flanged Fittings (*A.S.A.* B16e-1939). The back face of the lap shall be machined to conform to the surface of the flange on which it seats. Where ring joint facings are to be applied, use dimension *K* as given in *A.S.A.* B16e-1939.

and a 45-deg flare render most fittings interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with *S.A.E.* dimensions are not interchangeable with the American Standard fittings for water tubes.

Ammonia pipe fittings made of cast-iron were formerly used extensively in handling refrigerants in large installations. Replacement of ammonia by other refrigerants operating at lower pressures has seriously curtailed the market for these fittings. For this reason formulation of an American Standard for these fittings was abandoned by the *A.S.A.* in 1936.

FLANGE FACINGS AND GASKETS

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 psi and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$-in. raised face. The standard facing for steel flanged fittings for 150 and 300 psi is a $\frac{1}{16}$-in. raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees, and crosses for 125 lb cast-iron screwed fittings are given in Table 6, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Table 7.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a narrow recessed facing.

WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as an alternate method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding on low to medium pressure heating jobs.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds. Welding processes and procedure are described in various publications.

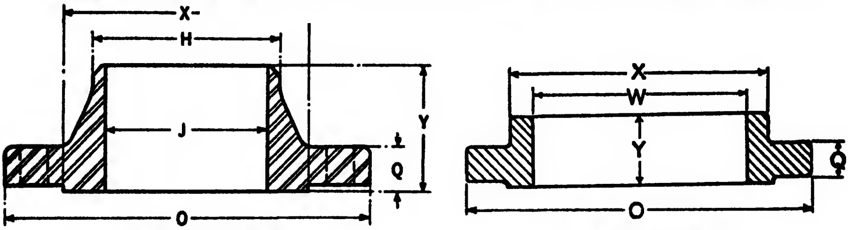
Welding application requires the same basic knowledge of design as do the other types of assembly, but, in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests.

Rules for fusion welding of pipe joints, the qualification of welding operators, welding procedures and the testing theoreof, are contained in the Standard Manual on Pipe Welding of the Heating, Piping & Air Conditioning Contractors National Association, and other nationally recognized groups[3, 4, 5]. In general, the wall thickness and chemical

analysis of the pipe are the governing factors, not the working pressure. There are a number of safety codes which govern the installation of welded piping in many cities and states. Some of the more prominent are listed at the end of this chapter[4, 5, 6].

A complete line of manufactured steel welding fittings is now available

Table 9. American Standard Dimensions of Steel Welding Neck and Slip-on Welding Flanges for Steam Service Pressure Rating of 150 PSI (Gage) at a Temperature of 500 F, and 100 PSI (Gage) at 750 F, *A.S.A.* B16e-1939



WELDING NECK SLIP-ON NECK

Nominal Pipe Size	Diameter of Flange	Thickness of Flg.[a] Min.	Diameter of Hub	Hub Diam. Beginning of *H*	Length Thru Hub[a]	Inside Diam. of Pipe Schedule 40[c]	Bore of Slip-on Flanges Min. *W*	Diam. of Bolt Circle	No. of Bolts	of Bolts
				0.84		0.62*	0.88		4	
				1.05		0.82*	1.09		4	
				1.32		1.05*	1.38		4	
			2 5/16	1.66		1.38*	1.72		4	
			2 9/16	1.90		1.61*	1.97		4	
			3 1/	2.38		2.07*	2.44		4	
				2.88		2.47*	2.94		4	
				3.50		3.07*	3.56		4	
3½				4.00		3.55*	4.06		8	
4			5 5/16	4.50		4.03*	4.56		8	
5			6 7/16	5.56		5.05*	5.66		8	
6			7 9/16	6.63		6.07*	6.72		8	
8			9 11/16	8.63		7.98*	8.72		8	
10			12	10.75		10.02*	10.88		12	
12				12.75		....	12.88		12	
14 O.D.				14.00		To Be	14.19		12	
16 O.D.			18	16.00		Specified	16.19		16	
18 O.D.				18.00		by	18.19		16	
20 O.D.			22	20.00		Purchaser	20.19		20	
24 O.D.			26⅛	24.00			24.19		20	1¼

All dimensions given in inches.

[a]A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub*.

[b]The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[c]Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *A.S.A.* B36.10-1939, Schedule 40.

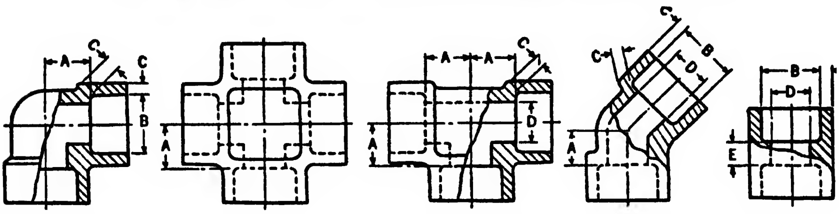
*These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes.

and a dimensional standard has been prepared under the procedure of the *American Standards Association* to unify heretofore divergent dimensions for the same type welding fittings as produced by different manufacturers. Standard dimensions for steel butt-welding elbows, tees, caps, and lapped-joint stub ends are given in Table 8. Dimensions for eccentric and concentric reducers, and 180-deg return bends are not shown in Table 8 but

are included in the American Standard. Larger sizes also are available in some types of fittings. The welding bevel which is a straight 37½-deg V for wall thickness ¾ in. and below, and a U-bevel for thicknesses heavier than ¾ in., conforms to the recommended practice of *A.S.A.* Standard B16e-1939, American Standard for Steel Pipe Flanges and Flanged Fittings. The latter also contains dimensions for steel welding neck flanges for pressures up to 2500 psi, and slip-on welding flanges for 150 and 300 psi. Table 9 gives these dimensions for welding-neck and slip-on welding flanges suitable for 150 psi gage pressure.

Socket-welding fittings also are commercially available. These fittings have a machined recess for inserting the pipe which is attached by a

TABLE 10. PROPOSED AMERICAN STANDARD DIMENSIONS OF SOCKET-WELDING ELBOWS, TEES, CROSSES, 45-DEG ELBOWS, AND COUPLINGS



Nominal Pipe Size	Minimum Depth of Socket	Center to Bottom of Socket				Couplings Distance Between Bottom Sockets	Bore Diameter of Socket, Minimum	Minimum Socket Wall Thickness			Bore Diameter of Fittings		
		90-Deg Ells, Tees, Crosses		45-Deg Ells									
		Sched. 40 & 80	Sched. 160	Sched. 40 & 80	Sched. 160			Sched. 40	Sched. 80	Sched. 160	Sched. 40	Sched. 80	Sched. 160
		A		A		E	B	Cª			D		
¼	⅜	17/32	....	5/16	....	¼	0.555	0.156	0.156	....	0.364	0.302	....
⅜	⅜	17/32	....	5/16	....	¼	0.690	0.156	0.158	....	0.493	0.423	....
½	⅜	⅝	¾	7/16	½	⅜	0.855	0 156	0.184	0.234	0.622	0.546	0.466
¾	½	¾	⅞	½	9/16	⅜	1.065	0.156	0.193	0.273	0.824	0.742	0.614
1	½	⅞	1 1/16	9/16	11/16	½	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1¼	½	1 1/16	1¼	11/16	13/16	½	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	½	1¼	1½	13/16	1	½	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	⅝	1½	1⅝	1	1⅛	¾	2.406	0 193	0.273	0.429	2.067	1.939	1.689
2½	⅝	1⅝	2¼	1⅛	1¼	¾	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	⅝	2¼	2½	1¼	1⅜	¾	3.535	0.270	0.375	0.546	3.068	2.900	2.626

All dimensions are given in inches.

ªDimension *C* is 1¼ times the nominal pipe thickness, minimum, but not less than 1/16 in.

Reducing sizes have same center to bottom of socket dimension as the largest size of reducing fitting.

fillet weld between the pipe wall and socket end. Use of socket-welding fittings generally is restricted to nominal pipe sizes 3 in. and smaller in which range commercial fittings are available. This type of fitting has gained rapid acceptance owing to its ease of installation, low cost, and ability to make a pressure tight joint without weakening the pipe as is the case with threading. Dimensions for socket-welding fittings as offered by most manufacturers of the product are given in Table 10.

VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

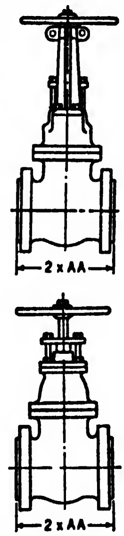
The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger

sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure.

Table 11. American Standard Contact Surface to Contact Surface Dimensions of Cast-Iron and Steel Flanged Wedge Gate Valves, A.S.A. B16.10-1939

Nominal Pipe Size	Contact Surface to Contact Surface Dimensions, (2 × AA)				
	Cast-Iron[a]			Steel	
	125	175[b,c]	250[b]	150[b]	300[b]
1	....	....	....	....	....
1¼	....	....	....	....	....
1½	....	....	....	....	7½
2	7	7¼	8½	7	8½
2½	7½	8	9½	7½	9½
3	8	9¼	11⅛	8	11⅛
3½	8½	10	11⅞	8½	11⅞
4	9	10½	12	9	12
5	10	11½	15	10	15
6	10½	13	15⅞	10½	15⅞
8	11½	14¼	16½	11½	16½
10	13	16¾	18	13	18
12	14	17½	19¾	14	19¾
14 O.D.	15	....	22½	15	30
16 O.D.	16	....	24	16	33
18 O.D.	17	....	26	17	36
20 O.D.	18	....	28	18	39
24 O.D.	20	....	31	20	45



All dimensions given in inches.

[a]These dimensions are the same for Cast-Iron Double Disc Flanged Gate Valves.

[b]These are pressure designations which refer to the primary service ratings in pounds per square inch of the connecting end flanges.

[c]The connecting end flanges of 175 lb valves are the same as those on 250 lb valves.

Note 1:—Where dimensions are not given, the sizes either are not made or there is insufficient demand to warrant the expense of unification.

Note 2:—Female and groove joint facings have bottom of groove in same plane as *flange edge*, and center to contact surface dimensions for these facings are reduced by the amount of the raised face.

The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum, but they should not be used where it is desired to throttle the flow; globe valves should be used for this purpose. Gate valves may be secured with either a rising or a non-rising stem, although in the smaller size the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers

a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

An American Standard, *A.S.A.* B16.10-1939, has been prepared giving the face-to-face dimensions of ferrous flanged and welding-end valves. The following types are covered: wedge gate, double disc gate, globe and angle, and swing check. One purpose of establishing these dimensions is to insure that gate valves of a given rating and flange dimension of either the wedge or double disc design will be interchangeable in a pipe line. Contact surface to contact surface dimensions of cast-iron and steel flanged wedge-gate valves are given in Table 11. End-to-end dimensions for steel butt-welding valves in sizes up to 8 in., inclusive, are the same as those given in Table 11 for steel valves.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve.

Automatic control of steam supply to individual radiators can be effected by use of direct-acting radiator valves having a thermostatis element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a $\frac{1}{16}$-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made for use in hot water heating systems are of simpler design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when a vacuum tends to be formed the air is drawn back into the radiator.

REFERENCES

[1] See (1) Piping Handbook, by Walker and Crocker (McGraw-Hill Co.); (2) A Manual for The Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, E. T. Cope, (The Detroit Edison Company).

[2] See *API Specification* 5L *for Line Pipe, American Petroleum Institute.*

[3] Standard Manual on Pipe Welding, *Heating, Piping and Air Conditioning Contractors National Association.* Welding Handbook, *American Welding Society,* 1942.

[4] ASME Power Boiler Code, *American Society of Mechanical Engineers*.

[5] American Standard Code for Pressure Piping, *A.S.A.* B-31.1—1942, *American Standards Association*.

[6] Marine Engineering Regulations of the Coast Guard, *American Bureau of Shipping*.

General Specifications for Inspection of Material, Appendix VII, Welding, *U. S. Navy*. Specifications for Welding, Appendix 5, Part 1—General—for vessels of the *U. S. Navy, Bureau of Ships*, April, 1940.

CHAPTER 28

PIPE INSULATION

Heat Losses from Bare and Insulated Pipes, Low Temperature Pipe Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

THE heat loss from uninsulated pipes may be of considerable magnitude if the temperature of the surrounding medium differs appreciably from that of the fluid conveyed. Losses are increased by rapid motion of the surrounding air or by contact of the pipe with bodies of high conductivity. Careful consideration must, therefore, be given to this factor in a properly designed system and adequate insulation provided, if necessary.

HEAT LOSSES FROM BARE PIPES

Heat losses from horizontal bare steel pipes, based on tests at *Mellon Institute* and calculated from the fundamental radiation and convection equations (Chapter 5), are given in Table 1. Heat losses from horizontal copper tubes and pipes with tarnished surfaces, are given in Table 2[1].

Heat losses from bare pipe of materials having lower emissivities may be calculated from data appearing in Chapter 5.

The area in square feet per linear foot of pipe is given in Table 3 for various standard pipe sizes, and Table 4 for copper tubing, while Table 5 gives the area in square feet of flanges and fittings for various standard pipe sizes. These tables can be used to advantage in estimating the amount of insulation required.

Very often, when pipes are insulated, flanges and fittings are left bare so as to allow for easy access to the fittings in case of repairs. The fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

Examples 1 and *2* show how the annual heat loss from uncovered pipe and its dollar value may be computed from the data in Table 1.

Example 1. Compute the total annual heat loss from 165 ft of 2 in. bare pipe in service 4000 hr per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

Solution. The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained by interpolation from Steam Tables. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 F. By interpolation of Table 1 between temperature differences of 157.1 and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 F is found to be 1.624 Btu per (hr) (linear ft) (F deg). The total annual heat loss from the entire line = 1.624 × 169.4 × 165 (linear ft) × 4000 (hr) = 181,600 Mb. (Mb = 1000 Btu.)

Example 2. Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in the previous example. If the system is operating at an over-all efficiency of 55 per cent, determine the monetary value of the annual heat loss from the line.

Solution. The cost of heat per 1000 Mb supplied to the system = 1,000,000 × 11.5 (dollars) ÷ [13,000 (Btu) × 2000 (lb) × 0.55 (efficiency)] = $0.804. The total cost of heat lost per year = 0.804 × 181.6 (thousand Mb) = $146.00.

PIPE INSULATIONS

Pipe insulations are of several general forms and are made of various types of material. The most common form is the rigid sectional covering either split longitudinally into halves or cut through on one side and scored on the other, to facilitate assembling on pipes. Preformed ma-

Table 1. Heat Losses from Horizontal Bare Steel Pipes

Expressed in Btu per (hour) (linear foot) (Fahrenheit degree difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.250	0.287	0.300	0.321	0.433	0.500	0.530
¾	0.340	0.381	0 409	0.429	0.533	0 543	0.654
1	0.440	0.475	0.509	0.536	0.636	0.746	0.803
1¼	0.500	0.559	0.618	0.622	0.764	0.878	0.934
1½	0.580	0.656	0.710	0.750	0.904	1.053	1.120
2	0.730	0.825	0.890	0.957	1.101	1.273	1.364
2½	0.880	1.000	1.091	1.143	1.305	1.490	1.605
3	1.040	1.175	1.272	1.343	1.560	1.800	1.940
3½	1.180	1.350	1.454	1.535	1.750	2.020	2.170
4	1.460	1.500	1.635	1.715	1.941	2.240	2.430
4½	....	....	....	....	2.131	2.465	2.650
5	1.600	1.812	1.980	2.071	2.387	2.770	2.990
6	1.840	2.125	2.270	2.430	2.740	3.210	3.440
8	2.400	2.685	2.910	3.110	3.310	4.050	4.370

Table 2. Heat Loss from Horizontal Tarnished Copper Pipe

Expressed in Btu per (hour) (linear foot) (Fahrenheit degree difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	299.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.455	0.495	0.546	0.584	0.612	0.706	0.760
¾	0.555	0.605	0.666	0.715	0.748	0.866	0.933
1	0.684	0.743	0.819	0.877	0.919	1.065	1.147
1¼	0.847	0.919	1.014	1.086	1.138	1.324	1.425
1½	0.958	1.041	1.148	1.230	1.288	1.492	1.633
2	1.180	1.281	1.412	1.512	1.578	1.840	1.987
2½	1.400	1.532	1.683	1.796	1.883	2.190	2.363
3	1.680	1.825	2.010	2.153	2.260	2.630	2.840
3½	1.900	2.064	2.221	2.433	2.552	2.974	3.215
4	2.118	2.302	2.534	2.717	2.850	3.320	3.590
5	2.580	2.804	3.084	3.303	3.470	4.050	4.385
6	3.036	3.294	3.626	3.886	4.074	4.765	5.160
8	3.880	4.215	4.638	4.960	5.210	6.100	6.610
10	4.760	5.180	5.680	6.090	6.410	7.490	8.115
12	5.590	6.070	6.670	7.145	7.500	8.800	9.530

terials are supplied in segments for assembly on large pipes. The sectional coverings are generally supplied with a pasted-on canvas jacket. Blanket insulations are sometimes used for wrapping large pipes, particularly where removal for frequent servicing of the pipe is necessary. Fittings and bends are commonly covered with portions of standard preformed

TABLE 3. EXTERNAL SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

insulation or, when irregular in contour, with plastic materials known as insulating cements. Insulation is secured to pipes with staples which are used to bridge the joint between half sections, and with metal pipe covering bands or rings of wire which secure individual sections and effect a junction between abutting sections. Surface finishes used over pipe insulation depend upon the service encountered and appearance desired. Canvas jackets are most common although asbestos paper or asbestos finishing cements are sometimes employed. Insulation outdoors should be waterproof, and is generally protected with an asphalt felt for piping and asphaltic cements for fittings. Insulation on lines carrying cold water, brine, or other cold fluids is carefully finished to obtain adequate sealing against the penetration of water vapor.

The selection of pipe insulation for a particular service condition must be made with full consideration of a number of properties in addition to thermal conductivity. Factors which may be of more importance than the thermal conductivity are: ease of application, fire resistance, heat stability, weathering stability, resistance to damage by physical abuse, and others which may apply to a particular installation. A complete evaluation of pipe insulation cannot be included here. Insulation manufacturers should be consulted in regard to the selection of insulation which is to meet specific requirements.

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water systems are given in Table 6. They are given as functions of the mean temperatures or the arithmetic mean of the inner and outer surface temperatures of the insulations. It should be emphasized that they are the average values obtained from a number of tests made on each type of material, also, that in the use of *conductivity* all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through 1, 1½, and 2-in. thick, 85 per cent magnesia type of insulation for temperature differences between the pipe and the surrounding atmosphere up to 280 F, are shown in Figs. 1, 2, and 3.

TABLE 4. EXTERNAL SURFACE PER LINEAR FOOT OF COPPER TUBING
Outside diameter ⅛ in. greater than nominal size

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
½	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949	....	....
1½	0.426	4	1.080	....	....

Table 5. Area of Flanged Fittings, Square Feet[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a] Including areas of accompanying flanges bolted to the fitting.

Standard thicknesses of 85 per cent magnesia pipe covering are not exactly 1 in. However, the loss through any given thickness of insulation can be obtained by interpolation. Also, the losses through any of the insulations given in Table 6 can be obtained by multiplying the losses obtained from Figs. 1, 2, or 3 by the factors given in Table 7.

Pipes operating at high temperatures are frequently insulated to the

Table 6. Thermal Conductivity (k) of Various Type Pipe Insulations for Medium and High Temperature Pipe[a]

Expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference per inch)

Types of Insulating Materials	Density Lb/Cu Ft	Temp. Range of Accepted Use	Mean Temperature, F Deg				
			100	200	300	400	500
85% Magnesia—Type	13–15	Up to 600 F	0.41	0.45	0.48	0.52	....
Corrugated Asbestos—Type							
4 Ply per 1 in.	11–13	Up to 300 F	0.57	0.68	0.80	....	....
6 Ply per 1 in.	15–17	Up to 300 F	0.51	0.59	0.69	....	....
8 Ply per 1 in.	18–20	Up to 300 F	0.49	0.57	0.65	....	....
Laminated Asbestos—Type (35–40 laminations per 1 in.)	30–35	Up to 700 F	0.39	0.44	0.49	0.54	....
Mineral Wool—Type	10–15	Up to 800 F	0.40	0.45	0.50	0.55	....
Diatomaceous Silica—Type	25–30	Up to 1900 F	0.63	0.66	0.69	0.72	0.75
Brown Asbestos Fiber—Type	13–15	Up to 1200 F	0.31	0.39	0.44	0.49	0.54

[a] Average values from various laboratories for insulating materials of various manufacturers.

Table 7. Pipe Covering Factors

Types of Insualting Materials	Temperature Difference, Pipe to Air, F Deg				
	100	200	300	400	500
Corrugated Asbestos—Type					
4 Ply per 1 in.	1.30	1.36	1.42	....	....
6 Ply per 1 in.	1.19	1.23	1.27	....	....
8 Ply per 1 in.	1.15	1.19	1.23	....	....
Laminated Asbestos—Type	0.96	0.98	1.00	1.02	1.04
Mineral Wool—Type	0.98	1.00	1.02	1.05	1.07
Diatomaceous Silica—Type	1.37	1.36	1.35	1.35	1.34
Brown Asbestos Fiber—Type	0.86	0.88	0.91	0.93	0.96

best advantage by combining a high temperature insulation near the pipe with a moderate or low temperature insulation around it as an outer layer. By this method an efficient material may be used for each of the two temperature ranges encountered. In calculating the heat loss

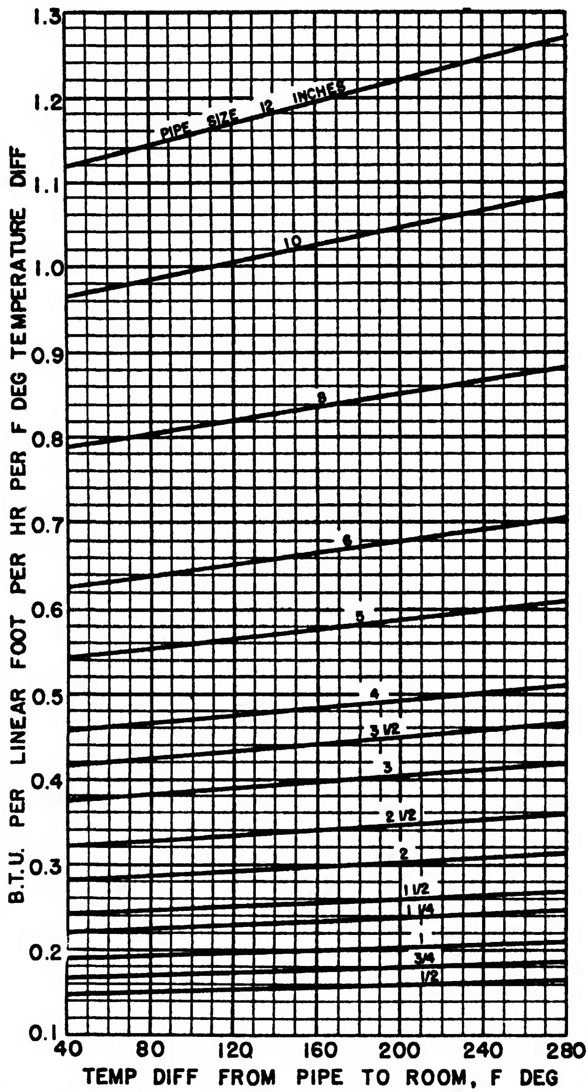


Fig. 1. Heat Loss Through 1 In. Thick 85 per cent Magnesia Type Covering

through such a combination the mean temperature of each layer must be determined along with the thickness of each. This is readily done in three or four calculations performed as a series of approximations in which assumptions of thickness and mean temperature are adjusted as indicated in the discussion which follows.

In the case of a single thickness of pipe covering, the quantity of heat

transferred per square foot of outer surface of the insulation is given by the equation:

$$\frac{k(t_1 - t_2)}{\log_e \frac{r_2}{r_1}} \qquad (1)$$

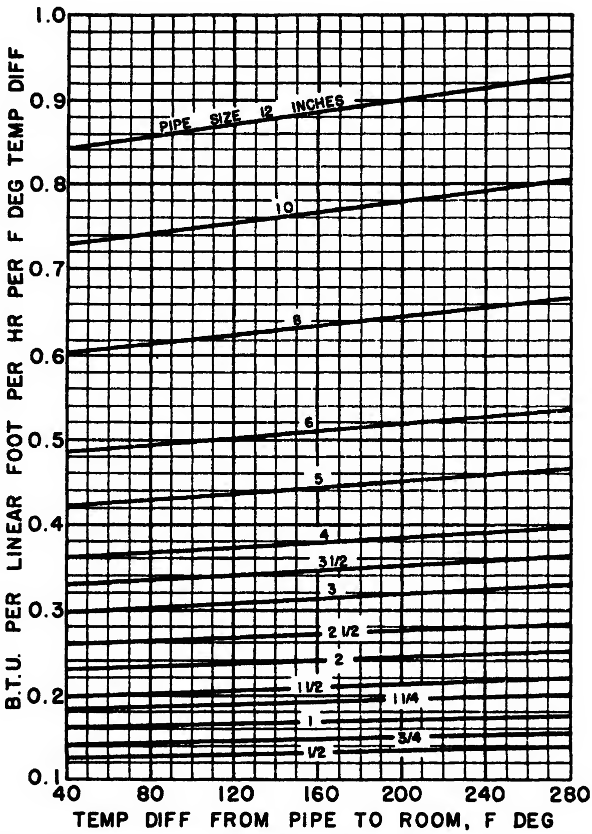


FIG. 2. HEAT LOSS THROUGH 1½ IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

where

q_0 = Btu per (hour) (square foot of outer surface of insulation).
r_1 = outer radius of pipe or inner radius of insulation, inches.
r_2 = outer radius of insulation, inches.
k = thermal conductivity of insulation, Btu per (hour) (square foot) (Fahrenheit degree per inch).
t_1 = temperature of inner surface of insulation, Fahrenheit degrees.
t_2 = temperature of outer surface of insulation, Fahrenheit degrees.

It is convenient to work from the outer surface of the insulation, since the loss through the covering must be determined from the outer surface loss by means of surface loss curves such as given in Fig. 4.

After the true heat loss is obtained, the loss per square foot of pipe surface can be calculated from the relationship:

where

q_i = Btu per (hour) (square foot outer surface of pipe).

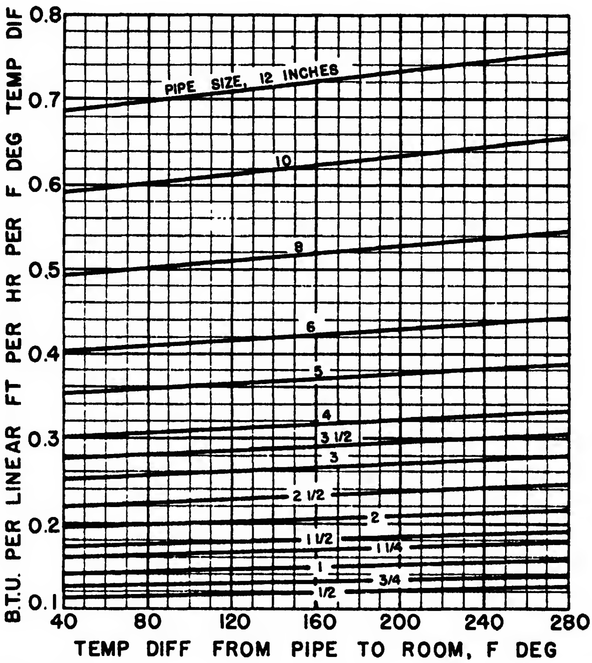


FIG. 3. HEAT LOSS THROUGH 2 IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

The heat loss through two or more thicknesses of insulation applied to a pipe can be calculated by means of the equation:

$$t_1 -$$

$$r_s \log_e \frac{r_2}{-} \quad r_s \log_e \frac{r_3}{-} \qquad (2)$$

$$k_1 \quad , \quad k_2 \quad ,$$

where

r_3 = outer radius of second layer of insulation, inches.

r_s = outer radius of last layer of insulation, inches.

The method of solving Equation 2, which is the most difficult of the two, is given in Example 3.

Example 3. Compute the heat loss per linear foot of pipe surface per hour from a 6-in. pipe, insulated with a 3-in. thickness of diatomaceous silica, and a 2-in. thickness of 85 per cent magnesia. The pipe is operating at a temperature of 1200 F and is exposed to a room temperature of 80 F.

Solution. In figuring the heat loss from Equation 2, it is necessary to first make an assumption for the outer surface temperature t_2 and the temperature between the diatomaceous silica and 85 per cent magnesia insulation, so that the mean temperature of each material can be obtained and the thermal conductivity corresponding to the mean temperature of each material substituted in the formula. First assume an outer surface temperature of 140 F and a temperature of 570 F between the two materials corresponding to a mean temperature of (1200 + 570) ÷ 2 or 885 F for the diatomaceous silica and (570 + 140) ÷ 2 or 355 F for the 85 per cent magnesia insulation. The conductivities of these two materials at mean temperatures of 885 and 355 F interpolated from Table 6 are 0.865 and 0.5 Btu, respectively.

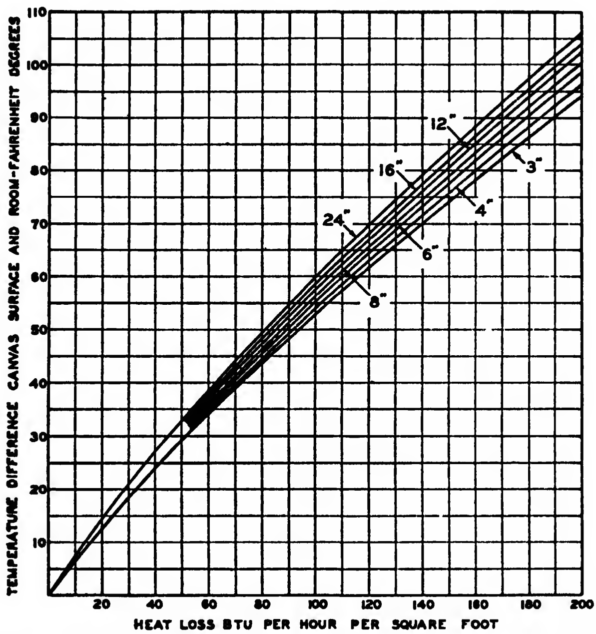


FIG. 4. HEAT LOSS FROM CANVAS-COVERED CYLINDRICAL SURFACES OF VARIOUS DIAMETERS

These values are substituted in Equation 2 and a trial calculation made. For a nominal 6-in. steel pipe $r_1 = 3.312$, $r_2 = 6.312$ and $r_3 = 8.312$ then,

$$\frac{1200 - 140}{\dfrac{8.312 \log_e \dfrac{6.312}{3.312}}{0.865} \qquad \dfrac{}{0.5}} \qquad \frac{1060}{6.2 + 4.58} \qquad 98.3 \text{ Btu.}$$

The temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 98.3 Btu is found from Fig. 4 to be 57 F for a 16-in. O.D. cylindrical surface, or 57 + 80 F room temperature = 137 F surface temperature. Since a surface temperature of 140 F was assumed, it is evident that a temperature closer to 137 F, or, for instance, 138 F should be used for recalculation:

$$\frac{1200 - 138}{6.2 + 4.58} \qquad 98.4 \text{ Btu.}$$

Since the temperature drop through each material is equal to the heat flow times the actual resistance of each material, the temperature drop through the diatomaceous silica is 98.4 × 6.2 = 610 F, or the temperature between the two insulating materials

is (1200 − 610) = 590 F. Since a temperature of 570 F between the two materials was assumed, it is obvious that a temperature closer to 590, or for instance 586 F may be selected. The mean temperatures of the two insulations corresponding to the new assumptions are (1200 + 586) ÷ 2 = 893 and (586 + 138) ÷ 2 = 362, and the interpolated conductivities corresponding to the new mean temperatures are 0.87 and 0.505 for the diatomaceous silica and 85 per cent magnesia, respectively. By substituting in Equation 2:

1200 − 138 1062

0.87 ' 0.505

Again referring to Fig. 4, it is seen that the temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 99.3 Btu = 58 F, which corresponds to the surface temperature of 138 F last assumed. The temperature drop through the diatomaceous silica is 99.3 × 6.16 = 612 F, corresponding to a temperature of 588 F between the two materials, which checks very closely with the temperature of 585 F last assumed. The heat loss is therefore 99.3 × 8.312 ÷ 3.312 or 249 Btu per sq ft of pipe surface. Since the surface area per linear foot of 6-in. pipe is 1.734 sq ft (Table 3), the heat loss per linear foot of pipe will be 249 × 1.734 = 432 Btu per hr.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces, the increases in losses due to air velocity are very small as compared with increases from bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the conductance of heat from surface to air, and cannot change the internal conductance of the insulation itself. The maximum increase in heat loss due to air velocity ranges from about 15 per cent in the case of 1-in. thick insulation, to about 5 per cent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface. If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be applied in such a manner that air circulation within it or between it and the pipe is avoided.

Fig. 4 shows the loss of heat from canvas-covered, cylindrical surfaces of various outside diameters when the surface to air temperature difference is low. The data are from tests made at Mellon Institute.

The frequent practice of omitting insulation on that portion of a pipe which passes through a masonry wall, or which may be in contact with other metals, should be avoided. Physical contact between the pipe surface and other structural materials of high thermal conductivity will result in heat transfer much greater than that shown in Tables 1 and 2 for transfer from bare pipe to air.

The saving due to use of insulation on piping is illustrated in *Example 4*.

Example 4. If the steam line given in *Examples 1* and *2* is covered with 1 in. thick 85 per cent magnesia, determine the resulting total annual loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

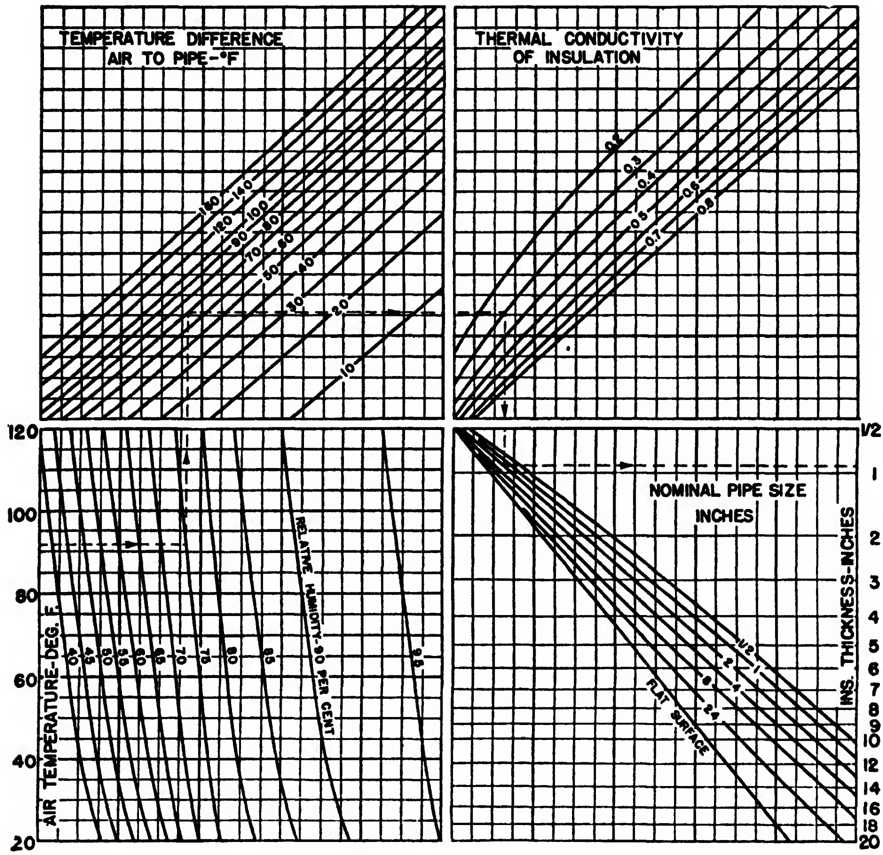
Solution. By referring to Fig. 1, the coefficient for 1 in. magnesia on a 2-in. pipe is found to be 0.300 Btu per (hr) (linear ft of pipe) (deg temperature difference) at a temperature difference of 169.4 F. The total hourly loss per linear foot of pipe will then be 0.300 × 169.4 = 50.8 Btu. The total annual loss through the insulation = 50.8 × 165 (linear ft) × 4000 (hr) = 33,500 Mb. The annual bare pipe loss as determined in the solution of *Example 1* was found to be 181,600 Mb. The saving due to insulation is then 181,600 − 33,500 = 148,100 Mb per year.

From the solution of *Example 2*, it was found that the heat supplied to the system

cost $0.804 per thousand Mb. Therefore, the monetary value of the saving = 0.804 (dollars) × 148.1 (thousand Mb) = $119.07, or 81.5 per cent of the cost when using uninsulated pipe.

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation.



[a] Solve problems as indicated by dotted line, entering chart at lower left-hand scale.

FIG. 5. THICKNESS OF PIPE INSULATION TO PREVENT CONDENSATION ON OUTER SURFACE[a]

The insulating material should absorb a minimum amount of moisture, because the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the insulation of surfaces that are below the dew-point of the surrounding air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material, and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the seal is in place and the temperature of the insulated surface reduced, release that

moisture to the cold surface. There are a number of methods of producing vapor seals, some of which have been worked out by insulation manufacturers to suit their products, and others by applicators and users. Unless time-proven methods are known, specifications of insulation manufacturers should be obtained and followed carefully.

The thickness of insulation required to prevent condensation on the outer surface is that thickness which will raise the temperature of the outer surface of the insulation to a point slightly higher than the dew-point of the surrounding vapor. The dew-point for various humidities can be readily ascertained from a psychrometric chart.

The approximate required thickness of insulation to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 5 in which a surface resistance of 0.606, corresponding to a film conductance o 1.65, was used in calculating the curves. This value provides a slight factor of safety and its use is known to give satisfactory field results. In using the chart it is advisable to specify the next thicker, rather than the next thinner, commercial insulation in cases where an intermediate thickness is indicated.

TABLE 8. HEAT GAINS FOR INSULATED COLD PIPES

Rates of heat transmission given in Btu per (hour) (Fahrenheit degree temperature difference between fluid in pipe and surrounding still air)

Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087.	0.394
¾	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0 118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0 191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.401	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0 383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

Heat gains for pipes insulated with a material having an installed conductivity of 0.30 Btu per (sq ft) (hr) (F deg per in.) are given in Table 8. This table may be used for any of the commercial insulations offered for this purpose since they have conductivities very near the 0.3 value used.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point.

Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 9 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below the freezing point of water (temperature difference, 60 F).

The last column of Table 9 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from

TABLE 9. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES WITH SURROUNDING AIR AT −18 F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42 F Water to Freezing Point			Water Flow Required at 42 F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches (Conductivity, $k = 0.30$)					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions, the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 9. However, if the water enters the pipe at 34 F, it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point

will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 9, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air, have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water, but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 9, the only safe way to insure against freezing is to install a steam or hot water line, or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

TABLE 10. THICKNESS OF PIPE INSULATION ORDINARILY USED INDOORS[a]

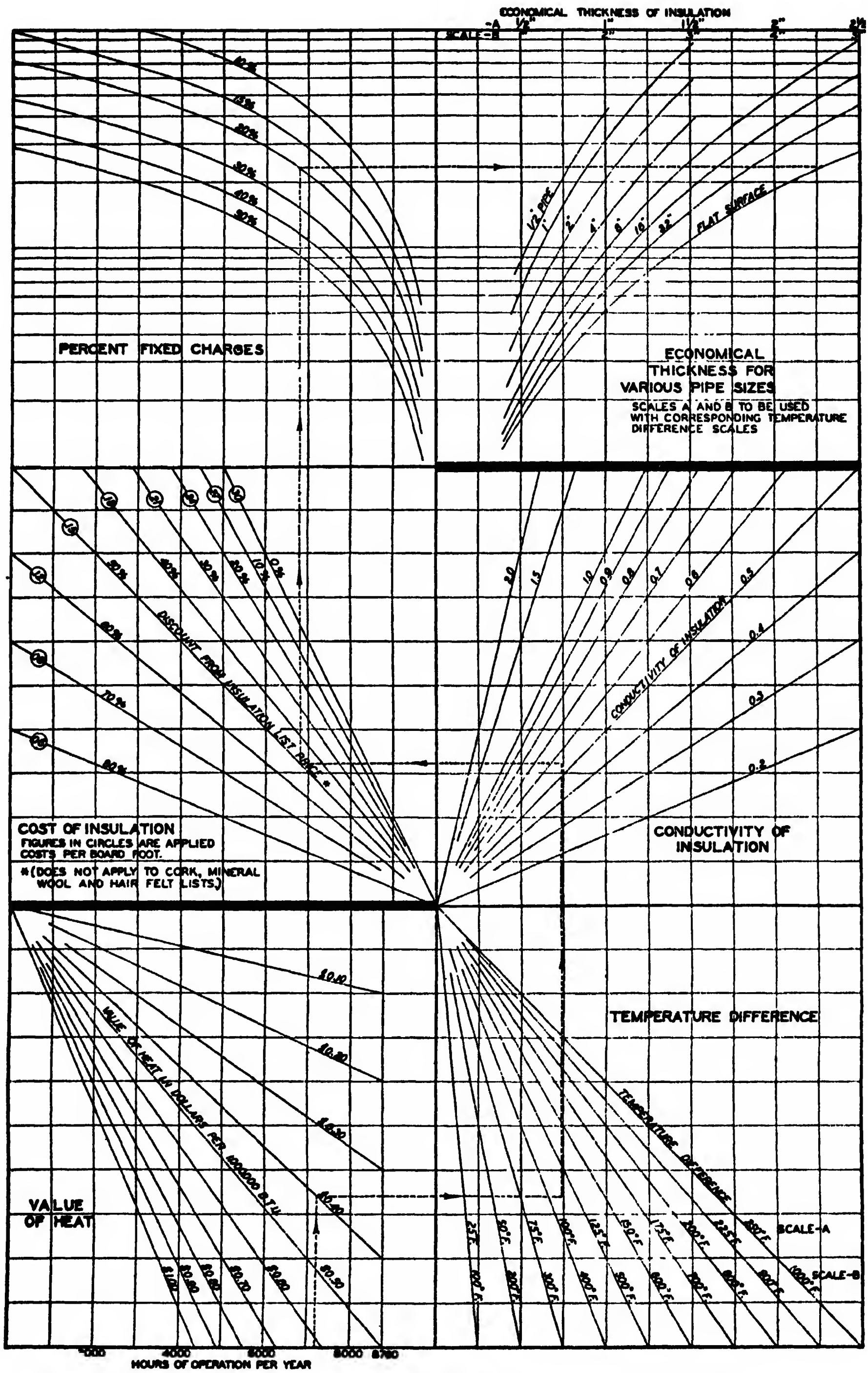
Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a] All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 10. Where a thorough analysis of economic thickness is desired, this may be accomplished through the use of the chart, Fig. 6.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe



(L. B. McMillan, *Proc. National District Heating Association*, Vol. 18, p. 138).

FIG. 6. CHART FOR DETERMINING ECONOMICAL THICKNESS OF PIPE INSULATION

size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

A rapid method for determining the economical thickness of insulation by use of tables has been published[4].

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes (See Chapter 29). Detailed data on commonly used forms of tunnels and conduit systems, have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency, and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. In some instances, where actual submersion of hot lines may occur, it has been found good practice to firmly secure the covering with corrosion resistant wire, then sew on a wire-inserted asbestos fabric jacket with wire. This jacket is porous. The principle of withstanding submersion is that water may enter as water, then actually boil at the pipe surfaces and escape as steam without rupturing the insulation or jacket. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

Table 11. Thickness of Loose Insulation for Use as Fill in Underground Conduit Systems

Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

The economical thickness of insulation for underground work is difficult to determine accurately due to the many variables which have to be considered. As a result of theories[3] previously developed, together with other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 11 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thick-

ness than that determined by the use of Fig. 6. The data in Fig. 6 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

REFERENCES

[1] Heat Loss from Copper Piping, by R. H. Heilman (*Heating, Piping and Air Conditioning*, September, 1933, p. 458).

[2] Handbook of the *National District Heating Association*, Second Edition, 1932.

[3] Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. TRANSACTIONS Vol. 26, 1920, p. 335).

[4] Rapid Method of Determining the Economical Thickness of Pipe Insulation, by Utley W. Smith (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October 1947, p. 118).

CHAPTER 29

DISTRICT HEATING

Steam Distribution Piping, Selection of Pipe Sizes, Conduits for Piping, Pipe Tunnels, Overhead Distribution, Inside Piping, Metering, Steam Requirements, Rates, Utilization, Automatic Temperature Control

THE term *district heating* refers to the heating of several buildings from a central plant as in the heating of portions of cities, or to the heating of groups of buildings as in institutions and factories. It is usually preferable, in a group of industrial or institutional buildings, that they be heated from a central plant rather than by individual plants. Fuel can generally be burned more efficiently, less labor is required, and often a central plant is cheaper to install. Those phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter since the local conditions control the layout little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per (hr) (sq ft of equivalent direct radiation) is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process should be individually calculated. The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, (3) pressure require-

ments of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 psi) are: (1) smaller heat loss per square foot of pipe surface, (2) less trouble with traps and valves, (3) simpler problems in pressure reduction at the buildings, and (4) general reduction in maintenance costs. With distribution pressures not exceeding 40 psi there is little danger even if the full distribution pres-

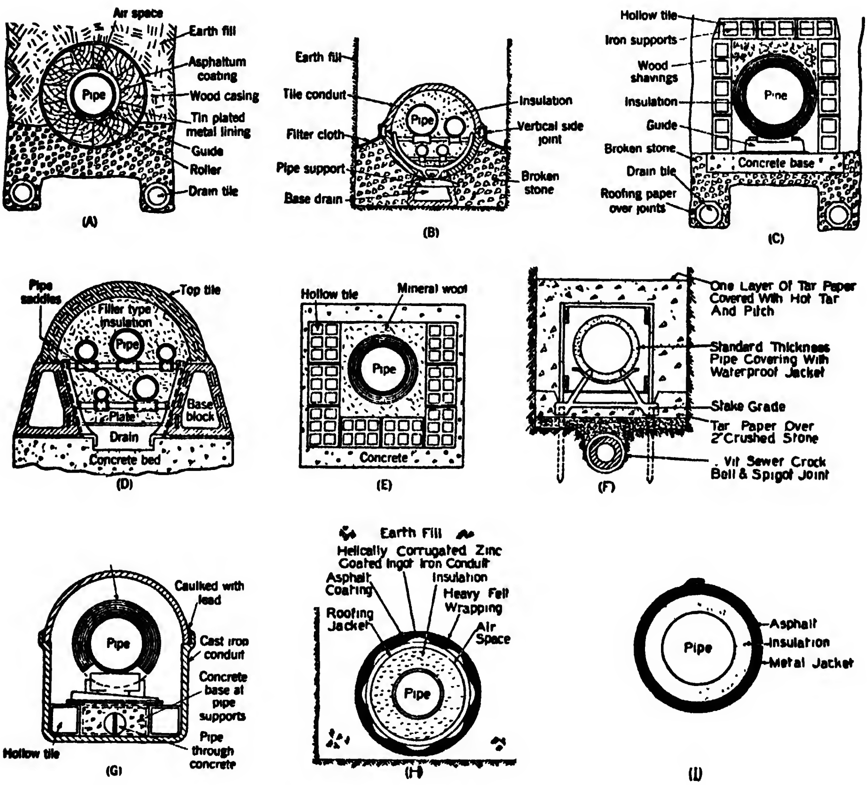


Fig. 1. Construction Details of Conduits Commonly Used

sure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than 50 psi a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are: (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating, (3) wider flexibility as to allowance for maximum pressure drop and ability to serve equipment requiring higher pressures.

Frequently the different kinds of apparatus which must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 psi, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, re-

sults in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 psi is usually demanded, although 30 psi is sufficient if the flat work ironer is equipped with a large number of rolls and if a slower rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 psi. Where pressures are not as high as desired higher pressures can be obtained by a steam compressor.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of Babcock's pressure drop formula:

$$P = 0.0000000367\left(1 + \frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

$$W = 5220\sqrt{\frac{PdD^5}{\left(1 + \frac{3.6}{D}\right)L}}$$

Numerical values of the various factors are given in Table 2, Chapter 23.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

Important points in laying out underground conduits are:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2. An expansion joint, offset, or bend should be placed between each two anchors. Advantage should be taken of the flexibility of piping to absorb expansion wherever possible. Information on provisions for expansion will be found in Chapter 27.

3. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic test pressure should be one and one-half times the maximum service pressure and it should be held for a period of at least two hours without evidence of leakage.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. Some of the more common forms are illustrated in Fig. 1.

The conduit (A) is of a wood casing construction which has been widely used in the past. The wood casing is segmented, lined with tin, and bound with wire. The outside of the conduit is coated with asphaltum. It is not suitable for high temperatures or poorly drained soils.

In Fig. 1 (B), (C), (D), (H) and (I) are patented forms of conduits. The insulation is sometimes a loose filler packed into the conduit. Con-

duits (H) and (I) are prefabricated. Both of these conduits are enclosed in metal jackets.

At (C) and (E) are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The addition of this loose insulating material to the sectional insulation is, of course, optional and is justified only where high pressure steam is used.

(E) and (F) are conduits used by two district heating companies, and have the advantage of being constructed of common materials.

Conduit (G) is of cast-iron construction, assembled with lead joints and

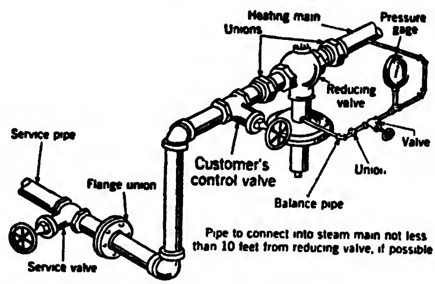


FIG. 2. CONNECTIONS FOR REDUCING VALVE WITHOUT BYPASS

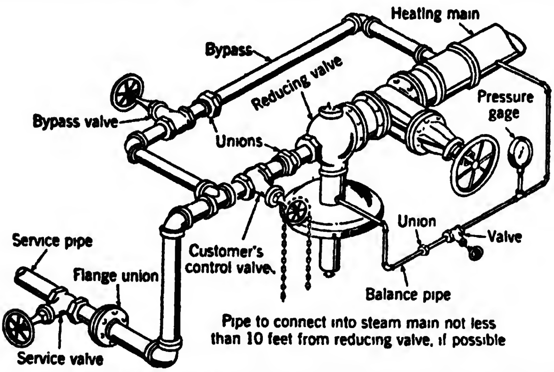


FIG. 3. CONNECTIONS FOR REDUCING VALVE WITH BYPASS

is water-tight, if properly laid. It is obviously expensive and is justified only in exceptional cases.

Since it is difficult to make a concrete or masonry conduit absolutely water-tight, provision should be made for some seepage. The pipe should be protected by a waterproof jacket over the insulation and the seepage drained from the inside of the conduit. Underdrainage of the conduit is generally provided for by a tile drain laid in crushed stone or gravel underneath the conduit. The tile underdrain should be carried to the sewer or some other drainage point. Manholes are required at intervals for access to valves, traps, and some types of expansion joints.

Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure.

Where it is possible to use basement or sub-sidewalk space for the distribution piping the cost of installation and maintenance is greatly reduced.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

Because of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes omitted along heating lines unless they are required to accommodate miscellaneous other services or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method of construction has the advantage of requiring no excavation and being easily maintained.

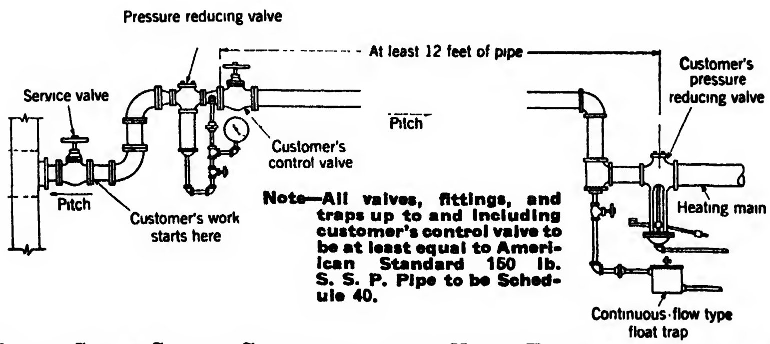


Fig. 4. Steam Supply Connection when Using Two Reducing Valves

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service.

Fig. 2 shows installation of a reducing valve without a bypass, which is usually omitted in the case of smaller size valves.

Fig. 3 illustrates the use of a reducing valve, with a bypass which is generally provided for larger installations. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service[1]. The first reducing valve effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

In a heating system the pipes carrying condensate are more subject to corrosion than other parts of the system. Care must be taken to give proper pitch to the pipes and provide proper venting of non-condensable gases. (See Chapter 51, Corrosion).

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests, but

principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing. It is necessary, however, to have an ample amount

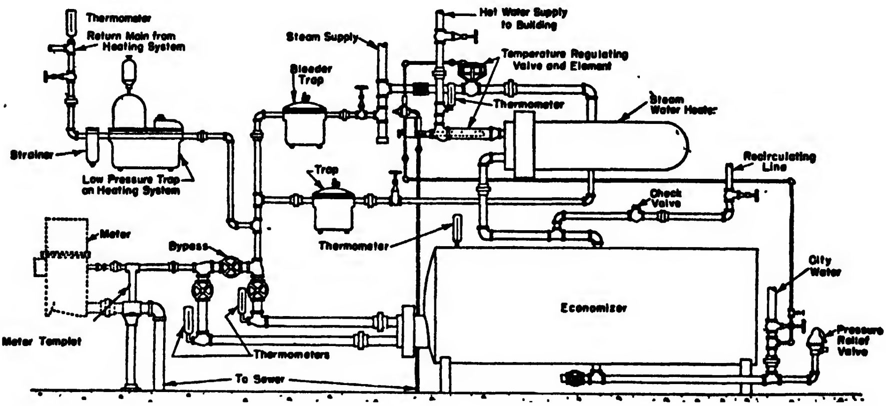


FIG. 5. METHOD OF INSTALLING A WATER HEATER AND ECONOMIZER IN A GRAVITY HEATING SYSTEM

of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building.

The condensate from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water heater passes through the economizer, absorbing heat from the condensate. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature. The maximum in economical operation and satisfactory heating can only be obtained by the use of some automatic temperature control system.

METERING

The perfection of fluid meters has contributed as much to the advancement of district heating as any other one thing. Meters are classified into two groups: *Condensate Meters* and *Rate of Flow Meters.*

Condensate Meters

The one type of quantity meter used is the condensate meter, which may be of the *tilting bucket* or *revolving drum* type.

The condensate meter is a popular type for use on small and medium

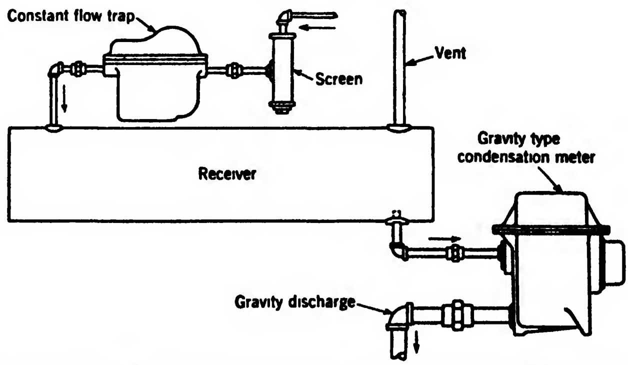


FIG. 6. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

sized installations, where all the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Condensate meters should not be operated under pressure; they are made for either gravity or vacuum installations. Where bucket traps are used, a vented receiver is essential ahead of the meter. Where continuous flow traps are used, a vented receiver is not necessary, but is desirable. Fig. 6 illustrates a gravity condensate meter installation using a vented receiver.

Rate of Flow or Flow Meters

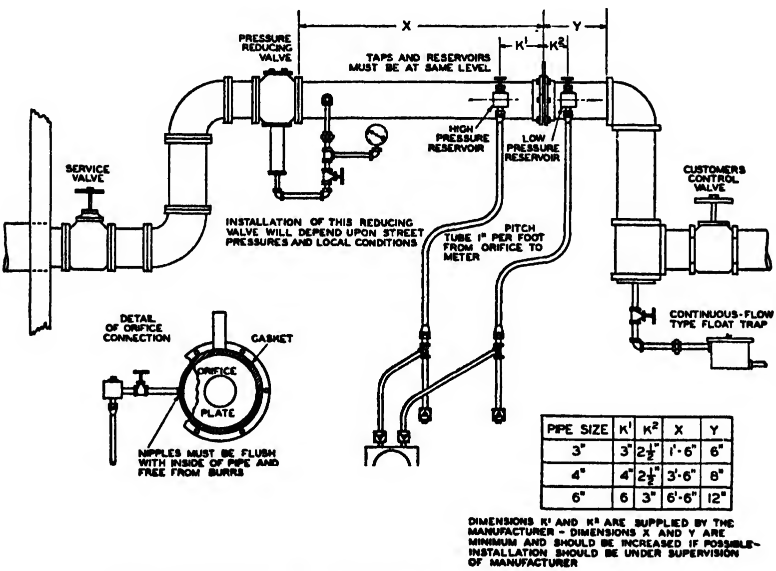
Flow meters used for district heating work are of three types: *Area Meters, Head Meters* and *Velocity Meters.* (See Chapter 4, Fluid Flow.)

Area meters are those, in the operation of which, a variation in the cross-section of stream under constant head is used as an indication of the rate of flow. A tapered plug is suspended in an orifice and moves axially with the flow, which is vertically upward. The weight of the plug provides a definite pressure differential and the plug floats at such a height as will provide enough orifice area to pass the flow at the pressure difference. The

position of the plug is transmitted by means of a lever and pencil and records the flow on a graduated strip chart.

Head meters are those in which the stream of fluid creates a difference of pressure, or differential head. This head is created by an orifice, Venturi tube, flow nozzle, or Pitot tube and will depend upon the velocity and density of the fluid. The secondary element must contain a differential pressure gage, which will translate the pressure difference into rate of flow or total flow. This mechanism may be either mechanical or electrical. The electric flow meter has the advantage of being able to locate the instruments at some distance from the primary element.

Fig. 7 is a typical example of an orifice-type meter installation. A few general points to be considered in installing a meter of this type are: (1)



PIPE SIZE	K¹	K²	X	Y
3"	3"	2½"	1'-6"	6"
4"	4"	2½"	3'-6"	8"
6"	6	3"	6'-6"	12"

FIG. 7. ORIFICE METER STEAM SUPPLY CONNECTION

It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available. (2) Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company. (3) The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above. (4) Meter piping should be kept free from leaks. (5) Sludge should not be permitted to collect in the meter body. (6) The meter body and meter piping should be kept above freezing temperatures. (7) It is best not to connect a meter body to more than one service. (8) Special instructions are furnished for metering a turbulent or pulsating flow.

Velocity meters are those in which the primary element is some device that is kept in continual rotation by the linear motion of the stream. The secondary element is, essentially, a revolution counter. The primary and secondary elements are combined into one unit.

For steam metering, the shunt meter is an example of the velocity type. This unit is connected directly in 2, 3 and 4 in. pipe lines. Larger size mains are metered by installing a 2 in. meter in a bypass with a restricting orifice in the main line.

Selection of Meter

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations: (1) Its use in a new or an old installation. (2) Method to be used in charging for the service. (3) Location of the meter. (4) Large or small quantity to be measured. (5) Temporary or permanent installation. (6) Cleanliness of the fluid to be measured. (7) Temperature of the fluid to be measured. (8) Accuracy expected. (9) Nature of flow: turbulent, pulsating, or steady. (10) Cost. *a.* purchase price, *b.* installation cost, *c.* calibration cost, *d.* maintenance cost. (11) Servicing facilities of the manufacturer. (12) Pressure at which fluid is to be metered. (13) Type of record desired as to indicating, recording or totalizing. (14) Stocking of repair parts. (15) Use of open jets where steam is to be metered. (16) Metering to be done by one meter or by a combination of meters. (17) Use as a check meter. (18) Its facilities for determining or recording information other than flow. (19) Whether or not the condensate can be returned to a central point.

STEAM REQUIREMENTS

Methods of estimating steam requirements for heating various types of buildings are given in Chapter 20.

Table 6 in Chapter 20 represents information obtained from all sections of the United States, and the group of buildings from which the information was taken represents a cross-section of all types of heating systems.

Steam requirements for water heating can be satisfactorily estimated by using a consumption of 0.0025 lb per (day) (cu ft of heated space) for office buildings, without restaurants, and 0.0065 lb per (day) (cu ft of heated space) for apartment buildings.

Complete information on water heating requirements is given in Chapter 50.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the *Handbook of the National District Heating Association*.

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not always conform to this principle. Lastly, the rate schedule must be reasonably simple and understandable.

Glossary of Rate Terms

Load Factor. The ratio, in per cent, of the average hourly load to the maximum hourly load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface, or required radiator surface, and the demand of the particular installation. It varies from 0.25 to 0.3 lb per (hr) (sq ft of surface).

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

The various types of rates to be found in use in district heating systems are:

1. *Straight-Line Meter Rate.* The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

2. *Block Meter Rate.* The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefited at the expense of the others.

3. *Demand Rates.* These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per thousand pounds of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(*a*) A charge based upon the demand, either estimated or measured.

(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

The *Doherty rate* is divided into three elements:

(*a*) A charge based upon demand.

(*b*) A charge based upon steam consumed.

(*c*) A customer charge.

In the *Hopkinson rate,* the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand. They are of benefit to utility companies and to consumers because the investment and operating costs can be divided, to suit the particular circumstances, into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per thousand pounds of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

Some utility companies include a labor surcharge as well as a coal surcharge.

UTILIZATION

Considerable savings can be made by the proper and intelligent operation of heating systems. It should be borne in mind that a heating system is

designed to heat a building to 70 F inside when the outside temperature is at its lowest point for that particular locality. There is a tendency to overheat the building at any time the outside temperature is above the design temperature unless some method of regulation is used, either automatic or manual.

The general rules for economical operation[2] are as follows:

1. *Reduce the heat losses from the building to a minimum.*
   a. Weatherstrip all windows, and caulk all window frames.
   b. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms from the remainder of the building by partitions so that the large doors will not ventilate the entire building.
   c. Eliminate all unnecessary ventilation. Ventilating equipment is usually sized to meet extreme requirements. In a theater or auditorium, do not supply enough ventilation for an audience of 2000 when there are only 200 present.
2. *Limit the hours of heating to those in which the required temperature is necessary.*
   a. Determine the hours that heating is required and see that steam is shut off for the maximum time when not required, such as nights, Sundays, and holidays.
   b. Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing plumbing.
   c. Install separate lines for those parts of the building that require long-hour or 24-hour heating. This is much cheaper than heating the entire building.
   d. Control the heat supplied to water storage tanks located on or above the roof. Such tanks require heat to prevent freezing when the outdoor temperature is below 32 F.
3. *Regulate the amount of heat so as to prevent overheating and to maintain uniform temperatures during the hours of occupancy.*
   a. Determine the temperature required for the occupancy of a building. Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.
   b. Shut off steam during the day whenever possible. An automatic control will do this, but it can be done by hand, with good results.
   c. Provide some good means of temperature control.
4. *See that the heat input is properly balanced throughout the building.*
   a. See that the entire heating system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.
   b. Place the radiation near the outside walls under the windows or where the exposure occurs, if possible.
   c. Do not obstruct radiators or prevent the free circulation of air around them; to do so seriously reduces the heating capacity of a radiator.
5. *Keep all heating equipment in first class condition.*
   a. Keep the system in good repair. This applies to all traps, valves, vents, steam and return piping, vacuum pumps, and temperature control apparatus.
   b. In a vacuum system, maintain the degree of vacuum recommended by the control manufacturer. If this is not possible, locate and eliminate all leaks.
   c. Insulate all steam pipes not used as heating surface.
6. *Arrange the heating system to obtain from it the highest possible efficiency.*
   a. Locate all valves and controls so as to be convenient and accessible. It is only human nature to delay or avoid doing that which is unnecessarily inconvenient.
   b. Investigate every complaint of "No Heat"; find the cause and correct it. Do not overheat an entire building to correct a local condition.

c. Extract the heat in the condensate for heating water or for some other useful purpose.

7. *Make a study of the heating system and heating requirements.*
   a. Provide thermometers and recording pressure gages so that the heating system may be operated with full knowledge of what is being accomplished.
   b. Keep daily consumption records and check against the theoretical requirements.
   c. Study the system and understand its functions and its operations.

AUTOMATIC TEMPERATURE CONTROL

As stated in Chapter 34, Automatic Control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. The use of adequate temperature control provides more healthful, comfortable, and efficient working conditions in buildings.

There are three general means of obtaining centralized control of heat output of radiators.

1. *Controlling the rate of steam flow* into the radiators. This is accomplished by equipping the radiator inlets with orifices and controlling the flow of steam through them into the radiator by controlling the difference in pressure between the supply and return.

2. *Controlling the temperature of steam* in the radiators by varying its pressure. This involves the use of high vacuums to obtain low steam temperatures. This must be supplemented by some other type of control for low heat output.

3. *Controlling the length of time steam flows* into the radiators by admitting steam to a heating system intermittently and varying the length of the *on* and *off* periods. Two types of controls are used. (1) A clock control providing *on* and *off* settings of various lengths, which can be changed in accordance with outside temperatures. In most cases these changes are made automatically by means of a thermostatic bulb, placed outdoors. (2) A control, having an outdoor bulb and a bulb attached to the radiator, which varies the length and frequency of the *on* intervals in such a way that the radiator temperature is varied according to the outside temperature. In some cases heat supply is controlled by combinations of the three methods described.

Before installing any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Because of this depreciation of the system, operation becomes more and more costly and parts of the building have to be greatly overheated in order to prevent underheating in other parts. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired if required. The piping should be of adequate size and graded properly. The return piping should be inspected, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly and may prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems and offer recommendations in regard to operation and changes in piping layout.

The selection of control equipment depends upon the type and size of building and the degree of saving which may be obtainable.

REFERENCES

[1] Code for Pressure Piping, B31-1, 1942, *American Standards Association*, Paragraph 408, p. 115.

[2] Principles of Economical Heating, *National Association of Building Owners and Managers*.

CHAPTER 30

ELECTRIC HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Boilers, Electric Hot Water Heating, Heating Domestic Water Supply, Control, Calculating Capacities, Radiant Drying, Induction and Dielectric Heating, Power Problems

ELECTRIC heating deals with the conversion of electrical energy into heat and the distribution and practical use of the heat so produced. In certain regions, where the cost of electricity is favorable, electric heating is used extensively. Its use is also frequently dictated by special conditions.

Definitions of the terms *Electric Resistor*, *Electric Heating Element*, *Electric Heater* and other terms applying to heating practice, will be found in Chapter 1.

RESISTORS AND HEATING ELEMENTS

Commercial electric heating elements usually have solid resistors such as metal alloys or non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by passing an alternating electrical current through it.

In one type of heating element, the resistors are exposed coils of nickel-chrominum wire or ribbon, or non-metallic rods, mounted on insulators. This type is used extensively for operation at high temperatures for radiant heat or at low temperatures for convection and fan circulation heating.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, in some types of convection air heaters and in low temperature radiant heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters.

Cloth fabrics woven from flexible resistor wires and asbestos thread are used for many low temperature purposes such as heating pads, aviators' clothing and radiant panel heating installations.

Special incandescent lamps are used as heating elements in certain applications where radiant heat is desired. These use carbon or tungsten filaments as resistors, and are designed to produce maximum energy in the infra-red portion of the spectrum.

ELECTRIC HEATERS

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such appliations as aviators' clothing, hot pads, soil heaters, and water heaters. Conduction

heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have heating elements and reflectors to concentrate the heat rays in the desired directions. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. For a discussion of electrically heated panels as applied to radiant heating, see Chapter 31.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water

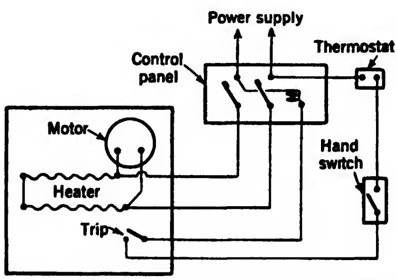


Fig. 1. Wiring Diagram for Unit Heater

and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

Induction and dielectric heaters are described in a later section.

UNIT HEATERS

Electric unit heaters include a built-in fan unit which circulates room air over heating elements. They are adapted to the same uses as other types of unit heaters where conditions are favorable to electric heating. They are very adaptable for heating of small offices, locker rooms, etc., in otherwise unheated buildings. In small unattended equipment rooms, thermostatically controlled electric unit heaters are frequently used to maintain a temperature above freezing.

The best location for electric unit heaters depends upon local conditions. Various designs and arrangements are available, as with steam unit heaters. See Chapter 26.

The arrangement of the wiring circuits is very important. In principle, they are all the same and include as essential elements, a magnetic control contactor, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnetic contactor and with the hand switch and thermostat. A typical wiring diagram for single phase power supply is shown in Fig. 1. A main disconnect switch should be provided. For three phase power supply, a 3-pole contactor should be used and the heater arranged for 3-phase connection. On larger sizes, separate over-load protection for the motor should be provided.

If the line voltage is more than 120 to ground, it is advisable to supply the thermostatic control circuit through a transformer.

CENTRAL HEATING AND AIR CONDITIONING

Electric heating elements can be used for the prime source of heat in a central fan heating system or in the heating phase of an air conditioning system. They can be used in the same manner as steam heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning, electric heating elements can be used to provide moisture by the evaporation of water.

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature

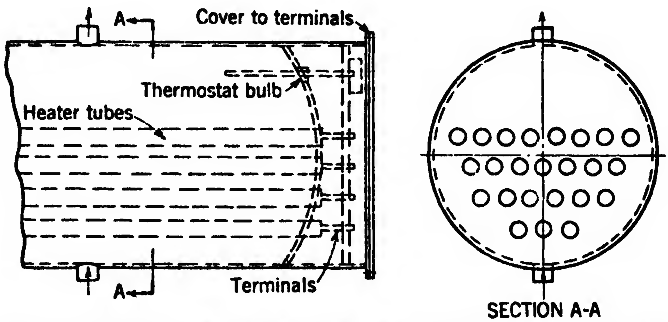


Fig. 2. Resistance Type Boiler for Steam or Hot Water

source of heat for any given pressure and a change in air volume flowing over steam coils does not greatly affect the temperatures of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered. This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic variation of the electrical heat input synchronized properly with the air flow can be successfully accomplished by various special methods of control. By-pass dampers as used with steam units will not control electric heat.

Electric heaters are useful in balancing the heat distribution in central fan systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermostats

mounted in each individual room to permit the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC BOILERS

Steam or hot water generating boilers using electrical energy are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used either with direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood of burning out of heating elements they should be of substantial construction, with a low heat density per unit of surface area and provision should be made for

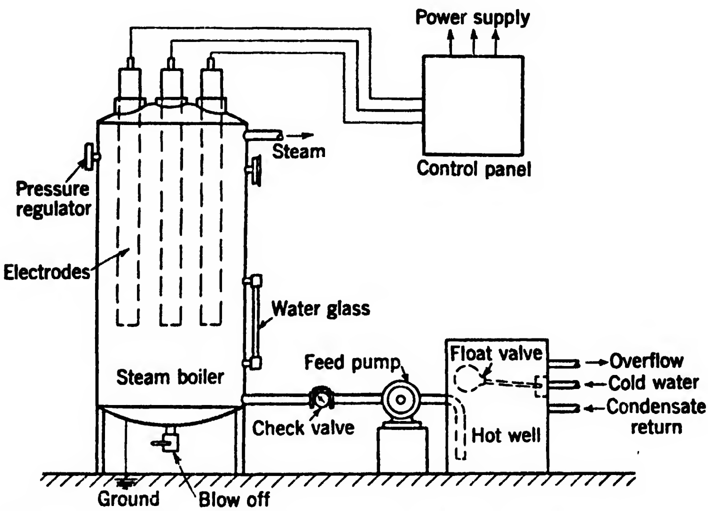


FIG. 3. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

cleaning off deposits of scale which restrict the heat flow. A typical resistance type of steam or hot water boiler is shown in Fig. 2.

Large electric boilers are usually of the type employing water as the resistor, using immersed electrodes. With this type only alternating current can be used, as direct current would cause electrolytic deterioration. Such a type of electrode boiler is shown in Fig. 3.

Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes and for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. The use of insulated water storage tanks, in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large,

well-insulated, pressure type steel tank, equipped with electric heating elements and automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard design. A system of this kind requires very careful design to avoid excessive over-all radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical water heating boiler is illustrated in Fig. 2.

HEATING DOMESTIC WATER BY ELECTRICITY

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable. In many sections of the country low electric rates have been established by the electric utilities to secure this load. In many localities, electric rate schedules divide the current

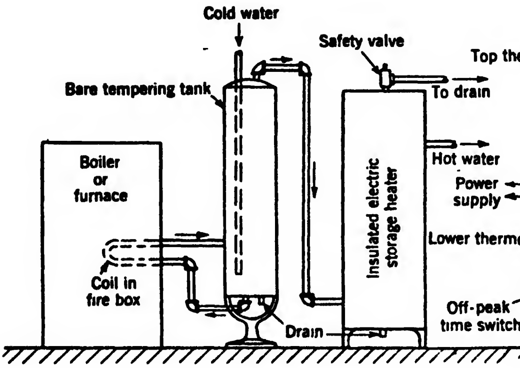


Fig. 4. Piping Arrangement for Connecting Electric Water Heater to Fire-Box Coil

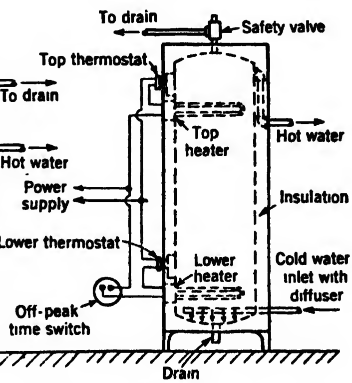


Fig. 5. Domestic Hot Water Heater for Off-Peak Service

used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to help carry over the periods when the off-peak element is timed out. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all electric service if an electric water heater is installed.

Competition with other fuels, especially gas, seems to be the major controlling factor in the use of electricity. The first cost of electric storage heaters is greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

In residential work, to effect a saving in the cost of operation, it is sometimes desirable to use a furnace coil or indirect heater in connection with an electric water heater. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 4, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to

the electric heater serves as a tempering tank, absorbing heat from the basement air and requiring the use of less energy in the electric heater.

A typical domestic hot water heater as shown in Fig. 5 is arranged with upper and lower heating elements for the usual type of off-peak heating service. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that, in case the supply of hot water in the tank becomes exhausted, the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

CALCULATING CAPACITIES

In calculating electric heating capacity one kilowatt is equal to 3413 Btu per hour or 14.2 sq ft equivalent direct steam radiation.

All of the energy applied to an electric resistor is transformed into heat. The output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and the total load on an electric heating system is the total wattage of the connected electric heaters.

RADIANT DRYING

Lacquers and similar surface films can be very effectively dried by radiation. Special electric lamp units have been developed which give off a high percentage of infra-red and similar heat rays. For continuous manufacturing processes these units are mounted in tunnels through which conveyors pass. For local applications, as for example paint drying in automobile repair shops, they may be mounted on portable racks. Objects of relatively large surface area in proportion to their weight, and fabricated materials having a rather high heat absorption, may be satisfactorily heated by such a source.

For drying, baking, pre-heating and de-hydrating, where a low temperature infra-red heat source is desired, or where the use of glass-enclosed radiant lamps is objectionable for safety reasons, electric heating units employing low temperature metal sheathed resistors, are available.

ELECTRIC HEATING BY INDUCTION AND DIELECTRIC MEANS

These methods differ radically from resistance heating. They have many important industrial uses and open up a whole new field of special application where extreme speed or control of heat location are vital.

Metals and other electrical conductors can be heated by induction. The work is placed in an alternating magnetic field within, or adjacent to, a coil, and heat is produced in the body of the piece by eddy currents. While induction heating has certain limitations, it has great advantages in certain applications such as melting metals, forging, brazing, heat treating and particularly for localized heating and zonal hardening of metals. It is possible to apply localized heat so rapidly that conduction cannot draw the heat away before it has time to accomplish the desired purpose at a particular spot. Surfaces and local areas can be hardened without distortion or scale formation.

Commercial 60-cycle alternating current may be used in special cases, such as induction heating of large pressure vessels, but generally special higher frequency generating equipment is required. This should be carefully selected for the particular kind of work to be done. Motor-

generators with frequencies in the vicinity of 250 cycles per second, are used for many melting furnaces. Motor-generators having frequencies between 2,000 and 10,000 cycles per second, are generally used for heat treating and hardening sizeable parts. For heating or brazing thin sections or small parts, electronic tube oscillators, spark discharge oscillators, or mercury arcs are used to produce frequencies ranging up to 500,000 cycles per second. Work coils used with high frequency induction heating are generally copper tubes through which cooling water is circulated. These must be specially designed for each application.

Non-conductors of electricity can be heated internally by dielectric means by placing the materials in a high frequency electrostatic field between electrode plates. This process is distinctly different from the induction heating process. High voltages and very high frequencies, often up to 50 million cycles, are needed to produce the desired rate of heating. The main field for dielectric heating is with materials which are poor thermal conductors. Food can be sterilized, plywoods bonded, plastics heated, granular or crystalline material dehydrated, deep-pile fabrics dried, and countless other products heated quickly and uniformly. Dielectric heating is well suited to many continuous production processes, as the materials can pass through the heating field quickly and without the necessity of contact with the electrode surfaces.

POWER PROBLEMS

The cost of electric energy varies because of several factors. Distribution costs differ for large and small users. The fact that electricity cannot be economically stored, but must be used as fast as generated, makes it impossible to operate electric plants at uniform loads; hence, even the time of use may affect the cost of electricity. Special low rates are sometimes available during certain prescribed hours of use.

Since cost of production and distribution depends not only upon the quantity of energy used but also upon the maximum rate of use, electric energy is often sold on a demand rate basis. In some cases, the demand charge is based upon the rated connected load; in other cases, upon the maximum demand indicated by a demand meter.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually the service lines will not permit more than plug-in devices. The *National Board of Fire Underwriters* permits approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles, but such heaters cannot be served on a circuit supplying much other load without overloading the circuits. There is an increasing trend toward supplying homes with three wire 115–230 volt service. Where homes have such service, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 208, 220, 440 or 550 volts. All polyphase heaters should be balanced between phases. In ordering electric heaters, proper voltage must be specified, as the heat produced will vary as the square of any variation in voltage.

BIBLIOGRAPHY

Electric Elements Well Adapted to the Air Conditioning Heating Cycle, by L. P. Hynes (*Heating, Piping and Air Conditioning*, January, 1940, p. 29).

Electric Heating for Los Angeles Building (*Heating and Ventilating*, 37:50–1; June, 1940).

House Heating Load Characteristics as They Affect Wiring Costs, by Jack B. Cochran (*Electrical World*, April 12, 1947).

Low-Voltage, High-Current Radiant House Heating (*Electrical World*, January 3, 1948, p. 39).

Radiant Heating by Electricity, by L. N. Roberson (*Heating and Ventilating*, September 1946, p. 89).

Radiant Heating, by F. M. Tiller (*Chemical Products*, March-April 1945).

Applications of Radiant Energy (Lighting Handbook, *Illuminating Engineering Society*, 1947, Section 16).

Industrial Applications of Infra-red, by J. D. Hall, (McGraw-Hill Book Co., 1947).

Faster Baking and Drying with Infra-red Heat, by Paul H. Goodell (*Electrical World*, June 21 and July 5, 1947).

Electric Radiant Heat vs. Steam Convection, by Douglas Dow (*Electrical World*, August 10, 1940, p. 61–62).

Infra-red vs. Convection Ovens for Drying Paint Coatings, by F. J. Gschwind (*Industrial Finishing*, September 1945).

Recent Advancements in Industrial Heating Processes and Equipment, by Eugene Mittelman (*Industrial Heating*, September 1944).

Heating of Non-magnetic Electric Conductors by Magnetic Induction, by R. M. Baker (*Electrical Engineering*, June 1944).

Operating Experience with HF Heating, by Henderson C. Gillespie (*Elcctronic Industries*, February 1944).

Unusual Methods of Applying Electronic Heat, by E. D. Tillson (*Electronics*, April 1945).

Dielectric Heating, by T. W. Dakin and R. W. Auxier (*Industrial and Engineering Chemistry*, March 1945).

Induction and Dielectric Heating, by Kennard Pinder (*Elcctrical Enginecring*, February 1947).

High Frequency Heating in the Radio Spectrum by W. C. Rudd (*Electrical Engineering*, June 1947).

Standard Handbook for Electrical Engineers, McGraw Hill Book Co.

CHAPTER 31

PANEL HEATING AND RADIANT HEATING

Influence of Heat Radiation on Human Comfort, Objectives of Radiant Heating, Practical Problems of Radiant Heating from a Physiological Standpoint, Fundamental Computations, Application Methods, Calculation Principles, Measurement and Control

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IT has been pointed out in Chapter 12 that the human body loses heat to its environment in three ways; by convection, radiation, and evaporation. The Effective Temperature Chart takes account of convection and evaporation, but does not provide for such radiative effects as occur when room air and its surrounding surfaces differ widely in temperature.

INFLUENCE OF HEAT RADIATION ON HUMAN COMFORT

When, however, the body is exposed to radiation from a hot surface or is radiating to a cold surface, the factor of radiative heat gain or heat loss may be important. This phenomenon is most marked in the case of exposure to the sun's radiative heat. On a cold day, with no wind blowing, while standing in the sunshine, one may feel perfectly comfortable but, when a cloud passes over the sun, one may instantly feel much cooler. The cloud acts as a shield to interrupt the radiant heat from the sun. The change in feeling of comfort is due to the instant change in rate of heat loss from the body caused by the shielding effect of the cloud. A shielded thermometer under the same condition would register no change in temperature.

The rate of heat loss by convection depends upon the average temperature difference between the surface of the body and the surrounding air, the shape and size of the body, and the rate of air motion over the body.

The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. This latter temperature is called the Mean Radiant Temperature (MRT).

Because these two types of heat loss supplement each other, a required rate of total heat loss can result either from a relatively low air temperature and a relatively high MRT, or vice versa.

At the temperature which produces comfort (and at all lower temperatures) the production of sweat is low and the heat loss by evaporation is relatively low and relatively constant, irrespective of the relative humidity of the atmosphere. Under such conditions the heat loss from the body is chiefly related to the combined effect of convection and radiation. For unclothed subjects in a reclining posture the heat demand of the environment, so far as these two factors of radiation and convection are concerned, may be measured by Operative Temperature, whichis defined by the following formula, modified from that of Gagge[1] by the expression of air velocity in feet per minute and the various temperatures in Fahrenheit degrees,

$$t_o = 0.81\, t_w + 0.135\, [\sqrt{V}\, t_A - (\sqrt{V} - 1.40)\, t_s],$$

where

t_o = operative temperature, Fahrenheit degrees.
t_w = mean radiant temperature, Fahrenheit degrees.
t_A = air temperature, Fahrenheit degrees.
t_s = mean skin temperature, Fahrenheit degrees.
V = air velocity in feet per minute.

Under comfortable still air conditions during the heating season, the mean skin temperature of persons normally clothed is between 90 and 93 F (with lower values for the extremities), and the mean clothing surface temperature is between 82 and 86 F.

The normal rate of heat production in an average sized sedentary individual is about 400 Btu per hour. The heat production for persons subjected to various rates of activity is given in Chapters 12 and 15. The human body is of complicated shape, and radiation takes place freely only from the exposed outer surfaces; there are considerable portions of the body such as the legs, arms, lower part of the head, etc., which radiate most of their heat to other portions.

It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total may be assumed to be about 19.5 sq ft for convection and 15.5 sq ft for radiation, in an average sized individual.

The loss by respiration and by evaporation from the nose and throat depends on the temperature and area of the moist surfaces (respiratory) of the body, the air temperature, air movement, and humidity. In air at a temperature of 70 F, this loss, for a sedentary individual of average size, will be approximately 90 Btu per hour; and at 60 F about 70 Btu per hour. These values are relative, because the total will vary materially with change of position, bodily activity, age, sex, race, etc.

The balance of the heat generated in the average human body, approximately 300 to 320 Btu per hour at about 70 F room temperature, is the approximate amount of heat given off by radiation and by convection from the external body surfaces. Under normal conditions (in still air), the radiation loss will be about 190 Btu per hour; and the convection loss about 120 Btu per hour. With an air velocity of 520 fpm, comfort will require an increase in Operative Temperature of nearly 12 deg; under such conditions the convection loss will rise to 250 Btu per hour but comfort may be attained if the subject is surrounded by heated walls which keep the radiation loss at about 50 Btu[2].

It is neither feasible nor desirable to change the relationships of convection and radiation very greatly in actual heating practice. In the laboratory, where the laws of radiative heat loss have been deduced, it is necessary to produce wide differences between radiative and convective heat loss. This can only be accomplished, however, by elaborate and powerful conditioning apparatus which simultaneously heats walls and cools air, or vice versa. Such a process would be very costly in practice and would not be justified unless marked improvement in comfort resulted from such a condition—an assumption which has not been demonstrated. In practice, where radiant heat is introduced into a room, it is absorbed by surfaces, furniture, and the like, and then transformed into convective heat so that air and surfaces tend to attain a generally uniform temperature.

OBJECTIVES OF RADIANT HEATING

Under ordinary circumstances the human being, indoors, is not subjected to marked variations between the factors affecting convection and radiation. Air and walls are not commonly very far apart in temperature; air movement and relative humidity are usually low. Where such conditions obtain, the ordinary air thermometer is a good measure of comfort—which is the reason why it has enjoyed such universal use. Where considerable window surfaces create heavy radiation loss, or where stoves or open fires, or very hot ceilings contribute to large radiation gain, the picture is changed and the air temperature productive of comfort must be correspondingly modified.

In general, however, radiant heating of occupied spaces is not a procedure designed to create differences between air and walls, but is merely one method of introducing heat into that space. The engineering factors used in determining desirable heat input will be essentially the same as if the heat were introduced by convection, or in any other way.

PRACTICAL PROBLEMS OF RADIANT HEATING FROM A PHYSIOLOGICAL STANDPOINT

It is convenient to distinguish two different methods of introducing radiant heat into an enclosed space. The first, which may be called High-Temperature Radiation, involves direct exposure of the occupied parts of the room to radiation emitted from relatively small heating units of very high temperatures (perhaps 1,000 F); the second, Panel Heating, involves exposure to relatively large surfaces at not over 130 F.

High-temperature radiant heating may be useful for temporary purposes, as in the use of a bathroom heater. It is, however, generally an undesirable process (except in rooms of great height) because of the marked unevenness of the effect produced on the human body. Studies at the John B. Pierce Laboratory of Hygiene have shown that this type of heating produces uncomfortable differences in the temperature of different parts of the body (an over-heated head, for example, if the heat comes from the ceiling).

Panel heating, on the other hand, is advantageous from the standpoint of temperature differentials. In actual practice, a well-designed system of this sort produces very uniform conditions, the air throughout the room differing at various points by only 5 deg. This is desirable from the comfort standpoint and may also be a factor in heat economy, since high temperatures in the upper part of the room favor excessive heat loss. The esthetic value of such a system is also considerable, since it avoids the presence of registers or free-standing radiators in the room.

In the design of panel heating, however, careful thought must be given to the location of the panels from the standpoint of comfort. The English commonly use the ceiling for their panels, but their rooms are generally high-studded, and outdoor winter temperatures moderate. With low ceilings even panels may produce an excessive directional heating effect if all the heat necessary in a cold climate is introduced from above. Similarly, if the floor alone is used, it may—in very cold weather—be necessary to make the floor too hot for comfort. Wall panels, or a combination of ceiling and floor panels, will perhaps produce the best results.

FUNDAMENTAL COMPUTATIONS

The mean surface temperature of an inert body, which will cause given rates of heat loss by radiation and by convection in a uniform environment,

having a given air temperature and a given mean wall temperature, may be calculated from fundamental equations[3] for radiation and natural convection, with substitution of comparable cylinders for the irregular human body.

$$q_r = 0.1730e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right] \quad (1)$$

$$q_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times (T_s - T_a)^{1.266} \quad (2)$$

where

q_r = heat loss by radiation, Btu per (square foot) (hour.)
q_c = heat loss by convection, Btu per (square foot) (hour.)
T_s = absolute temperature of the body surface, Fahrenheit degrees
T_w = absolute temperature of the walls, Fahrenheit degrees.
T_a = absolute temperature of the air, Fahrenheit degrees.
$T_m = \dfrac{T_s + T_a}{2}$
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission.

If it is assumed that an average adult has a height of 5 ft 8 in., a body surface of 19.5 sq ft for convection, and 15.5 sq ft for radiation, an equivalent effect can be worked out for two cylinders, 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter, respectively. However, while the effects on a cylinder, of a particular size and shape may be used to estimate average similar effects on the human body, it should be remembered that the heat loss from the body varies greatly. Every movement alters not only its shape, but also the heat generated by the body, the velocity of the air passing over it and the surface exposed to radiation. This fact renders the results of any such computation only approximate.

APPLICATION METHODS

The several methods of applying panel and radiant heating to a structure are:

1. *By warming the interior wall and ceiling surfaces of the building.* Pipe coils are imbedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe ½ or ¾ in. I.D. and spaced about 6 to 9 in. apart. See Fig. 1. This has the effect of warming the entire concrete or plaster surface in which the pipes are imbedded. Since the temperature of the heating medium should never exceed about 130 F, due to the possibility of cracking the plaster the area of the warmed surface must be sufficient to supply the requisite quantity of heat at this low temperature. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All coils and circulating pipes are welded together and tested after erection to a hydraulic pressure of 300 psi.

2. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in Fig. 2. This method was used 2000 years ago in many parts of the Roman Empire. While this method is more expensive in construction, it is effective and quite suitable for cathedrals and large public buildings. To provide a uniform floor temperature, special consideration should be given to the design of the air ducts so that equal heat distribution is obtained.

3. *By placing hot water pipes in or under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. Floor heating is recommended for schools and hospitals where large quantities

of outside air are desirable. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat, and while it is true that heat will be conducted through all materials used in floor construction, it is important that due consideration be given to the emissivity of the floor. In some cases where pipe coils are installed in the air space under the floor, special floors are constructed in sections so that the whole floor can be lifted to examine the coils. See Fig. 3. Pipes supported thus may be larger and the heating medium maintained at a higher temperature than

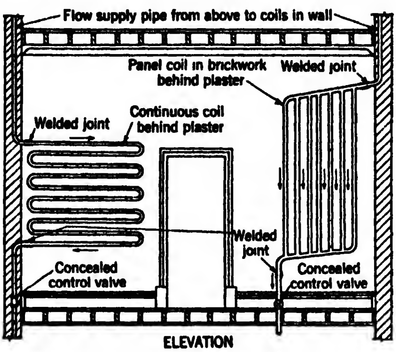


Fig. 1. Coils in Wall Surfaces

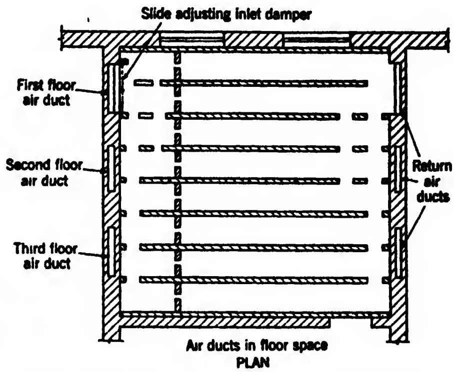


Fig. 2. Air Ducts for Floor Heating

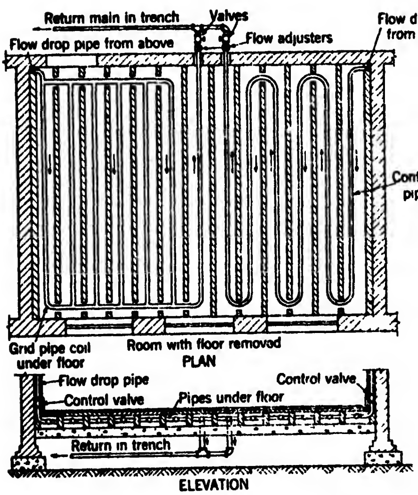


Fig. 3. Continuous Coil in Floor

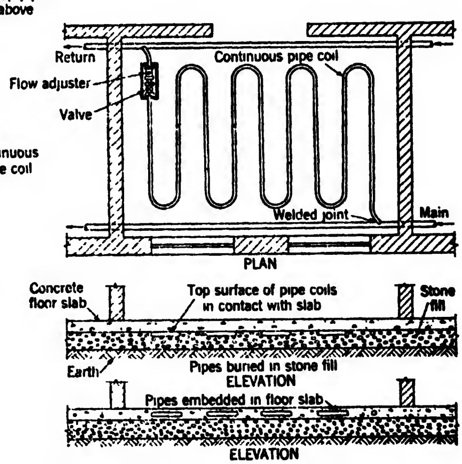


Fig. 4. Coils Imbedded in Floors

when pipes are actually imbedded in the floor. Pipes may be $1\frac{1}{2}$ or 2 in. in the former, but for the latter $\frac{3}{4}$ or 1 in. pipes are recommended. See Fig. 4. Where the heat losses from a room are exceptionally high it may be necessary to supplement the warm floor by either adding some coils in the ceiling or forming heated panels in the side walls.

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other parts of the room, or they may be cast into panels to imitate oak or other wood designs. With flat plate panels it is common practice to use a frame of plaster, wood, metal or composition to allow for expansion. These plates may be heated with either

hot water or steam and connected as in an ordinary radiator system. See Figs. 5 and 6.

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the construction, as found desirable. They should not have a surface temperature much above 200 F. Some have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall.* For this purpose the screen is woven with an electric continuous conductor. Such screens are useful to plug in at any position for emergency local heating without taking care of a large room or office.

If all of a heating panel is installed at one end of a large room there may be a marked difference between the equivalent temperature on the two sides

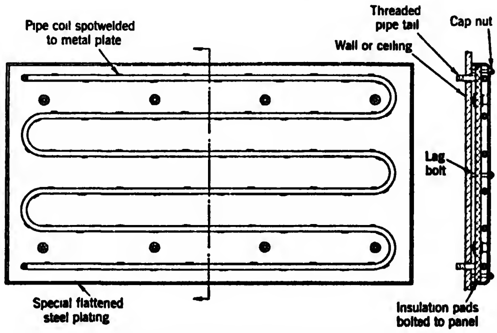


FIG. 5. PILLAR TYPE RADIANT HEAT PANEL

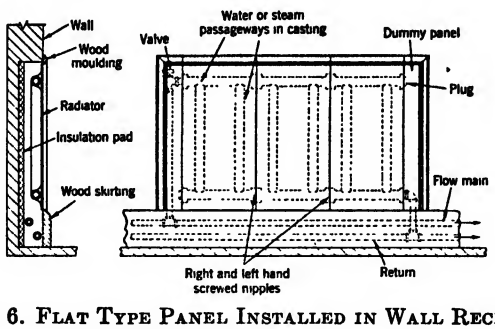


FIG. 6. FLAT TYPE PANEL INSTALLED IN WALL RECESS

of the body. It is usually desirable, therefore, that the heat be distributed at different parts of the walls and ceilings so that no uncomfortable effect will be felt from unequal heating.

CALCULATION PRINCIPLES

Part I—Panel Heating

The term *panel heating*, involving both radiation and convection, is applied in this chapter to a system in which the heat is transmitted from panel surfaces to both air and surrounding surfaces, as is the case under indoor conditions.

Panel heating systems for buildings may be designed as illustrated and described in the following design of a panel heating system for the room shown in Fig. 7. The design is based on continuous heating. Panel heating systems should, in general, be operated continuously since the panels

have large thermal capacities. Where panels are heated by high temperature radiation from a heat source directed toward them, this qualification does not apply.

1. *Assume the location and the approximate size of the heating panel*

Heating panels may be located in ceilings or floors or walls. Ceiling panels have the advantage that their heat emission is not affected by tapestry or furniture and that they can be used, to a limited extent, as cooling panels during the summer months. If used in low rooms, however, they may produce an undesirable heating effect upon the head. Floor panels have the advantage that they can be easily installed and that much of the radiated heat is delivered to the lower portions of the walls; they have the disadvantage that their heat emission is rather uncertain, since it may be affected by covering material such as rugs, carpets, furniture, and machinery.

It is best to make the panels as large as practicable; for example, if the floor or ceiling is used as the heating panel, it is best to use the entire floor or the entire ceiling, or both. Heating a room by means of panels is very similar to lighting a room. Heat radiation is exactly like light radiation, except that it has a longer wave length If a room is lighted by means of a large number of small units distributed uniformly over the ceiling, the room is lighted more uniformly than if it is lighted by means of a single unit of equal capacity. Similarly, if the entire ceiling is the heating panel, the room is heated more uniformly than if only a fractional part of the ceiling is used as the heating panel.

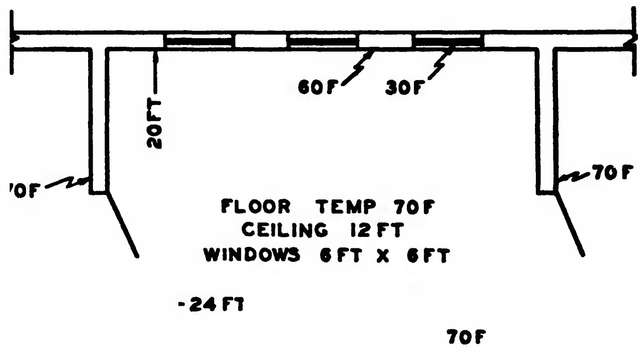


Fig. 7. Room Plan Used for Illustration of Method of Designing a Panel Heating System

In the following example, *the entire ceiling will be used as the heating panel.*

2. *Select the desired mean temperature of the air in the room.*

In a panel-heated room, the mean air temperature is a few degrees lower than the mean temperature of the surfaces of the enclosing walls, floor, and ceiling. In a room heated by introduction of warm air or by means of radiators, convectors, or other similar heating appliances, located within the room, the mean temperature of the air is a few degrees higher than the mean temperature of the surfaces of the enclosing walls, floor, and ceiling. Under ordinary conditions, in a panel-heated room, the mean temperature of the air ranges from approximately 65 F to 72 F.

In the following example, *68 F is selected as the mean air temperature.*

3. *Determine as accurately as practicable, the mean temperature of the inside surfaces of the enclosing walls, floor, and ceiling.*

The two inside walls are assumed to separate rooms, which are filled with 68 F air, so that both surfaces of each wall are in close contact with 68 F air. The surface temperatures of these walls, therefore, cannot be lower than 68 F and must actually be higher than 68 F because, in addition to their contact with 68 F air and their contact with the heated ceiling, they are exposed to the heat radiation from the ceiling.

In the following example, *70 F will be selected as the mean surface temperature of the inside walls.*

For the two outside walls, the mean inside surface temperature can be calculated with fair accuracy. For a heat transmission coefficient of 0.25 and a temperature difference of 68 deg, heat flows through the wall at the rate of 17 Btuh per square foot.

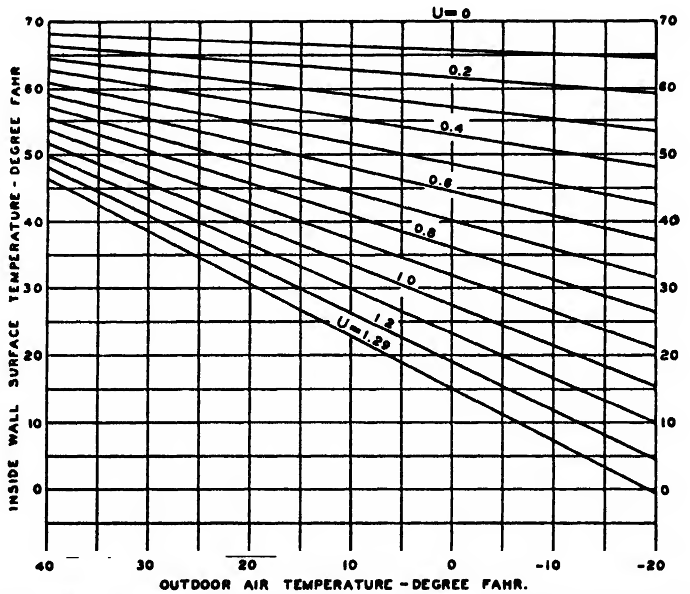


FIG. 8. CHART FOR ESTIMATING INSIDE SURFACE TEMPERATURES OF OUTSIDE WALLS[a]

[a]NOTE: The value of U, the over-all coefficient of heat transmission, cannot exceed 1.29 if the inside and outside film coefficients are 1.65 and 6.0 respectively (*e.g.* $\frac{1}{U} = \frac{1}{1.65} + \frac{1}{6} = \frac{1}{0.773}$. Therefore $U = 1.29$ maximum).

If the indoor film coefficient is 1.65, the temperature difference, indoor air to inside wall surface, is 17/1.65 or 10 deg and the wall surface temperature is 68 — 10, or 58 F. This value may be taken directly from Fig. 8. However, the film coefficient 1.65 was determined to represent the sum of the heat flow into the wall, by conduction from the air in contact with the wall, and by radiation from the warmer surfaces *seen* by the wall surface. In a panel-heated room, the rate of heat flow into the outside wall by radiation is greater than it is in a radiator-heated room; consequently, the film coefficient is higher, and the temperature difference, air to wall surface, is smaller, and therefore, the wall surface temperature is higher than the calculated 58 F. It is impossible to determine accurately how much higher than 58 F the temperature of the wall surface will be until the corresponding indoor air film coefficient has been determined accurately.

In the following example, *60 F will be selected as the probable mean inside surface temperature of the outside walls.*

The probable mean inside surface temperatures of the floor and the glass may be determined by calculations and by reasoning similar to that employed to determine the inside surface temperature of the outside walls.

TABLE 1. CALCULATED HEAT LOSS OF ROOM

SURFACE	AREA SQ FT	U	CALCULATION	HEAT LOSS BTUH
Outside Walls	360	0.25	360 x 0.25 x 68	6,120
Glass	216	1.13	216 x 1.13 x 68	16,597
Inside Walls	480	—	No heat loss	
Ceiling	480	—	Heating Panel	
Floor	480	0.10	480 x 0.10 x 38	1,824
Infiltration	—	—	5,760 cu ft x 1.50 x 68 x 0.018	10,576
			Total.	35,117

In the following example, *30 F and 70 F will be selected as the probable inside surface temperatures of the glass and floor,* respectively.

4. *Determine the heat loss of the room.*

In the following example, the heat loss calculation will be based on an outdoor air temperature of 0 F. Since the functioning of a panel-heating system differs very little from that of a radiator-type heating system, the heat loss shown in Table 1 may be calculated according to Chapter 14.

The heat loss through the outside walls and through the glass is probably a little greater than calculated because the calculation is based on an indoor air film coefficient of 1.65 Btuh, whereas, for a panel-heated room, this coefficient is a little higher, but the difference is probably not sufficiently large to be considered in design calculations for a heating system.

5. *Estimate the Mean Radiant Temperature.*

The Mean Radiant Temperature of the surfaces enclosing the room but not including the heating panels may be estimated as follows:

SURFACE	AREA	FAHR DEG	PRODUCT
Interior Walls	480	70	33,600
Exterior Walls	360	60	21,600
Glass	216	30	6,480
Floor	480	70	33,600
	1,536		95,280

The sum of these products divided by the sum of the surface areas is: 95,280/1,536 or 62.03 F, the required mean surface temperature.

In the following example, *62 F will be selected as the MRT of walls, glass and floors.*

6. *Determine the temperature of the ceiling panel.*

Determine the temperature of the ceiling so that the ceiling panel will deliver heat to the room at a rate equal to the rate at which the room is calculated to lose heat, namely, 35,117 Btuh.

When a room is heated by means of a panel, air convection currents are developed in the room similar to those which are developed when the room is heated by means of a free-standing radiator. Consequently, the heating panel delivers heat to the room partly by radiation and partly by convection. The proportion of the total heat flow delivered by convection varies with the location of the heating panel, with the height of the ceiling, and with the size, number, and location of pieces of furniture and other articles which interfere with the free flow of air along the floor and along the walls. It is generally sufficiently accurate to assume that a ceiling panel will deliver 70 per cent of its heat by radiation and 30 per cent by convection; a floor panel 55 per cent by radiation and 45 per cent by convection; and a wall panel 65 per cent by radiation and 35 per cent by convection.

In the following example, *it will be assumed that the ceiling panel must deliver 70 per cent of its heat or 24,582 Btuh by radiation*, since the total calculated heat loss is 35,117.

When two plane surfaces of infinite size are parallel to each other and their surfaces are at different temperatures, the exchange of heat between the two is proportional to the difference between the fourth powers of their absolute temperatures. This is also true when one surface is completely surrounded by another surface; for example if one sphere is placed within another sphere, the flow of heat between the outer surface of the smaller sphere and the inner surface of the larger sphere is proportional to the fourth power of the absolute temperatures of the two surfaces.

In a panel-heated room, the heated panel may be considered to be completely enclosed by the remaining surfaces, because all heat radiated by the heated panel is intercepted by those surfaces. Consequently, the flow of heat from the heated ceiling to the room, by radiation, is proportional to the difference between the fourth powers of the absolute temperature of the ceiling and the absolute mean radiant temperature of the remaining surfaces.

The rate at which a surface emits heat varies with the temperature of the surface and with other characteristics of the surface. For ordinary heat flow calculations it

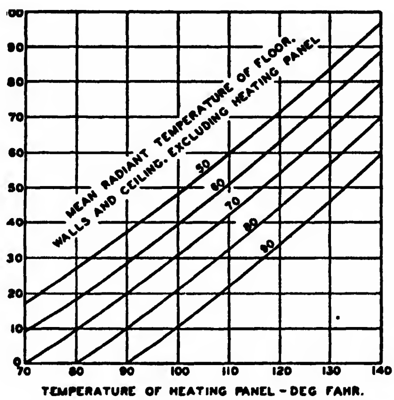


FIG. 9. HEAT DELIVERED TO ROOM BY RADIATION FROM PANEL

is sufficiently accurate to assume that the materials which are commonly used in building construction emit heat at a rate of:

$$0.156 \left(\frac{T}{100}\right)^4 \text{ Btuh per square foot}$$

where

T is the absolute temperature of the surface in Fahrenheit degrees.

On this basis the flow of heat from the ceiling to its surrounding surfaces is at the rate of:

$$480 \times 0.156 \left[\left(\frac{T}{100}\right)^4 - \left(\frac{522}{100}\right)^4\right] \text{ Btuh.}$$

In order that this rate may be equal to 24,582 Btuh, T must be 572 and the ceiling temperature about 112 F.

Instead of calculating this temperature it may be taken from Fig. 9, as follows: The ceiling must deliver heat to the room, by radiation, at the rate of 24,582/480 or 51 Btuh per square foot. Find 51 on the left margin and move horizontally to the intersection with a 62 MRT line, and from the point of intersection to the lower margin and read about 112 F.

With a ceiling temperature of 112 F, the MRT of the room will be 480 × 112 + 1,536 × 62; the sum divided by 2,016, or 74 F.

If an air temperature of 68 F and an MRT of 74 F should not produce satisfactory

TABLE 2. TOTAL HEAT EMISSION BY RADIATION[a]

Body or Mean Radiant Temperature F Deg	Radiation in Btu per (square foot) (hour) emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature F Deg	Radiation in Btu per (square foot) (hour) emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.7	94.7	89.8	79.7	71	137.0	130.1	123.3	109.7
35	103.9	98.7	93.6	83.1	72	137.9	131.0	124.0	110.3
40	108.0	102.8	97.2	86.4	73	138.9	132.0	124.9	111.0
45	112.5	106.9	101.3	89.0	74	140.2	133.1	126.1	112.1
46	113.3	107.7	102.0	90.8	75	141.6	134.4	127.4	113.2
47	114.3	108.6	102.9	91.5	80	147.2	140.0	132.5	117.9
48	115.2	109.5	103.8	92.3	85	152.9	145.2	137.6	122.4
49	116.0	110.3	104.5	92.8	90	158.5	150.5	142.7	126.9
50	116.9	111.0	105.3	93.6	100	170.3	161.7	153.2	136.2
51	117.9	112.0	106.4	94.4	110	182.3	173.2	164.2	146.0
52	118.8	112.9	106.9	95.1	120	195.6	185.7	176.1	156.5
53	119.8	113.8	107.8	95.9	130	210.9	200.4	189.8	168.8
54	120.6	114.6	108.6	96.6	140	224.1	212.9	201.8	179.2
55	121.7	115.5	109.4	97.3	150	238.0	226.1	214.4	190.5
56	122.6	116.4	110.3	98.1	160	252.1	239.8	226.9	201.8
57	123.5	117.4	111.3	98.9	170	271.6	258.0	244.5	217.2
58	124.4	118.2	112.0	99.6	180	289.1	274.9	260.1	231.3
59	125.3	119.0	112.8	100.3	190	307.7	292.1	276.9	246.1
60	126.3	119.9	113.8	101.1	200	326.5	310.2	293.9	261.3
61	127.1	120.7	114.4	101.8	210	349.3	331.9	314.3	279.5
62	128.2	121.8	115.3	102.6	220	373.0	354.4	335.7	298.2
63	129.1	122.6	116.2	103.3	250	439.5	417.6	395.6	351.6
64	130.1	123.5	117.1	104.1	300	577.3	548.2	519.6	461.8
65	131.0	124.4	117.9	104.8	350	743.0	705.8	668.6	594.3
66	132.1	125.5	118.8	105.8	400	945.8	898.5	850.8	756.5
67	133.0	126.4	119.7	106.4	450	1181.0	1121.0	1063.0	944.0
68	134.0	127.3	120.5	107.2	500	1470.0	1396.0	1323.0	1176.0
69	135.0	128.3	121.5	108.0	550	1798.0	1708.0	1619.0	1439.0
70	136.0	129.3	122.3	108.8	600	2181.0	2072.0	1962.0	1745.0

[a] These factors are calculated from the formula

$$q_r = e\left(\frac{0.173 \times T^4}{100,000,000}\right)$$

where

q_r = total radiation, Btu per (sq ft) (hr)
e = emissivity.
T = absolute temperature, Fahrenheit degrees.

conditions, the ceiling temperature can easily be changed as necessary by changing the temperature of the circulating water.

Calculations like the preceding may also be made with the aid of Table 2. The rate at which the ceiling must radiate heat exceeds by 51 Btuh per square foot the rate at which the ceiling receives radiant heat from its surroundings. Assuming the emissivity of the walls, floor, and ceiling to be 90 per cent of that of a *black* body, the heat radiated to the ceiling, from the surfaces whose MRT is 62 F, is (Table 2) at the rate of 115.3 Btuh per square foot; the ceiling must therefore radiate heat at the rate of 115.3 plus 51 or 166.3; its temperature must be (Table 2) between 110 F and 120 F, and, by interpolation, 112 F, as calculated.

7. *Select the medium for heating the ceiling panel.*

The medium may be electricity, steam, air, or water, but usually is air or water. If air is used it is generally heated in the basement, passed up through hollow inside walls or through ducts in those walls, allowed to flow between the ceiling and the floor above, and returned to the basement through hollow outside walls or through ducts in those walls.

If the walls and floors are constructed of hollow tile, the cells in the tile can be placed so that they will form continuous ducts through which the warm air can flow up the inside walls, then between the ceiling and the floor above, and down the outside walls. In this way the walls and ceiling become heating panels.

Table 3. Highest Safe Surface Temperatures for Heating Panel

Type of Panel	Surface Temperature F Deg
Plastered Ceiling (Pipes Imbedded)[a]	115
Plastered Walls (Pipes Imbedded)[a]	120
Floor, Any Method	85
Floor, Border and Aisles	120
Iron, Hot Water Medium[b]	160
Iron, Steam Vapor[a]	180
Electrically Heated Panels[b]	200

[a]*Gypsum Association* recommends that panels in which gypsum plaster or gypsum products are used should not have a surface temperature exceeding 115 F.
[b]Low surface temperature radiation is recommended regardless of the heating medium employed.

If water is used as the medium, the pipes through which the water circulates—almost always under forced circulation—are placed in the floor, walls, or ceiling in such a manner that as much as possible of the heat emitted by the pipes will be delivered to the space to be heated.

Practical limits for surface temperatures of heating panels are given in Table 3.

In this example water will be selected as the medium.

8. *Determine the size, length, and location of the pipe coils in the panels.*

When hot-water pipes are imbedded in concrete slabs or attached to plastered surfaces, their rate of heat emission varies with many factors. If the pipes are imbedded in dense concrete slabs, it may be assumed that the rate of heat emission of ½-in. pipe, spaced 6 in. on centers; ¾-in. pipe, spaced 9 in. on centers; and 1-in. pipe spaced 12 in. on centers; per foot of length of pipe and per degree difference between the temperature of the water in the pipe and that of the air in the space to be heated, is 0.8, 1.0, and 1.2 Btuh, respectively. If the distance between the pipes is increased, the rate of heat emission, per foot of pipe, is also increased; if the distance is doubled, the rate of heat emission is increased about 15 per cent. If the pipes are attached to plastered ceilings, the rate of heat emission is slightly less, probably about 10 per cent less, than when the pipes are imbedded in concrete slabs. The data given regarding heat emission of panels are intended as general guides for the designer. Additional experience and research are needed to develop definite and complete data. However, after a heating panel has been designed and installed, any small error can easily be corrected by modifying the temperature of the water circulating through the coils.

When the heating pipes are attached to a plastered ceiling, a portion of the heat emitted by the pipes is delivered to the space below the ceiling and a portion to the space above the ceiling. The relative quantities depend on the degree of insulation applied above the heating coils.

When the heating pipes are imbedded in a concrete floor slab a portion of the heat emitted by the pipes will flow upward into the space to be heated, and the remainder will flow downward into the ground.

When the heating pipes are placed below the concrete floor slab instead of being imbedded in the slab, a larger portion of the heat will flow into the ground, and a smaller portion into the space to be heated.

In the following example *it is assumed that the insulation above the pipe coils is such that 90 per cent of the heat emitted by the pipe coils will flow into the room and 10 per cent into the space above.*

Since the room is to receive 35,117 Btuh, and since the room is assumed to receive only 90 per cent of the heat emitted by the coils attached to the plastered ceiling, the coils must emit 35,117/0.9 or 39,000 Btuh. If ¾-in. pipe and a mean water temperature of 140 F are selected, the heat emitted, per foot of pipe, will be 0.9(140 — 68) or 65 Btuh. The quantity of pipe required will therefore be 39,000/65 = 600 lineal feet.

The pipe coils can be arranged in any convenient manner, but should be arranged so that the temperature of the water in the pipe will vary only slightly; otherwise, the temperature distribution over the ceiling will not be uniform. Generally, it is best to arrange the pipes so as to form two-pipe reversed-return flow circuits in the sepa-

rate panels. By using 33 runs of ¾-in. pipe, welded to two 1¼ in. mains, sufficient pipe surface is secured; the ¾-in. pipes will then be spaced about 8¾ in. on centers, which is satisfactory.

While coils can be designed with pipe and fitting resistances which will insure proper distribution to each coil it is advantageous to provide adjustable flow control valves or resistances for final regulation of the water temperature or flow to the various coils. It is desirable to divide large heating systems into sections and to install valves so that individual sections can be disconnected without interfering with the operation of the system as a whole.

Part II—Radiant Heating

The term *radiant heating* is applied in this chapter to a system in which only the heat radiated from the panel is effective as in outdoor and semi-outdoor conditions.

The outstanding example of radiant heating is the transfer of heat from the sun to the earth. The sun radiates large quantities of energy of which a very small portion is intercepted by the earth. A part of the intercepted radiation is transformed into heat when it strikes the earth's surface. In this manner *heat* is received by the earth from the sun by *radiation.*

In industry, radiant heating is employed in manufacturing processes, particularly in drying, baking, and dehydrating operations; in agriculture, it is employed to improve living and growing conditions for young plants and young animals.

The heating engineer employs radiant heat primarily in the heating of open-air schools and open-air hospitals. When a surface radiates heat, and every surface does unless its temperature is absolute zero, every point of the surface radiates heat in all directions. The total quantity of heat radiated by a point or by an elementary area is π times the quantity of heat radiated at right angles to the surface.

Thus, if in an elementary cube the upper face is the heating panel, the lower face would receive only about 32 per cent of the radiated energy and the four sides would receive each about 17 per cent.

For larger surfaces the conditions are different. If two parallel plane surfaces of considerable size are near each other, the rate of heat exchange between the two can be determined fairly accurately by means of the chart of Fig. 9. This is possible because the larger part of the heat radiated by one of the surfaces is intercepted by the other surface and only a small portion is radiated in such directions that it will not impinge upon the opposite surface.

As the distance between the two surfaces is increased, the proportion of the heat radiated by one of the parallel plane surfaces and intercepted by the other decreases almost as the square of the distance between the surfaces increases, because the intensity of heat radiation, like the intensity of light radiation, varies inversely as the square of the distance from the source of radiation.

For the purpose of designing radiant heating systems in which the heating panel is practically square and is radiating heat toward a parallel surface of equal size and shape, as shown in Fig. 10, the rate of heat exchange between the two surfaces will be equal to that shown in Fig. 9, multiplied by a factor, p, which depends upon the ratio of h to s (Fig. 10) as shown in the following table:

$h/s =$	1	2	3	4	5
$p =$	0.200	0.070	0.034	0.020	0.013

For example, if a panel 3 ft square is located parallel to, and 9 ft above,

a bed in an open-air hospital, and if the temperature of the panel is 112 F and that of the bed is 70 F, the rate of heat transfer from the panel to a 9 sq ft section of the bed directly beneath the panel, will be 3.4 per cent of the rate shown in Fig. 9, or 0.034 × 9 × 44, or 14 Btuh, approximately. The rate of heat transfer from the panel to a section of the bed other than the 9 sq ft directly beneath the panel will be lower than 14/9 Btuh per square foot.

This is a crude way of designing a radiant heating system for an open-air hospital, but it is sufficiently accurate, because the required temperature of the bed and the required rate of heat flow into it will vary with the temperature of the outdoor air, with the air movement over the bed, with the

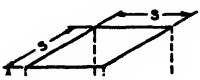


Fig. 10. Effect of Height Upon Radiation Received from a Panel

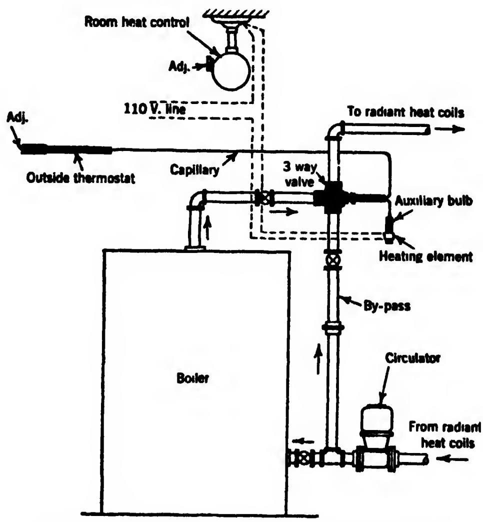


Fig. 11. Typical Panel and Radiant Heat Control System

thickness and the character of the bedding, and with the physical condition of the patient.

A radiant heating system for an open-air school may be designed as described for the open-air hospital. The heating panel in such a case should be almost as large as the ceiling and, in order to keep the heat loss by radiation at a minimum, should be placed so that a maximum portion of the heat radiated by the panel will be directed toward the pupils and a minimum toward the outside walls and particularly the windows.

MEASUREMENT OF RADIANT HEATING

Radiant heating is intended to control the rate of radiant heat loss from the human body and should be measured by calorimetric methods.

The apparatus for this purpose consists essentially of a cylinder, maintained at the accepted mean surface temperature of the human body, together with an accurate (usually electrical) measuring of the varying rate of heat supply required to maintain this exact temperature. This instrument, the *eupatheoscope*, is readily adapted to function like a thermostat so as to turn heat on or off, when the desired temperature of 80 F, or any other predetermined surface temperature of the cylinder, decreases or increases as a result of changes in the Operative Temperature.

For testing work, the *globe thermometer* is a useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black and sometimes covered with cloth. The temperature recorded by a thermometer with its bulb in the center of the sphere is termed the *radiation-convection temperature.* See Chapter 11.

CONTROL OF PANEL AND RADIANT HEATING

The effectiveness of any type of control will depend largely on the time lag of the system. With warm air passing through floor ducts the time lag is usually too long for any kind of room thermostat, in fact a thermostat will not prove suitable with any system if the building is constructed with massive brickwork and masonry, unless it operates in conjunction with a time control responsive to changes in outside conditions.

The heat emitted by hot water pipes imbedded in the plaster of the ceiling and walls or in the concrete base of a floor can be effectively controlled by an instrument designed to modulate the temperature of the water circulating in the system according to the outside conditions. Metal panels which can be installed in the ceiling or side walls may be either controlled by an instrument responsive to outside weather conditions or by a specially designed instrument responsive to both air temperature and radiation. Any purely *on or off* control system is not recommended for panel heating.

A typical control system operated from an outside thermostat and supplemented with a room heat control instrument is illustrated in Fig. 11. The outside thermostat modulates the temperature of the circulating water in the coils by mixing some of the hot water leaving the boiler with a proportionate amount of return water which is diverted to the three-way valve.

One type of room instrument consists of a blackened copper sphere of 6 or 8 in. in diameter, in which a cylindrical sump contains a volatile liquid. A small electric heating coil creates in the sphere a vapor pressure which remains constant as long as the total heat loss from the sphere is at the desired rate. If the Operative Temperature becomes too high for comfort, a greater vapor pressure results from the smaller heat loss from the sphere. This acts on a diaphragm and reduces the supply of heat to the room. With too low an Operative Temperature the reverse action occurs. A similar instrument which has an electric heating element for warming the air inside the sphere and the thermostat-operated switch, is also used for controlling room conditions.

In addition to a thermostatically controlled device for modulating the temperature of the circulating water, it is advantageous to insert in each coil a locked flow control or adjustable resistance to give uniform conditions throughout all rooms. Owing to unforeseen difficulties with varying frictional losses in pipes, emission factor, and exposures, it is an advantage to be able to regulate permanently the flow through each circuit by means of a key operated valve as indicated in Fig. 4.

REFERENCES

[1] Standard Operative Temperature, A Generalized Temperature Scale, Applicable to Direct and Partitional Calorimetry, by A. P. Gagge (*American Journal Physiology*, 1940, Vol. CXXXI, p. 93).

[2] The Influence of Air Movement Upon Heat Losses from the Clothed Human Body, by C.-E. A. Winslow, A. P. Gagge and L. P. Herrington (*American Journal Physiology*, 1939, Vol. CXXVII, p. 505).

[3] Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transactions, Fuels and Steam Power Section*, Vol. 51, No. 22, September–December, 1929).

BIBLIOGRAPHY

Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 425).

A.S.H.V.E. RESEARCH REPORT No. 1192—Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 35).

A.S.H.V.E. RESEARCH REPORT No. 1193—Radiation as a Factor in the Feeling of Warmth in Convection, Radiator and Panel Heated Rooms, by F. C. Houghten, Carl Gutberlet and E. C. Hach (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 55).

Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 285).

Operating Results of a Residence Radiant Wall Heating System, by E. J. Rodee (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 123).

Performance of a Residential Panel Heating System, by H. F. Randolph and J. B. Wallace (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 235).

Radiant Heating (*Heating and Ventilating*, March, 1941, p. 35).

Radiant Heating and Cooling, by F. E. Giesecke (*Heating, Piping and Air Conditioning*, June, July, August, September and October, 1940).

Calculations for Radiant Heating, by T. Napier Adlam (*Heating and Ventilating*, October, 1931).

Radiant Heating and Cooling, Part I, by C. O. Mackey, L. T. Wright, Jr., R. E. Clark, and N. R. Gay (*Cornell University Engineering Experiment Station*, Bulletin No. 32, 1943).

Design Method for Panel Heating Systems Using Copper Tubing, by R. G. Vanderweil (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Nov. 1947, p. 123).

Air Temperature Gradients in a Panel Heated Room, by J. M. Ayres and B. W Levy (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Oct. 1947, p. 113).

Graphical Solution of Radiant Panel Areas, by W. P. Chapman and R. E. Fischer (*Heating and Ventilating*, Jan. 1948, p. 88).

Embedding Coils in Radiant Heating Panels, by D. L. Mills and L. J. LaTart (*Heating and Ventilating*, Dec. 1947, p. 75).

Radiant Heat with Copper Tubing, by D. L. Mills and L. J. LaTart (*Heating and Ventilating*, Nov. 1947, p. 95).

Panel Heat with Copper Tubing—Experiment in Practice, by D. L. Mills and L. J. LaTart (*Heating and Ventilating*, Oct. 1947, p. 65).

Experimental Studies on Panel Heating Tube Spacing, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Aug. 1947, p. 111).

Panel Heating—A Basic Discussion, by S. Konzo (*American Artisan*, Oct. 1946, p. 68).

Solar House Heated by a Warm Air Floor Panel, by John E. Peterson (*American Artisan*, Dec. 1946, p. 83).

Design Data for a Warm Air Floor Panel, by Axel B. Algren (*American Artisan*, Jan. 1947, p. 141).

The Mechanism of Heat Transfer, Panel Cooling, Heat Storage, by Charles S. Leopold (*Refrigerating Engineering*, July 1947, p. 33).

Books

Hot Water Heating, Radiant Heating and Radiant Cooling, by F. E. Giesecke, 1947 (Technical Book Co., P. O. Box 62, Austin, Tex.).

Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson, 1945 (John Wiley and Sons, New York).

Radiant Heating, by T. Napier Adlam, 1947 (Industrial Press, 148 Lafayette St., New York).

Radiant Heating, by R. W. Schoemaker, 1948 (McGraw-Hill Book Co., New York).

CHAPTER 32

FANS

Classification, Performance, Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Fan Designations, Control, Motive Power, Attic Fans

IN heating, ventilating, and air conditioning practice, fans and blowers are used to produce air flow. All fans or blowers are classified according to the direction of air flow through the fan with relation to the axis of rotation and are either of the (1) *axial flow* or *propeller* type, in which the flow is parallel to the axis, or (2) *radial flow* or *centrifugal* type, in which the flow is at right angles to the axis.

Axial flow fans are made in various designs and sizes. The form and number of blades may vary for impellers of the same or different diameters. The blades may be of uniform thickness and made of cast or sheet metal, and either flat or cambered or of screw form; or they may vary in thickness, in the latter case usually being designed to conform to so-called *airfoil* sections of known characteristics, similar to those which have been developed for airplane propellers. Likewise, blade angle, or the angular relation of the blades to the plane of rotation, varies over a wide range. For operation against comparatively high pressures, it is customary to resort to enlarged hubs in proportion to fan diameter (large hub ratio) and correspondingly short blade length. The term *disc fan* has sometimes been loosely applied to such large hub fans, though it has long been generally used in connection with any propeller fan of comparatively short axial length whose blades are relatively flat; in other words, for fan wheels which occupy a space which is more or less disc-shaped. Single stage air foil axial flow fans can operate at pressures up to 4 in. at a quite low noise level and, at considerably higher pressures, they are comparable in regard to noise to the centrifugal fan type.

Radial flow or *centrifugal fans* used in heating, ventilating, and air conditioning practice, are in general of two types; one with forward-sloped blades, and the other with backward-sloped blades. Many modifications may be made in the proportions, the curvature, and the slope or angularity of the blades. The slope or angularity of the blades determines the operating characteristics of a fan; a forward-curved or sloped blade is found in a fan having low speed operating characteristics, while a backward-curved or sloped blade is found in a fan having higher speed operating characteristics.

A wide variety of heating, ventilating, and air conditioning systems creates a wide variety in the demands that have to be met by the fan or blower. The requirement may be to move small or large quantities of air against little or no resistance, or to move small or large quantities of air against higher resistance. Between the two limits innumerable specific requirements must be met. Although fans of any class in either of the two general types can in general be made to perform the same duty, certain factors such as mechanical difficulties, space and noise limitations, efficiency and power limitations will usually determine the selection.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in predicting the effect upon performance of changes in the conditions of operation, the duty required of the installation, or the size of the equipment due to the space, power, or speed limitations. The following laws in which Q = air volume and P = static, velocity or total pressure, apply to all types of fans:

1. Variation in Fan Speed:
   Constant Air Density—Constant System
   (*a*) Q: Varies as fan speed.
   (*b*) P: Varies as square of fan speed
   (*c*) Power: Varies as cube of fan speed.

2. Variation in Fan Size
   Constant Tip Speed—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) Q: Varies as square of wheel diameter.
   (*b*) P: Remains constant.
   (*c*) RPM: Varies inversely as wheel diameter.
   (*d*) Power: Varies as square of wheel diameter.

3. Variation in Fan Size:
   At Constant RPM—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) Q: Varies as cube of wheel diameter.
   (*b*) P: Varies as square of wheel diameter.
   (*c*) Tip Speed: Varies as wheel diameter.
   (*d*) Power: Varies as fifth power of diameter.

4. Variation in Air Density:
   Constant Volume—Constant System
   Fixed Fan Size—Constant Fan Speed
   (*a*) Q: Constant.
   (*b*) P: Varies as density.
   (*c*) Power: Varies as density.

5. Variation in Air Density:
   Constant Pressure—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) Q: Varies inversely as square root of density.
   (*b*) P: Constant.
   (*c*) RPM: Varies inversely as square root of density.
   (*d*) Power: Varies inversely as square root of density.

6. Variation in Air Density:
   Constant Weight of Air—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) Q: Varies inversely as density.
   (*b*) P: Varies inversely as density.
   (*c*) RPM: Varies inversely as density.
   (*d*) Power: Varies inversely as square of density.

Examples 1 to 4 illustrate the application of the preceding fan laws.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp.}$$

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.075 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.0602 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.0602}{0.075} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.0602}{0.075} = 3.20 \text{ hp.}$$

Example 3. If the speed of the fan of *Example 2* is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.075}{0.0602}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.075}{0.0602}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.075}{0.0602}} = 4.46 \text{ hp.}$$

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.075}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.075}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.075}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.075}{0.0602}\right)^2 = 6.20 \text{ hp.}$$

Law of Homologous Fans

The laws applying to different sizes of homologous fans are as follows:

Capacity varies as the ratio of size cubed, times the ratio of the rpm.

Pressure varies as the ratio of size squared, times the ratio of the rpm squared.

Horsepower varies as the ratio of the size to fifth power, times the ratio of the rpm cubed.

Example 5. Assuming that a fan with a 36 in. diameter blast wheel will deliver 12,000 cfm at 70 F at 1 in. static pressure, requiring 4.0 brake hp when operating at 400 rpm, what is the capacity, pressure and horsepower of a homologous fan having a 45 in. wheel at the same speed?

$$\text{Capacity} = \left(\frac{45}{36}\right)^3 \times \left(\frac{400}{400}\right) \times 12{,}000 = 23{,}400 \text{ cfm}$$

$$\text{Static Pressure} = \left(\frac{45}{36}\right)^2 \times \left(\frac{400}{400}\right)^2 \times 1 = 1.56 \text{ in.}$$

$$\text{Horsepower} = \left(\frac{45}{36}\right)^5 \times \left(\frac{400}{400}\right)^3 \times 4 = 12.2 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^{[1]} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \tag{1}$$

When the static pressure is used in the computation in place of total pressure it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but, owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure. In the standards for published capacity tables as adopted by the *National Association of Fan Manufacturers*, the term *static pressure* refers to the true resistance to air flow. Such tables charge both the inlet and outlet velocity of the fan to the fan performance, and may be used directly where the static pressure of the system as calculated represents only the actual resistance to flow of the air.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static Efficiency}^{[1]} = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{2}$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total pressure. This efficiency is variously known as the total, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total Efficiency}^{[1]} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{3}$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different

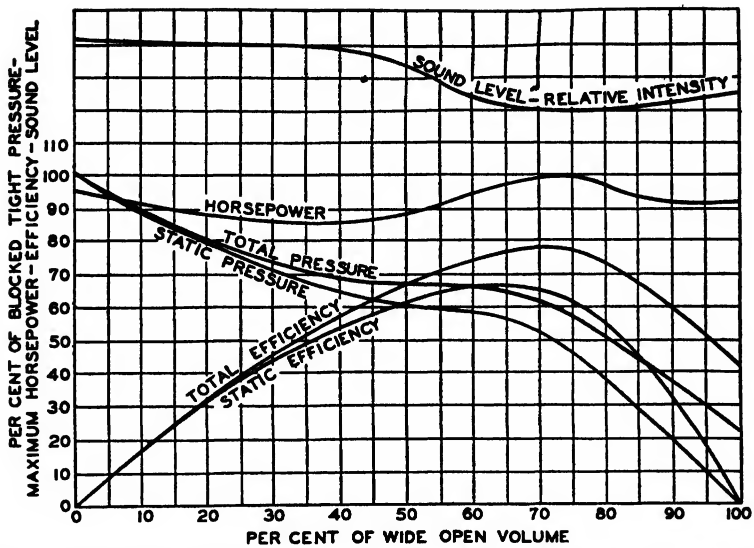


FIG. 1. OPERATING CHARACTERISTICS OF AXIAL FLOW AIRFOIL TYPE FAN

designs the peak of efficiency occurs when the fans deliver different percentages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans based upon tests performed in accordance with the Standard Test Code for Centrifugal and Axial Fans[1] prepared

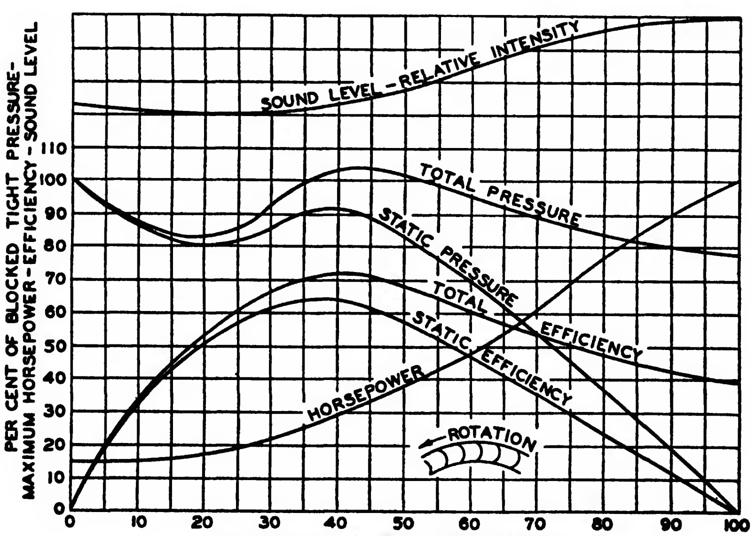


FIG. 2. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

jointly by the American Society of Heating and Ventilating Engineers and the *National Association of Fan Manufacturers* are generally plotted to show total and static pressure, mechanical and static efficiency, and horsepower in relation to air delivery as a basis. Results may also be plotted against per cent of wide open volume or discharge. Examples of fan performance curves are shown in Figs. 1, 2 and 3.

In the selection of all but very small fans, power consumption is usually a major consideration. It must be borne in mind that the horsepower at peak efficiency alone may be misleading, as actual operation is apt to occur at some point on the pressure-volume curve varying considerably from that specified, due to inaccuracies of the estimated system resistance or to fluctuating resistance caused by damper or louver adjustments. To cope

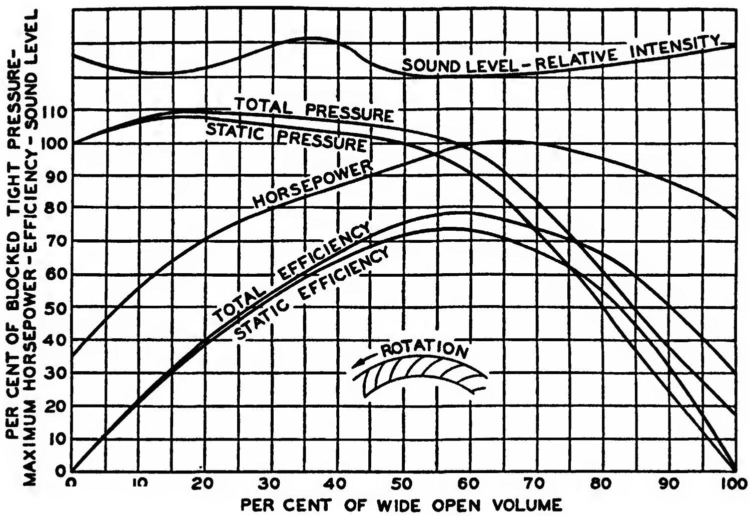


Fig. 3. Operating Characteristics of a Fan with Blades Curved Backward

with such variations a fan should be selected having a high efficiency over a wide range, that is, a *flat* or broad efficiency curve is more desirable than a sharp or narrow curve which, though reaching a high peak, falls off rapidly to either side of a narrow range. When the point of operation varies only within narrow limits and both volume and pressure requirements are accurately known in advance, the designer can select a fan operating at maximum efficiency, irrespective of performance over the entire range.

Generally, fans are selected either at the peak of the static efficiency or to the right of the peak depending on the requirements of the particular installation. Fans selected to the right of the peak will be smaller but will require more power, run at higher speeds and may have a higher sound rating. Where first cost is important and added horsepower and noise are not important, smaller fans may be used. Where efficient and quiet operations are most important, fans are selected at or near the peak of the static efficiency curve. Fans are not ordinarily selected to the left of the peak of the static efficiency curve as this results in larger, more costly fans, requiring more power and in some cases producing objectionable noise.

The curves in Figs. 1, 2, and 3 show operating characteristics for axial

flow and two classes of centrifugal fans, namely, forward-curved multiple blade and backward-inclined blade design, for comparison purposes. These curves are not applicable for rigid comparison or actual selection, but are shown to indicate variations in operating characteristics.

The curves in Fig. 1 of a *typical axial flow* fan show characteristics of non-overloading horsepower and high efficiency. These results are obtained by producing a more uniform pressure throughout the blade annulus, so that back flow does not occur except at high pressures. This avoidance in turbulence has a tendency to reduce noise. Fans of this type are operating against static pressures as high as 4 in. water. The capacity and efficiency of axial flow fans when operating above the low pressure range can be improved by the use of either inlet or outlet guide vanes or both. The effect of such vanes is to increase the level of the pressure volume curve, and properly designed vanes on the discharge side of the fan have the advantage of eliminating the rotational component of the air stream, thus restoring uniform axial flow. As high pressures usually require large hubs in proportion to the fan diameter, performance is improved by the use of round-nosed or conical forms mounted coaxially with the direct-connected fan (sometimes partly or wholly enclosing the motor) so as to make the changes in velocity to and from the fan blade annulus as uniform as space conditions permit. When axial flow fans are installed in ducts, provisions may be made to install the driving motor outside by employing slots in the duct to permit a belt drive from motor to fan sheave, or by extending the shaft for a direct-connected motor placed outside of a Y fitting or elbow in the duct system.

The *forward-curved multiblade* fan and the *backward-curved* type are used extensively in heating, ventilating, and air conditioning work. The forward-curved type has a low peripheral speed and a large capacity. (See Fig. 2.) The point of maximum efficiency for this fan occurs near the point of maximum pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continuously from low to peak capacity and, if reasonable care is exercised in calculating resistance, a moderate reserve in power in the motor selection will prevent overloading.

The *backward-sloped type* includes the full backward-curved blade and the double-curved blade having a forward-curved heel and a backward-curved tip. This type has steep pressure curves, non-overloading power characteristics, and relatively high speed (see Fig. 3). This fan operates at a peripheral speed approximately 175 to 200 per cent of that of the forward-curved multiblade fan for like performance. Pressure curves for this type begin to drop at very low capacity, with the most rapid drop beginning at about 60 per cent of wide open volume. The steep portions of the pressure curves tend to produce nearly constant capacity under changing pressures. Where wide fluctuations in demand occur, especially where the regulation is obtained by damper control and particularly through by-passes, this type of fan is desirable to prevent overloading of motor. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type makes it adaptable for direct connected electric motor drives.

Between the extremes of the forward and backward curved blade type centrifugal fans there exists a number of modified designs differing in

angularity and in the shape of the blades. Characteristic curves of these types show varying degrees of similarity to the curves in Figs. 2 and 3.

SYSTEM CHARACTERISTICS

Any ventilating system consisting of duct work, heaters, air washers, filters, etc., has a system characteristic which is individual to that system and is independent of any fan which may be applied to the system. This characteristic may be expressed in curve form in exactly the same manner that fan characteristics may be shown. Typical system characteristic curves are shown as A, B and C in Fig. 4. These curves are drawn to follow the simple parabolic law in which the static pressure or resistance

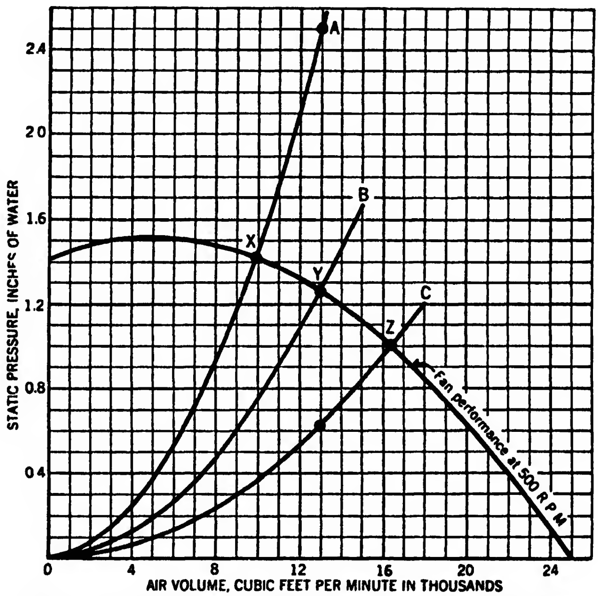


FIG. 4. PARABOLIC SYSTEM CHARACTERISTIC CURVES

to flow of air varies as the square of the volume flowing through the system. Heating and ventilating systems follow this law very closely and no serious error is introduced by its use.

When a constant speed fan curve for a given size fan is super-imposed upon a system characteristic curve, the relation between the two is at once apparent. The only point common to the two curves is the point at the intersection of the system characteristic curve and the fan characteristic curve, and it is at this point that the combination will operate. In Fig. 4, system characteristic curves A, B and C cross the fan characteristic curve at points X, Y and Z. The fan whose curve is shown, when applied to systems having characteristic curves A, B and C, will deliver 10,000, 13,000 or 16,400 cfm respectively.

The curves in Fig. 4 also illustrate the effect of errors which may be made in calculating the resistance of a ventilating system. For instance, if a given system requires 13,000 cfm and the resistance to flow of the system has been computed as 1.25 in. static pressure, such a system would be represented by system characteristic curve B in Fig. 4. If a 100 per

cent error had been made and the resistance were 2.5 in. instead of 1.25 in., then the system characteristic would be as shown in curve A and would cross the fan curve at 10,000 cfm. Such an error would cause the flow of air to be decreased from a design volume of 13,000 cfm to 10,000 cfm. If the resistance to flow had been over estimated and the resistance actually were 0.625 in., the system characteristic curve would be as shown in curve C and the fan would deliver 16,400 cfm to the system instead of the design volume of 13,000 cfm.

In this example extreme errors have been selected to emphasize the effect the square function of the system characteristic has in maintaining the fan performance within comparatively narrow limits. In the first example a system estimated at half what it should have been, resulted in a drop of 23 per cent in volume; and in the second example, a system estimated at twice what it should have been resulted in an increase of 26 per cent in volume.

In some instances fans may be applied to variable flow systems. In such cases the limiting systems may be plotted and the effect on fan performance examined. For instance, a system might have a characteristic curve between A, shown in Fig. 4, as one limit, and B as the other limit. The fan performance will then fall between points X and Y on the fan curve at a point determined by the system characteristics at that particular time. If A and B are the limiting characteristic curves of the systems, the fan performance will never be outside the points X or Y.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Density of air to be moved.
4. Type of motive power available.
5. Whether fans are to operate singly or in parallel on any one duct.
6. What degree of noise is permissible.
7. Nature of the load, such as variable air quantities or pressures.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures: (1) volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.075 lb per cubic foot), (2) outlet velocity, (3) revolutions per minute, (4) brake horsepower, (5) tip or peripheral speed, and (6) static pressure. The most efficient operating point is usually shown by either bold-face or italicized figures in the capacity tables.

Other important factors to be considered in selecting fans are: (1) efficiency, (2) space ocupied, (3) sound emission, (4) first cost, and (5) speed (both peripheral and revolutions per minute). These factors are not necessarily shown in the order of importance. In some installations, space occupied may be of first importance; in others lowest power consumption is desirable. In many cases quietness of operation of the entire system is essential. Practically all fans operate at their lowest sound level when selected at or near the peak of the static efficiency so that in selecting a fan for highest static efficiency the quietest operating

range of the fan will also be obtained. Table 1 shows desirable outlet velocities and tip speeds, or peripheral velocities, for various static pressures. Fans selected accordingly will operate at or near the peak of the static efficiency with resulting low power consumption and noise levels. Smaller fans with higher outlet velocities may be used if the conditions are such as to warrant the additional power and increased sound level. When space for duct expansion from a fan outlet is not available there may be advantages in selecting a larger fan for reducing duct noises, although lower outlet velocities generally result in lower fan efficiencies which cannot always be justified on the basis of increased cost and space. Fans for schools, churches, residences, and all public buildings should be selected for lower outlet velocities and tip speeds than would be required for other types of installations.

Having selected a fan for its quietest operating point consistent with the requirements of the installation (note sound level curves on Figs. 1, 2 and 3), it must be recognized that ventilating fans, even so selected, emit noise and precautions must be taken in the installation of the fans to prevent this noise from being transmitted to occupied portions of the building. Fans operating against high static pressures produce more noise than fans operating against low static pressures. Consequently, from a noise standpoint, the system should be designed to operate against the lowest static pressure possible. In many modern air conditioning systems it is necessary to introduce devices into the air stream for conditioning the air in various ways, the result of which is to set up a rather high static pressure against which the fan must operate. In such cases the sound level at the fan may be too high to be neglected and special sound treatment of the installation must be considered. When a fan is operating against higher pressures it should be located in a room either removed from the occupied areas, or in a room which has been acoustically treated to prevent sound being carried through the walls to adjoining spaces. The fan should be mounted on a resilient base along with its driving motor to absorb any noise or vibration which might be transmitted to the floor and thence to the building structure. All ducts should be connected to fans with un-

Table 1. Good Operating Velocities and Tip Speeds for Multiblade Ventilating Fans

Static Pressure Inches of Water	Forward Curved Blade Fans		Backward Tipped and Double Curved Blade Fans	
	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
¼	1000–1100	1520–1700	800–1100	2600–3100
⅜	1000–1100	1760–1900	800–1150	3000–3500
½	1000–1200	1970–2150	900–1300	3400–4000
⅝	1200–1400	2225–2450	1000–1500	3800–4500
¾	1300–1500	2480–2700	1100–1650	4200–5000
⅞	1400–1700	2660–2910	1200–1750	4500–5300
1	1500–1800	2820–3120	1200–1900	4800–5750
1¼	1600–1900	3162–3450	1300–2100	5300–6350
1½	1800–2100	3480–3810	1400–2300	5750–6950
1¾	1900–2200	3760–4205	1500–2500	6200–7550
2	2000–2400	4000–4500	1600–2700	6650–8050
2¼	2200–2600	4250–4740	1700–2800	7050–8550
2½	2300–2600	4475–4970	1800–2950	7450–9000
3	2500–2800	4900–5365	2000–3200	8200–9850

painted canvas, or other flexible material, to prevent any vibrations being transmitted to the duct work. Ducts leading into the fan room or from the fan should be acoustically treated on the interior and in special cases should be provided with sound traps or filters. Many ventilating systems encounter noises which are connected with the fan in no way. Noises due to high duct velocities, abrupt turns, grilles, etc., may be present. Treatment of such problems is covered in Chapter 42.

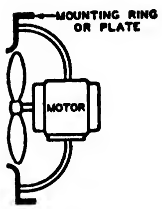
If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. Proper installation and location are as important as the selection of the size of fan for that particular installation. All fans will work according to their characteristic curves if properly selected and properly installed and operated.

FAN DESIGNATIONS

Fig. 5 shows the names and definitions of types of fans published by the *National Association of Fan Manufacturers.*[2]

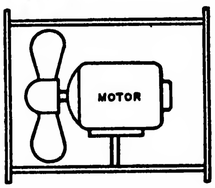
In order to prevent misunderstandings, which may cause delays and losses, the arrangements and designations adopted by the *National Association of Fan Manufacturers*[3] and indicated in Figs. 6 and 7 should be used.

Fig. 5. Names and Definitions of Types of Fans



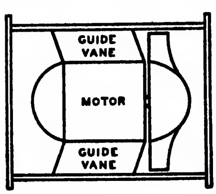
Propeller Fan

A propeller fan consists of a propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports either for belt drive or direct connection.



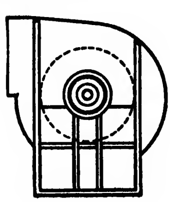
Tubeaxial Fan

A tubeaxial fan consists of a propeller or disc type wheel within a cylinder and including driving mechanism supports either for belt drive or direct connection.



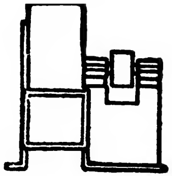
Vaneaxial Fan

A vaneaxial fan consists of a disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports either for belt drive or direct connection.

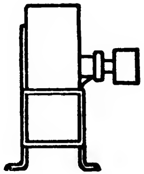


Centrifugal Fan

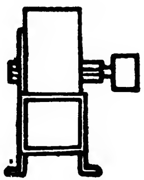
A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing and including driving mechanism supports either for belt drive or direct connection.



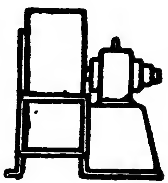
Arr. 1.
For belt drive
Wheel overhung. Bearing on pedestal.



Arr. 2.
For belt drive
Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.

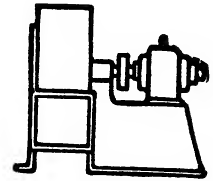


Arr. 3.
For belt drive
Pulley overhung. Bearings supported on fan housing.

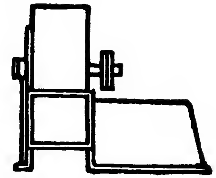


Arr. 4
For direct drive
Wheel overhung. No bearings on fan. Wheel mounted on motor or engine shaft. Pedestal for motor or engine.

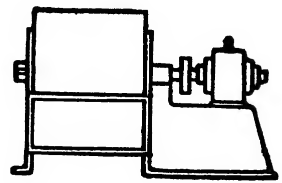
Arr. 5.
For direct drive
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive
Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.
Similar to Arr. 6, but with two bearings on fan, and flexible instead of rigid coupling.



Arr. 8.
Similar to Arr. 5.
But with two bearings on pedestal with motor, and flexible instead of rigid coupling.

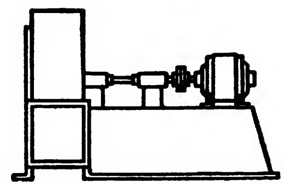


FIG. 6. ARRANGEMENT OF FAN DRIVES

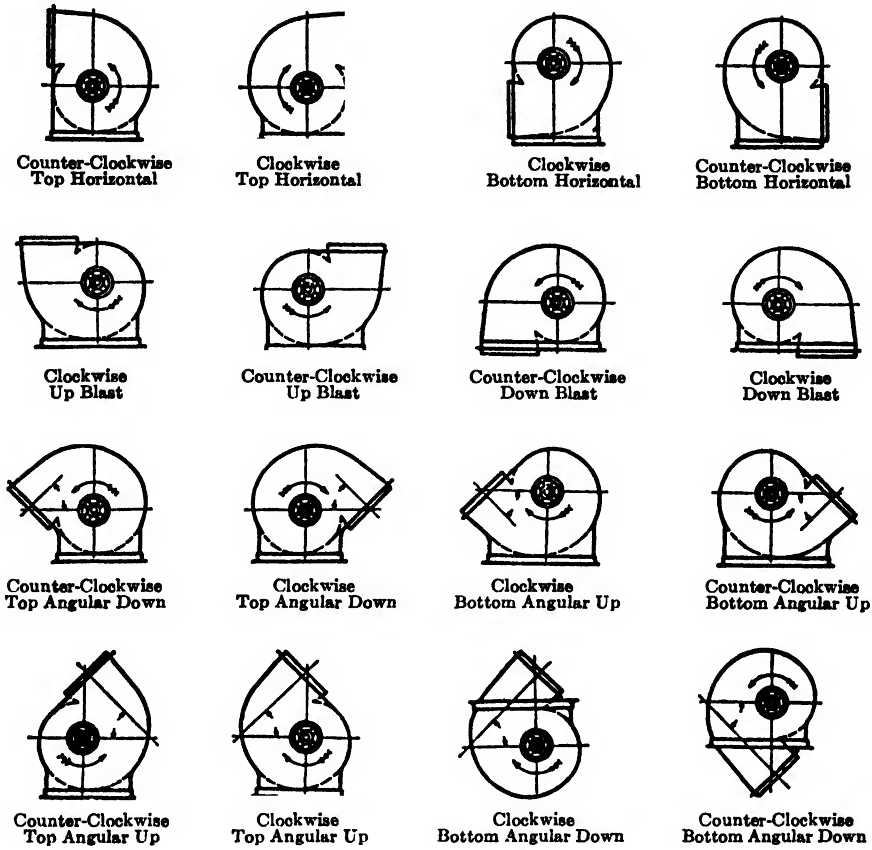


FIG. 7. DESIGNATION OF DIRECTION OF ROTATION AND DISCHARGE

Note: Direction of Rotation is determined from the drive side for either single or double width or single or double inlet fans. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)

FAN CONTROL

In some heating and ventilating systems it is desirable to vary the volume of air handled by the fan, and this may be accomplished by a number of methods. Where the change is made infrequently, the pulley or sheave on the driving motor, or fan, may be changed to vary the speed of the fan and alter the air volume. Dampers may be placed in the duct system to vary the volume. Variable speed pulleys or transmissions, such as fan belt change boxes, electric or hydraulic couplings, may be used to vary the fan speed. Variable speed motors and variable fan inlet vanes may also be used to adjust the fan volume. All of these methods will give control. From a power consumption standpoint, a reduction of the fan speed is most efficient. Inlet vanes save some power and dampers save the least. From the standpoint of first cost, dampers usually are the lowest in cost. In some installations adjustments of volume are desirable at various times during the day or continuously. In others an increased supply of air in summer over that needed in winter is demanded. The

demands of each case will dictate which type of control is most desirable. Where noise is a factor, lowering the fan speed if possible is preferred as a control means, because of the resulting reduction in sound level.

MOTIVE POWER

Heating, ventilating and air conditioning fans are usually driven by electric motors, although other prime movers may be used. The small sizes of fans, and especially those operating in the higher speed range, are equipped with direct-connected motors. For larger size fans and those operating at lower speed V-belt drives are generally used.

In selecting the size of motor for operating a fan, it is advisable to select at least the standard size next larger than the fan requirements. Direct-connected motors do not require so great a safety factor as belted units. Justification for liberal power provision exists only in systems where it is possible that larger volumes of air may be required at intervals and made available by use of by-pass dampers, thus greatly reducing the system resistance. If such a system includes a fan with forward-curved blades, it would be necessary that the motor be sized for the maximum volume and duty. If such a system includes fans with backward-curved blades, the volume peak would not make it necessary to provide additional motor power. In selecting fans for such a system, sound ratings should be given careful consideration.

Where a system is constant, and has no provision for volume change that would materially reduce the resistance, and when the resistance calculations are reasonably accurate, there is no necessity for too liberal a motor allowance, even where fans with forward-curved blades are used, if the fan has been properly selected. Fig. 4 shows that the system resistance varies as the square of the volume and the fan static pressure varies approximately inversely as the volume, thus greatly offsetting the trend toward both increase in air delivery and motor load. Reference to Fig. 4 indicates that there is no justification for allowing large spare motor capacity, and it is generally more economical to operate motors well loaded.

ATTIC FANS

Attic fans are used during the warm months of the year to draw large volumes of outside air through a house and offer a means of using the comparative coolness of outside evening and night air to lower the inside temperature.

Because the low static pressures involved are usually less than $\frac{1}{8}$ in. of water, disc or propeller fans are generally used instead of the blower types. The fans should have quiet operating characteristics, and they should be capable of giving 20 to 30 room air changes per hour in northern areas. In the South usual specification requires one air change per minute which provides appreciable air movement in addition to lowering the inside air temperatures[4].

Types

Open attic fans are units in which the fan is installed in a gable or dormer of the attic and one or more grilles are provided in the floor of the attic, permitting air to flow from the hall below. Outdoor air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged outside by the fan. An attic stairway may be used in place of the grilles. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fans are units in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Outdoor air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings.

Another version of the attic fan is the *window fan* for use when attic application is not feasible or no attic is available. Supplied with a perforated or expanded metal enclosure and mounted in either the upper or lower window section, this fan is easy to install or transfer.

The locations of the fan, the outlet openings, and grilles should be selected after consideration of the room and attic arrangements in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind as in the case of the boxed-in fan, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

The window fan may be located in a hall or an unused bedroom. Noise of operation is more of a problem with the window fan than with the attic type, although care should be taken to locate either type of fan so that occupants are not disturbed.

These fans range in capacity from 3,000 to 30,000 cfm. The window type usually does not exceed 8000 cfm, while the most generally used attic type ranges from 8000 to 16,000 cfm. Power consumption is under 50 watts an hour per 1000 cfm of rated output for the 8000 cfm fan and larger, while the watts input for smaller fans is greater than this figure. Improved results can be secured with the window fan by closing off parts of the house where ventilation is not desired.

REFERENCES

[1] See Standard Test Code for Centrifugal and Axial Fans, 3rd Edition, 1938, Bulletin No. 103, *National Association of Fan Manufacturers*.

[2] Recommendations adopted by the *National Association of Fan Manufacturers*.

[3] Supplement No. B to Form X-12, *National Association of Fan Manufacturers*.

[4] Comfort Cooling with Attic Ventilating Fans, by G. B. Helmrich and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 155). A.S.H.V.E. RESEARCH REPORT No. 979—Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 167). A.S.H.V.E. RESEARCH REPORT No 1198—The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and A. F. Poor (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 145). The Installation and Use of Attic Fans, by W. H. Badgett (Agricultural and Mechanical College of Texas, *Bulletin* No. 52, 1940).

BIBLIOGRAPHY

Mine Ventilation, by J. J. Walsh (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 659).

Fan Blower Design, by H. F. Hagen (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 175).

The Specific Characteristics of Fans, by M. C. Stuart and J. B. Lusk (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 57).

Non-Dimensional Fan Characteristics, by H. Carlton Moore (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1937, p. 580).

Fans, by Theodore Baumeister, Jr. (McGraw-Hill).

Fan Engineering, Buffalo Forge Co.

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932 (John Wiley & Sons).

The Centrifugal Fan, by Frank L. Busey (A.S.H.V.E. Transactions, Vol. 21, 1915, p. 43).

The Theory and Performance of Axial-Flow Fans, by L. S. Marks and J. R. Weske.

The Axial Flow Fan and Its Place in Ventilation, by W. R. Heath and A. E. Criqui (A.S.H.V.E. Transactions, Vol. 50, 1944).

The Centrifugal Pumps and Blowers, by A. H. Church (John Wiley & Sons).

CHAPTER 33
AIR CLEANING DEVICES

Filters, Dust and Lint, Viscous Impingement Type Filters, Dry Air Filters, Electric Precipitators, Air Filter Performance, Testing, Selection and Maintenance, Filter Installation, Adsorption of Vapors, Dust Collectors, Degree of Dust Removal Required, Factors Affecting Selection, Types and Application of Dust Collectors, High Voltage Electrostatic Precipitators, Wet and Centrifugal Collectors, Settling Chambers, Testing Methods

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AIR cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet various air cleaning requirements. Degree of removal required, quantity and characteristics of the contaminant to be removed, and conditions of the air or gas stream will all have a bearing on the device selected for a given application. Definitions and a discussion of contaminant characteristics are given in Chapter 10, together with some consideration of their origin.

Air cleaning devices are divided in this chapter into two basic groups: Air Filters, described in Part I and Dust Collectors described in Part II.

Air filters are designed to remove dust concentrations such as are found in outside air, and are employed in ventilation, air conditioning, and heating systems where dust content seldom exceeds four grains per thousand cubic feet of air.

Dust collectors are designed for the heavier industrial loads encountered in local exhaust ventilation where dust content (loading) ranges from 100 to 20,000 grains per thousand cubic feet of air. This heavy loading from industrial dust control systems is often overlooked. Because of it air cleaning devices in the air filter group can seldom be used for control of process dust in industrial applications.

PART I—AIR FILTERS

Air filters are installed in ventilating systems to remove the airborne dusts which tend to settle out in the still air of the ventilated space and become a nuisance. Such dust comes under the classification of temporary atmospheric impurities listed in Fig. 1, Chapter 10, and includes most of the pollens, house dust, and similar allergens which motivate attacks upon persons of allergic sensitivity[1].

The air filters classified as viscous impingement types are widely used in general ventilating systems and will usually justify their cost through a reduction in housekeeping costs in the ventilated space. For industrial applications and in locations where the larger part of the atmospheric impurities are of fine particle size such as smoke and fumes, more effective dry filters and electric precipitators are proving economical. By use of combinations of filters advantage may be taken of specific characteristics of the various types of filters.

There is at the present time no generally accepted standard test method for measuring the cleaning efficiency of an air filter. Air filters are usually

selected on the basis of experience or judgment guided by the results of various test procedures.

A description of various types of air filters available commercially will be found under *air filters* in the catalog section of The Guide.

DUST AND LINT

Airborne solid matter from the viewpoint of air filtering is considered as lint and dust. Some lint originates outdoors, as animal hair, vegetable fibers, etc., but much is generated within buildings by the wear and brushing of fabrics in the form of clothes, draperies, carpets, etc. Lint is comparatively easy to capture in an air filter because of its comparatively great length. So far as air filter performance and testing are concerned, lint is chiefly important because of its tendency to impede or stop the flow of air through the filter. In general, lint, if not captured, will accumulate in corners and under furniture in a building in areas of slight air motion and in some cases may seriously obstruct heating and cooling coils. Dust settles, or is precipitated by heat or air motion, upon furniture, fixtures and walls and the only satisfactory treatment is washing or re-painting. Dust is more difficult to capture than lint, and, obviously, small particles are more difficult to capture than large ones. The air cleaning problem is complicated by the vast difference in size of dust particles, the range of which is shown in Fig. 1, Chapter 10.

As a general rule, the removal of the coarser dust particles and lint from the ventilating air produces tangible results in so far as cleanliness in a house or building is concerned, because much of the finer dust remains suspended in the air and is removed from the building by the circulating air. However, since some of the fine dusts are undoubtedly deposited by means other than settling, such as electrical or thermal precipitation[2,3] and by contact, the ability to remove small particles is desirable in an air cleaner if it can be obtained at not too great a cost.

VISCOUS IMPINGEMENT TYPE FILTERS

The medium in a viscous impingement type filter is usually a fiber pack for non-automatic types or a series of metal plates for automatic self-cleaning types. In either case, the medium is treated with a viscous substance, often an oil or grease, called the *adhesive* or the *saturant*, intended to retain dust particles which come in contact with it. Also, in either case, the arrangement is such that the air stream is broken up into many small air streams and these are caused to change direction abruptly a number of times in order to throw the dust particles, by momentum, against the adhesive. Several desirable characteristics of an adhesive for air cleaners of this type are: (1) Its surface tension should be such as to produce a homogeneous film or coating on the filter medium, (2) The viscosity should vary only slightly with normal changes of temperature, (3) It should prevent the development of mold spores and bacteria on the filter medium, (4) The liquid should have high capillarity, or ability to wet and retain the dust at all operating temperatures, (5) Evaporation should be slight, (6) It should be fire resistant, (7) It should be odorless.

Various fibrous materials have been used as filtering media in unit filters of the viscous impingement type. These include glass fiber, steel wool, similar *wool* of non-ferrous metals, wire screen, animal hair, hemp fibers, and other materials. In such filters, the medium is often packed more densely on the discharge than on the approach-side in order to increase the dust holding capacity. This results in a selective arrestance of dust with

the larger particles nearer the approach face. The arrangement also permits some penetration of lint into (but not through) the filter, so that the amount of lint which can be tolerated on the filter is also increased. Due to plane surface area the viscous impingement type filter, however, may be inferior to some dry types if the air carries a high percentage of lint.

The resistance of air filters obviously increases with the rate of air flow through them. Face velocities of about 300 fpm and resistances in the range from 0.1 to 0.2 in. water, when the device is new and clean, are usual for ventilation system filters. Special filters with low resistances are available for use with gravity warm air furnaces and for other applications where only low pressure is available.

The resistance of these filters increases with dust or lint loading and it is the resistance due to this cause which ordinarily necessitates servicing. The rate of loading obviously depends upon the amount as well as the kind of dust in the air and for this reason periods between servicing cannot be predicted. Manometers are often installed to indicate the pressure drop across filter banks and they serve to indicate when the filter requires cleaning. The pressure drop tolerated differs between operators and system designs. The resistance of a filter bank can be kept desirably low by periodically servicing some but not all of the units in the bank at one time.

The method of cleaning viscous impingement unit filters differs for different types of filters and kinds of dust. Much dry dust or lint can often be removed by rapping the filter.

Throw-away filters are constructed of inexpensive materials and are designed to be discarded after one use. The frame is frequently a combination of cardboard and wire.

Cleanable types usually have metal frames. Various cleaning methods have been recommended including: air jet, water jet, steam jet, washing in kerosene, and dipping in an oil. The latter may serve both to clean the filter and add the necessary adhesive.

Automatic Viscous Filters

In an automatic air filter, means are provided to remove the dust from the medium mechanically. Automatic filters with moving cloth media have been constructed but are not now in wide use.

The medium in a typical automatic filter at present consists of a series of specially formed metal plates mounted on a pair of chains. The chains are mounted on sprockets located at the top and bottom of the filter housing, so that the filter medium can be moved as a continuous curtain up one side and down the other side of the sprockets. The arrangement is such that, at the bottom, the medium passes through a bath of special oil which both serves to remove the dirt from the plates and acts as an adhesive when the cleaned plates next pass through the air stream. The plates forming the filtering medium or curtain usually overlap each other and due to their special shape many small air passages are formed between them. These air passages turn abruptly one or more times in order to give the impingement effect.

An electrically driven rotating device is usually supplied with an automatic filter. The device may be set to move the curtain periodically, or a special switch, actuated by pressure drop, may be used to govern its motion. In operation, the resistance of an automatic filter will remain approximately constant as long as proper operation is obtained. A resistance of $\frac{3}{8}$ in. water at a face velocity of 500 fpm is typical of this class.

DRY AIR FILTERS

The media in such filters are usually fabrics or fabric-like materials. Media of wool felt, cotton batting (both glazed and unglazed), cellulose fiber and other materials have been used commercially. The medium in a filter of this class is usually supported by a wire frame in the form of pockets or V-shaped pleats in order to increase the area exposed to the passage of air. A 2 ft square unit may contain from 15 to 30 sq ft of medium.

Dry air filters are likely to have a comparatively high lint-holding capacity on account of the large area of medium used. Wool felt media are troublesome to clean when impregnated with greasy dust and they are too expensive to discard frequently. Both vacuum cleaning and dry cleaning have been used for reconditioning wool felt filters.

ELECTRIC PRECIPITATORS

The fact that a particle exposed to an electric field will assume a charge

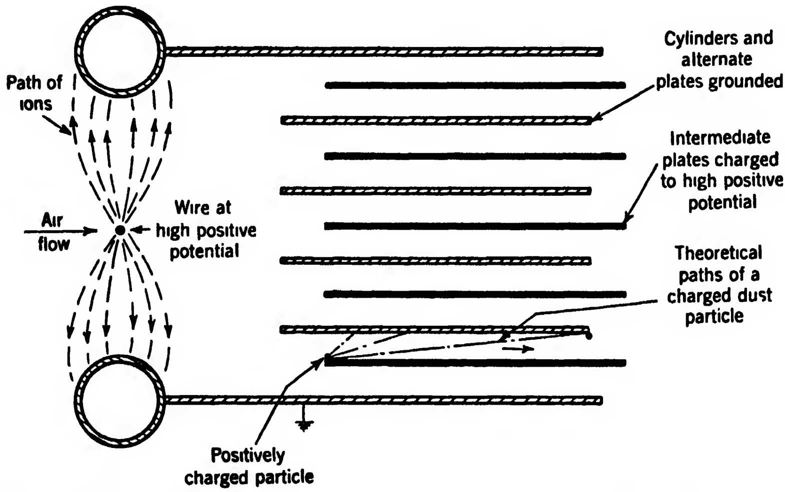

Fig. 1. Diagrammatic Cross-Section of Electrostatic Precipitator

and migrate toward one of the electrodes has been utilized for some years in boiler plants as a means of smoke abatement. The same principle has been used in equipment developed for air cleaning in air conditioning without generating ozone in intolerable quantities. The air stream in a precipitator passes first through a relatively high-tension electric field, known as the ionizing field, and then through a secondary field where the precipitation of the dust occurs. The arrangement is as shown in Fig. 1.

In a typical case, a potential of 12,000 volts may be used to create the ionizing field, and some 5000 volts between the plates upon which the precipitation of dust occurs. These voltages, which are capable of shock to personnel similar to that of a spark plug, necessitate some safety measures. A typical arrangement provides means for automatically making the unit inoperative when a door to the precipitator is opened. To resume operation the procedure necessitates closing the door and turning an electric switch, the latter of which should be located at a reasonable distance from the equipment. The voltages necessary for the operation of the precipitator are usually obtained from an alternating current building service line

by means of a step-up transformer. Precipitation with alternating current is possible but is not nearly so effective; so the current is usually rectified by means of vacuum tubes. The transformer and tubes are collectively termed the power pack.

Only a very small amount of electric energy is necessary to operate an electric precipitator and the resistance to air flow through the device is practically negligible. Some care is necessary in arranging the duct approaches on the entering and leaving sides of precipitators to assure that the air flow is distributed uniformly over the cross-sectional area. The efficiency of the precipitator is sensitive to air velocity and the device itself has much less tendency to rectify the air stream than filters, which have much higher resistances.

Electric precipitators are available in both automatic and non-automatic types. The plates of non-automatic precipitators are commonly coated with a light oil as an adhesive. Cleaning is accomplished with a water hose and, for this reason, the bottom of the equipment is made water tight and provided with a drain. In one automatic type, precipitation units are mounted on chains and are alternately dipped in oil and exposed to the air stream with an action similar to that of an automatic impingement filter. An arrangement of sliding contacts maintains the necessary electric circuits.

AIR FILTER PERFORMANCE AND TESTING

Air filters are generally rated in terms of the total air flow for which they are designed expressed in cubic feet per minute. Face velocity is defined as the average velocity of the air entering the filter, and it is determined by taking the air flow and dividing it by the area of the duct connection to the cleaner in square feet. Filters are often rated at a face velocity in the range of 250 to 500 fpm. Resistance to air flow is usually measured in inches of water column. The resistances of filters when new and clean and when operated at rated capacity are generally available from the manufacturer (see *Catalog Data Section*).

The ability of air cleaners to clean air is called the efficiency or the arrestance, and may be denoted by the symbol E. The efficiency of an air cleaner differs with the size and nature of the dust on which the cleaner operates. The efficiency of an air cleaner, algebraically expressed, is:

$$E = \frac{D_1 - D_2}{D_1} \tag{1}$$

where

D_1 = amount of dust per unit volume in uncleaned air.
D_2 = amount of dust per unit volume in cleaned air.

Several methods have been investigated for evaluating D_1 and D_2. The particle count method is not used for efficiency evaluation except in investigation of filter performance on specific particles such as pollen or on certain industrial dusts harmful to health. Dust particles can be captured on microscope slides by means of one of the various kinds of impingement devices. The process is useful if an inspection and analysis of dust is desired, but particle counting is not sufficiently precise for evaluating the efficiency of a cleaner operating on a heterogeneous dust.

The weight method of evaluating efficiency has found wide utility and was recognized by the AMERICAN SOCIETY OF HEATING AND VENTILATING

ENGINEERS and incorporated in a code[4]. For this test, a known weight of a prepared dust is injected into air supplied to the filter and the quantity of dust in the cleaned air is determined by extracting and weighing the dust from a known volume of the cleaned air. Dust extraction from the air is accomplished by drawing the air through a porous crucible or thimble by means of a high vacuum.

The *dust-spot* or *blackness* test for cleaner efficiency was developed at the *National Bureau of Standards*[5]. The test consists of drawing samples of cleaned air and of uncleaned air through filter papers simultaneously. The ratio of the areas of paper through which the air samples are drawn and the ratio of the amount of air drawn through the papers are adjusted during successive trials to yield spots of approximately equal blackness on the papers. The ratios of the areas and of the volumes of the air samples are then indicators of the filter effectiveness. A special photometer is provided for comparing the blackness or opacity of the papers by transmitted light. For tests of ordinary air filters by this method, a dust is injected into the air stream. The dust consists of precipitated smoke particles from a Cottrell precipitator used in a local power plant for smoke abatement. For

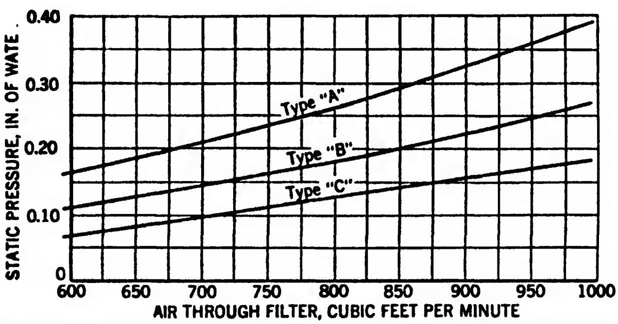

FIG. 2. RESISTANCE TO AIR FLOW OF TYPICAL UNIT AIR FILTERS

tests of electrostatic air cleaners, no dust is added to the air. Tests are commonly made with the dust existing in the air at the location of the installation on a clear day. Some specifications for this type cleaner have required that in dust spots of equal area the downstream spot shall not be any dirtier than the upstream spot when 10 times as much air is drawn through it as is drawn through the upstream spot. When this condition is met the cleaner is said to have an efficiency or arrestance of 90 per cent or better on atmospheric air.

Dust-holding capacity is defined as the amount of dust which a filter can retain and have a resistance less than some arbitrary value. The term applies only to non-automatic air cleaners. Determination of dust-holding capacity is an objective of each test under the A.S.H.V.E. Standard Code[4]. Curves are obtained during such tests to show the relation between dust load and resistance.

Fig. 2 indicates the general range of resistance to air flow through unit air filters. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work; and Type C is a low resistance unit, for use where low resistance is the important factor and maximum cleaning efficiencies are not essential.

At the *National Bureau of Standards* two injectors are provided on the

air cleaner testing apparatus. One injector is used to contaminate the air stream with Cottrell precipitate, previously described. This dust is used to make both efficiency determination and dust-holding capacity tests. The other injector contaminates the air stream with cotton linters with which lint-holding capacity tests are made. The curves in Fig. 3 illustrate the difference in the characteristics of two filters, one a viscous-impingement type and the other a dry filter with a cellulose fiber medium. The two injectors can be operated either separately or simultaneously. A total dust deposit of 4 per cent cotton linters and 96 per cent Cottrell precipitate gives a deposit on a filter closely resembling those that occur in Washington, D. C.

SELECTION AND MAINTENANCE

For removing fine dust or liquid particles which show no gravitational settling tendency electrical precipitators are highly effective in air cleaning

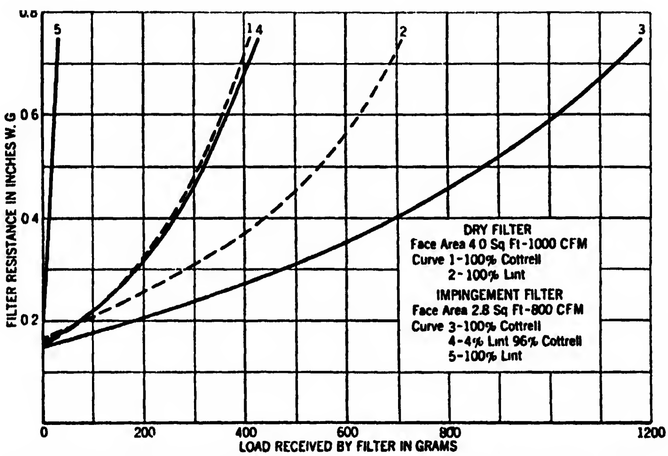

Fig. 3. Dust and Lint Holding Capacity of Two Filters

applications. The electrostatic equipment is comparatively expensive to install and maintain. However, in many applications where the cleanest air possible is needed, this expense is justified by the results obtained with properly installed and operated electrostatic air filters.

The advantage of the automatic impingement type filter consists in the small amount of attention which it requires. Such devices are therefore to be recommended where labor is scarce or where reliable and frequent attention to filters cannot be assumed. This type of equipment is not any better in dust arrestance than some unit filters, and it ranks next to precipitators in first cost.

Unit filters constitute the majority of air cleaners now in use, and some choice is possible between the types available. Where lint in an eminently dry state predominates, a dry filter obviously may be preferable to other types because of its lint-holding capacity. If the lint is greasy or if oil vapor exists in the air, the dry filter, if it is of the cleanable type, may be troublesome, since grease tends to make it difficult to clean. The cleaning difficulty is avoided if a throw-away type of medium is used. Some dry filters are capable of high efficiencies, compared to other unit filters on fine

particles, but their dust-holding capacity for such dust may be inferior to that of the viscous impingement type.

Viscous impingement unit filters represent the general type of air cleaner now in use. They have approached standards in size and their over-all dimensions are small when compared with their ratings.

Throw-away units are often installed in series so that the one in front, which usually becomes plugged with lint, can be discarded after which the downstream unit is moved to the front and replaced by a new unit.

Viscous impingement unit filters do not have efficiencies as high as can be expected with some other types of unit filters, but their first cost and upkeep are generally lower, whether of the cleanable or the throw-away type. They require more careful attention than the automatic oil type if the resistance is to be maintained within reasonable limits.

FILTER INSTALLATION

Many air cleaners are available in units of convenient size for handling when installing, cleaning or replacing. Such units are usually designated as filters or unit filters. A typical unit filter may be 20 in. square and from one to several inches thick, depending on the manufacture and proposed use. In large systems, a number of such units are installed adjacent to each other and collectively called a bank of filters.

Air cleaners are commonly installed in the outdoor air intake ducts of buildings and, often, in the recirculating air ducts as well. Cleaners are logically placed ahead of heating or cooling coils and other air conditioning equipment in the system to protect them from dust. The character of the dust arrested by the filters in an air intake duct is likely to be mostly particulate matter of a greasy nature, while lint may predominate in dust from within the building.

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

Failure of air filter installations to give satisfactory results can in most cases be traced to faulty installation or improper maintenance or both.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 per cent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.

2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.

3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.

2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.

3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.

4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.

6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.

7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.

8. Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

Safety Requirements

An investigation of safety ordinances should be made by the engineer when the installation of an air cleaner of any considerable size is contemplated. It is possible that combustible filtering media may not be permitted in accordance with some existing local regulations. Combustion of dust and lint on a filtering medium is possible, though the medium itself may not burn.

ADSORPTION OF VAPORS OTHER THAN WATER

Many of the foreign gases in the atmosphere are selectively adsorbed by charcoal. Included are many of the organic gases, such as those emanating from animals and people, some of the gaseous constituents of combustion, alcohols, ketones, esters, and gaseous products of putrefaction.

Charcoals differ widely in their adsorptive capacity. Those which have marked adsorption characteristics, such as properly prepared coconut shell charcoals, are sometimes called *activated charcoals* or *activated carbon.* These materials can adsorb approximately 50 per cent of their own weight of many organic gases at 70 F. The charcoal may be used for a long time by reactivation at high temperatures, under which condition it gives up the adsorbed gases. Temperatures of approximately 1000 F are desirable for reactivation. Charcoals for use in air handling systems should be able to stand physical handling, including reactivation, without excessive loss by breakage or dusting.

As applied in air handling systems, the charcoal is placed in perforated metal containers which are grouped in frames and set in the air stream. The percentage removal of an organic gas, such as carbon tetrachloride, is 95 per cent or above when placed in intimate contact with the carbon at 70 F. In commercial apparatus there may be a by-pass effect which depends on the physical arrangement of the charcoal containers. This by-passing reduces the percentage removed in the total gas passing through the adsorber. Resistance to air flow is usually selected within the general range of resistance of impingement filters.

The required quantity of recirculated air to be treated is determined by the requirements for contaminant-free air, minus the outdoor air, divided by the fraction denoting the percentage removal of the gas in question in the adsorber bank which is to be used.

Adsorbers may be applied to reduce objectionable gases entering through

the outdoor air inlet. They may also be used to reduce the odors caused by exhausts from processing. Adsorber beds, in all cases, should be protected from dust, free oil and grease.

PART 2—DUST COLLECTORS

Industrial development and growth of industrial areas have had a cumulative effect upon the problem of dust control. Not only has the atmosphere in many cities become more polluted, but the intensity of pollution at the points of control has increased. Accompanying this increase in pollution there has been a growing consciousness of the need for more effective air cleaning among housewives, store managers, industrialists and legal inspectors and this has resulted in increasing severity of regulations pertaining to collection of dust and contaminants.

Air cleaning for the supply system is usually accomplished by means of some type of air filter. To prevent escape of industrial dust into the atmosphere some type of dust collector is required. An industrial air cleaning installation is designed to perform one or more of the following functions:

1. Prevent a nuisance or physical damage to a plant or adjacent property.
2. Prevent re-entry of contaminants to working spaces.
3. Reclaim usable material.
4. Reduce fire, explosion or other hazards.
5. Permit recirculation of cleaned air to working spaces.

DEGREE OF DUST REMOVAL REQUIRED

Minimum standards of dust removal required of a dust collector may be established by local or state regulations, prepared by municipal smoke abatement or health departments or by state labor or health departments. Regardless of minimum standards, it is good practice to install the most effective collection equipment available in the light of practical operation features and installation and equipment costs. This is warranted because the required degree of effluent removal is increasing continually. Public nuisance complaints often occur even when the effluent concentration discharged to the atmosphere is below the permissible limits of concentration and visibility. Plant location, contaminants involved and meteorological conditions of the areas must be evaluated in addition to existing regulations or codes of good practice.

Air cleanliness must be of the highest order where cleaned air is recirculated to the workroom. Such recirculation is considered poor practice and is prohibited by many states where toxic materials are involved except for those cases where discharge to atmosphere is impossible or decidedly impracticable. Where air is recirculated air contamination must not exceed the established maximum allowable concentrations listed in Chapter 10. Usual requirements are a fraction, often $\frac{1}{3}$ to $\frac{1}{2}$, of this standard, depending on: State involved, air quantities recirculated in relation to the cubical content of working space, and the presence of other exhaust systems discharging to the atmosphere.

FACTORS AFFECTING SELECTION

Selection of a dust collector for a given application requires an evaluation of:

1. Dust load and particle size of the contaminant.
2. Degree of removal required.

3. Conditions of air or gas stream with reference to temperature, moisture content, and chemical composition.

4. Characteristics of contaminant whether corrosive, sticky, packing, and its specific gravity, particle size and shape.

5. Methods of disposal or salvage that meet the conditions imposed by material, process, or plant location.

With the range of variables that must be considered, selection of proper dust collection equipment is based largely on experience, and manufacturers of such equipment should be consulted for their recommendations.

TYPES AND APPLICATION OF DUST COLLECTORS

Dust collectors are available in a multiplicity of designs, frequently involving more than one principal of operation. Basic principals of operation are described in following sections. Manufacturers of various types can be found in the catalog section of THE GUIDE, or in handbooks featuring such equipment.

Electrostatic Dust Precipitators

Electrostatic dust and fume collectors differ materially from the low voltage designs described in Part I of this chapter, although the principles, methods of operation and efficiencies are similar. For dust collector loads, it is obvious that more severe demands are made upon methods of cleaning the collector, disposal of dust accumulations and servicing practices. Low voltage cleaners to date do not have sufficient inherent dust holding capacity. One exception in the field of exhaust systems is that of the oil mist collector, which functions satisfactorily with the conventional low voltage type electrostatic precipitator.

The *heavy duty dust collectors* employ an assembly of parallel collector electrodes of various constructions including corrugated plate, rod curtains or perforated plate. Air flow is usually horizontal although special construction may permit vertical air flow. The earlier high voltage collectors were made in the form of vertical pipes, and are still used for high operating pressures and for *wet collector* designs where water continuously flows downward inside the pipe walls of the collector electrode. The negative discharge electrodes or rods are accurately centered between the usual 9 in. collector electrode spacing, the latter being of positive charge. Precipitation occurs in a single stage, wherein ionization and collection are carried on simultaneously throughout the unit and depend on high potentials of 60,000 to 75,000 volts. The two-stage type which has the added primary stage of ionizer and receiving electrodes, however, fulfills some needs better than the single stage type.

High efficiencies are obtained by allowing suitable time for contact in the collector zone and by proper ratio of air flow velocity to that of transverse velocity of the negatively charged particles toward the positive charged collector plates. Air velocities vary from 4 fps to 8 fps with negligible pressure drops.

Attention should be directed to the prohibition of air recirculation to occupied spaces. High voltage cleaning equipment produces ozone in excessive quantities and, in addition, develops oxides of nitrogen.

Collector electrode cleaning is conveniently managed by a rapping device, either electrical or pneumatic, without interrupting operation. The dry plate surfaces release the dust readily into hoppers below the plates.

For longer than two decades electrical precipitators of the heavy duty high voltage type have been used for the control of hazardous materials

and for nuisance abatement. They have been applied to flue gases from cement kilns, smelters and paper plants; to exhaust systems serving crushers, grinders and conveyors; to chemical and metal working plants; and to many pulverized-fuel, fly-ash applications. To date, precipitators have been used principally for elevated temperature installations and dry products that are free of condensation or dampness.

Fabric Collectors

Dust collectors in this group, often known as *cloth dust arresters* or *cloth filters*, remove dust by passing air at low velocity thru a filter material. Cotton cloth is the usual material although wool, glass, asbestos, and metal screen are sometimes employed. Filter velocities depend on dust concentration, particle size, and permissible vibration interval. Normally at about 3 fpm, they often are reduced to 1 fpm or lower where heavy dust loads of very fine material are involved. Velocities in excess of 6 fpm are seldom used except in automatically vibrated sectional collectors where velocities as high as 20 fpm are frequently employed.

Collection efficiency is high even for low micron dust sizes when the collector is properly maintained. As collected material builds up on filter surfaces, increasing the resistance to air flow, such a system must be stopped at 4 to 8 hour intervals so that the dust load can be vibrated from the filter surfaces to reduce the pressure loss.

Filter cloth is supported in the form of envelopes or bags in a suitable steel housing. Space requirements are quite large, generally necessitating an outdoor location. Pressure drop is normally 2 to 4 in. water column. Material is collected dry. Fabric arresters are limited to applications where air is above the dew point as condensation packs the collected material with resultant high pressure drop and prevention of removal by vibration. Temperatures should not exceed 180 F for cotton, 200 F for wool.

Wet Collectors

In a wet dust collector, the contaminant is brought into contact with a liquid, usually water, for removing the dust from the gas stream. The various available designs represent combinations of methods that make cataloging according to principles, pressure drop, or efficiency difficult. Wet type dust collectors have the ability to handle high temperature and moisture laden gases. The collection of dust in a wetted form eliminates a secondary dust problem in disposal of collected material. However, the use of water may introduce corrosive conditions within the collector and freezing protection may be necessary if collectors are located outside in cold climates. Space requirements are nominal. Pressure losses and collection efficiency vary widely with design.

1. *Static Washers*. These units unlike most *air washers* are designed to handle heavy concentrations of dust. Both scrubber and eliminator plates (each having flooding nozzles) are employed in addition to the bank of sprays ahead of the scrubber plates and 2 banks of opposed sprays located ahead of the eliminators. A hopper-bottom tank with recirculating pump completes the assembly. Pressure drop is about $\frac{1}{2}$ to $\frac{3}{4}$″ of water. Spray towers can be placed in this same group, usually consisting of a tower structure with various nozzle arrangements and usually including an eliminator section at the top.

Wet glass cell washers have special sprays playing on filter cells filled with fiberglass or other filter media. Flooded eliminator plates are used to remove free moisture from the air stream. Pressure drop is about $\frac{3}{4}$″ of water and the length of the unit is comparable to that of a single stage air washer. Applications are usually limited to low dust concentrations.

2. *Packed Towers.* Collectors in this group are essentially contact beds through which gases and liquids pass either con-currently, counter-currently, or in cross flow and are used primarily for nuisance abatement of highly corrosive contaminants. The liquid usually enters at the top of the tower while the gases may enter at the top, at the bottom, or through an open side.

Water flow rates of 5 to 10 gpm per 1000 cfm (70 F volume) are distributed frequently through V-notched ceramic or plastic weirs. High temperature deterioration is avoided by use of brick linings permitting 1600 F gases direct from furnace flues.

Air flow pressure loss for four foot beds of irregular shaped materials such as ceramic saddles or coke range from 1½" to 3½" wg with respective face area velocities of approximately 200 to 300 fpm.

3. *Wet Centrifugal.* A number of designs utilize a combination of centrifugal force and water contact to affect collection. In designs of this group, collector is cylindrical in either the shape of a tower or with the axis horizontal. Air is introduced tangentially and frequently directed counter current to flow of water by baffles or directional plates. Water may be brought into contact with the dust particles by keeping collector surfaces washed by spray nozzles, by induced water picked up by the air or by fall of water due to gravity. Pressure losses range from 2½" to 6".

4. *Dynamic Precipitator.* This type uses water sprays within a fan housing and obtains precipitation of the dust particles on the wetted surfaces of an impeller with special fan blade shape. No external pressure drop is involved although mechanical efficiency is somewhat lower than the mechanical efficiency of standard exhaust fans.

5. *Orifice Type.* In this type the air flowing through the collector is brought in contact with a sheet of water in a restricted passage. Water flow may be induced by the velocity of the air stream, or maintained by pumps and weirs. Pressure losses vary from 1" or less in water wall spray booth collector designs, to from 3" to 6" in most industrial collector arrangements. Pressure losses as high as 20" are used with some collectors designed to collect very small particles.

6. *Disintegrator.* This type of unit generally consists of one or more stages and is largely used for cleaning producer, blast furnace, or other gases where the gas is to be used in engines and must be practically free of dirt. The spray is generally in the fan inlet and elimination is effected largely on the fan blades and also on the surfaces beyond.

A special two element fan is used, the air with its dust content and water spray enters one side of the wheel and is discharged from the *inlet* of the other wheel. As the air passes through the cyclonic chamber a high degree of scrubbing action takes place.

Relatively high pressure losses are encountered with resulting high horsepower requirements.

CENTRIFUGAL DUST COLLECTORS

Centrifugal collector design can be divided into four groups according to their effectiveness in removal of smaller dust particles.

1. *Cyclone Collector.* This type is commonly applied for the removal of coarse dusts from an air stream, as a precleaner to more efficient dry or wet dust collectors, or as a separator in product conveying systems using an air stream to transport material. Principal advantages are low cost and low pressure drop (¾ to 1½ in. water) but this type cannot be used for high efficiency collection of fine particles.

2. *High Efficiency Centrifugal Collectors.* These have been developed to obtain higher centrifugal force action on dust particles in a gas stream. Centrifugal force is a function of peripheral velocities and angular acceleration, and improvement in dust separation efficiency has been obtained by (a) increasing velocities through a cyclone shaped collector; (b) utilizing a skimmer or other design feature; (c) by using a number of small diameter cyclones in parallel, and (d) in some unusual applications by placing units in series.

While such collectors do not generally reach an efficiency on small particles equal to that of the electro-static, fabric or some wet type units, their effective collection range is extended appreciably beyond that of the conventional cyclone. Pressure losses of collectors in this group range from 3 to 8 in. water column.

3. *Dry Type Dynamic Precipitator.* In this type dust is precipitated by centrifugal

force upon specially shaped blades of an exhauster wheel, and then conveyed through a dust circuit in the fan casing to the dust storage hopper.

4. *Cinder Catching Fan*. This is a special type of collector in which removal of solids is effected by a slotted scroll. It is a fan unit which is often used for the dual purpose of dust removal and induced draft service.

SETTLING CHAMBERS

While it would be possible in theory to remove dust by settling in a large chamber when conveying velocities are reduced to the point where the particles would no longer be held in suspension, such devices have little practical application in dust collecting equipment. Extreme space requirements and the presence of eddy currents which may nullify the effective velocity, means that settling chamber type of collectors can be used only for removal of extremely coarse particles. Pressure drop should be $\frac{1}{4}$ to $\frac{1}{2}$ in. water column plus that necessary to accelerate the air motion to the required conveying velocity at discharge of the settling chamber.

TESTING METHODS

Methods of determining the performance of industrial air cleaning devices will depend upon the nature of the air contaminant, its quantity, the required accuracy of the test, and the type of air cleaning device. The technics used in collecting the samples are the same as those utilized in the field of industrial hygiene.

The tests may be facilitated by feeding uniformly a known amount of the material to be removed. The performance efficiency may be calculated from the amount of material introduced, the quantities of air involved, the amount of material intercepted and the quantity that escapes. When it is necessary to test the device under actual use, the material entering and leaving must be sampled simultaneously over a sufficiently long period of time to collect an adequate amount for analysis. If feasible, the quantity of material removed by the cleaning device during the test period should also be determined.

Unusual care must be exercised in collecting the samples to insure that the sampling areas selected are actually representative of the material entering and leaving the cleaning device. With gases, vapors and fresh fumes this problem is not great. On the other hand, mists, dusts and aged fumes may present considerable trouble. When the material is confined in a duct system, it is common practice to collect the sample along the center-line. The sampling tube is located parallel with and facing the flow. The sampling velocity should be approximately the same as the air velocity in the duct. With large ducts it may be desirable to make traverse tests to locate an optimum sampling position.

REFERENCES

[1] Bronchial Asthma and Allied Allergic Disorders, by S. S. Leopold and C. S. Leopold (*Journal of the American Medical Association*, March 7, 1925, Vol. 84, p. 731–734).

[2] Dirt Patterns on Walls, by R. A. Nielsen (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1941, p. 247).

[3] Industrial Dust, by Philip Drinker and Theodore Hatch (McGraw-Hill Co., New York, N. Y.).

[4] A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

[5] A Test Method for Air Filters, by Richard S. Dill (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 379).

BIBLIOGRAPHY

Design and Application of Oil-Coated Air Filters, by H. C. Murphy (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 73).

Operation and Maintenance of Air Filters, by W. G. Frank (*Heating, Piping and Air Conditioning*, May, 1931, p. 378).

Size and Characteristics of Air-Borne Impurities, by W. G. Frank (*Heating, Piping and Air Conditioning*, January, 1932, p. 35).

Fundamental Principles in the Design of Dry Air Filters, by Otto Wechsberg (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1933, p. 217).

The Economic Factors in Converting Recirculated Air for Ventilation, by H. E. Ziel and Henry Sleik (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1943, p. 367).

Operation, Maintenance of Cloth-Screened Dust Collectors, by W. F. Terry (*Heating, Piping and Air Conditioning*, May, p. 259, June, p. 304, 1933).

Testing and Rating of Air Cleaning Devices Used in General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

A.S.H.V.E. RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity Through Filter, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 415).

A.S.H.V.E. RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 339).

A.S.H.V.E. RESEARCH REPORT No. 1145—The Effect of Lint on Air Filter Performance, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 25).

A.S.H.V.E. RESEARCH REPORT No. 1169—Comparison of the Weight, Particle Count and Discoloration Methods of Testing Air Filters, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 29).

A.S.H.V.E. RESEARCH REPORT No. 1187—Economical Air Velocities for Mechanical Air Filtration, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 391).

A.S.H.V.E. RESEARCH REPORT No. 1218—Overloading of Viscous Air Filters During Accelerated Tests, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 437).

The Dust-Free Space Surrounding Hot Bodies, by H. H. Watson (*Transactions of he Faraday Society*, Vol. 32, 1936, Pt. 2, p. 1073).

A New Electrostatic Precipitator, by G. W. Penney (*Electrical Engineering*, January, 1937, p. 159).

Electrostatic Precipitation for Aircraft, by Howard E. Corbitt and Norman J. Clark (*Aero Digest*, December, 1940, p. 132).

Cleaning of Air and Gas by Thermal Repulsion, by Samuel C. Blacktin (*Journal Society of Chemical Industry*, *58*, 1939, p. 334–8, *59*, 1940, p. 153–4).

Pointers on Selecting Equipment for Industrial Gas Cleaning, by C. E. Miller (*Chemical and Metallurgical Engineering*, *45*, March, 1938, p. 132–5).

Electrical Precipitation, by W. A. Schmidt and E. Anderson (*Electrical Engineering*, *57*, August, 1938, p. 332–338).

Electrostatic Precipitation for Cleaning Industrial Gases, by H. W. Wagner (*Fuel Economist*, *10*, 1935, p. 895–9, 942–5, 971–3).

Dust Precipitation, by M. Pauthenier and Mme. Moreau-Hanot (*Electrician*, *113*, August 10, 1934, p. 187–9).

Alternating Current Precipitators for Sanitary Air Analysis: II. Acid Formation in Electric Precipitators, by W. G. Hazard and Tomoyoshi Ishikawa (*Journal of Industrial Hygiene*, October, 1932, p. 367–70).

Electrical Precipitation, by A. W. Simon and L. C. Kron (*Electrical Engineering*, *51*, February, 1932, p. 93–5).

The Mathematical Theory of the Cottrell Electric Precipitator, by A. W. Simon (*Iron and Steel Engineering*, *6*, 1929, p. 143–6).

Progress in the Art of Electrical Precipitation Since 1900, by P. E. Landolt (*Transactions American Electrochem. Society*, *51*, 1927).

Some Factors and Principles Involved in the Separation and Collection of Dust, Mist and Fume from Gases, by Evald Anderson (*Transactions American Institute of Chemical Engineers*, *16*, 1924, Pt. 1, p. 69–86).

Electrical Precipitation of Solids from Smelter Gases, by Ross B. Rathbun (*Transactions American Institute of Electrical Engineers*, *41*, 1922, p. 815).

Electrical Engineering Features of the Electrical Precipitation Process, by G. H. Horne (*Transactions American Institute of Electrical Engineers*, *41*, 1922, p. 808).

Characteristics of Unit Dust Collectors, by Arthur C. Stern, Jack Baliff, Arthur E. Perina, Robert Crowley, Benjamin Feiner and Arthur A. Urbano (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, May, 1946, p. 117).

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CHAPTER 34
AUTOMATIC CONTROL

Basic Types of Control, Types of Controllers, Actuating Devices, Actuated Controls, Residential Control Systems, Zone Control, Automatic Control Application, Control for Central Fan Systems, Panel Heating Control, Indicating and Recording Equipment

THE function of *automatic control*, as applied to the heating, ventilating and air conditioning industry, may be broadly subdivided into the maintenance of *temperature*, *humidity*, and *pressure*, within pre-determined ranges. It automatically coordinates the operation of the various controlled devices in proper sequence to produce the desired result.

BASIC TYPES OF CONTROL

Available automatic control equipment may be divided into four main groups depending on the primary source of power and can be classed as:

1. *A self-actuated regulator* is one in which all the energy necessary to operate a valve or damper motor is supplied by the responsive element or bulb. Temperature changes at the bulb result in pressure changes of an enclosed fluid which are transmitted directly to the valve or damper motor. Instruments of this type are available either with a rigid bulb or with flexible tubing from the bulb to the operating motor. The flexible tubing may be furnished in varying lengths and is generally protected by a flexible metal armor.

2. *Electrically operated* equipment utilizes electric current as a primary source of energy, its flow being regulated as required to operate motors, relays, or other controlled items.

3. In *pneumatically operated* equipment the primary source of energy is compressed air usually at a pressure of 15 to 25 psig. The flow of this air is proportioned as required to operate valves, dampers, relays, or other controlled devices.

4. In *hydraulically operated* equipment the primary source of power is a suitable liquid at a pressure of 15 to 25 psig or higher which is handled in the same manner as compressed air.

TYPES OF CONTROLLERS

The basic types of controllers which are available may be classed as:

1. *Two position* or *on-off controllers* are the simplest type and are clearly described by the name. With controllers of this type the valve or damper motor can assume only two positions, either open or shut.

2. *Proportional* or *gradual acting controllers* function to re-position the controlled device by small increments of travel to regulate the flow as the controller senses a slight change in the controller condition.

3. *Floating controllers* act to produce valve or damper movement whenever there is a deviation from the control point. Whenever the condition to be controlled is above the control point, the valve closes at a constant rate, and continues to close until the temperature returns to the control point. Below the control point the valve reverses its action and moves in the other direction until the control point is again reached.

4. *Automatic reset (or proportional plus reset) controllers* function to reposition a valve or damper by small increments of travel as in a proportioning controller. In addition, a mechanical device in the controller automatically and constantly resets the instrument to offset the normal drift (inherent in a proportional controller) between maximum and minimum load. The rate of reset is manually adjustable and must be set to meet the load requirements of the individual system.

Controllers may also be designated by types as: (1) a *non-indicating controller*, when it does not indicate the controlled condition and performs

the control function only; an *indicating controller*, when fitted with a pointer, thermometer, or gauge which indicates the controlled condition; a *recording controller*, when it is combined with a clock mechanism and chart which records the controlled condition.

ACTUATING DEVICES

The starting point of any control system is the thermostat, hygrostat, pressure regulator, or other mechanism which is sensitive to a change and responds in the desired manner.

Thermostats are usually of the room, duct or immersion types. Various types of thermostats found in common use are defined in the following paragraphs.

1. A *thermostat* is an instrument which is responsive to changes in temperature and initiates a force that repositions valves, dampers, etc. to maintain selected temperatures.

2. A *room thermostat* is usually mounted on the wall of the space to be controlled, with the measuring element arranged so that it is affected by the room temperature.

3. A *duct thermostat* is provided with fittings suitable for installation in duct work. The *insertion* type is equipped with a rigid bulb and is arranged so that the temperature responsive element or bulb extends through the wall of the duct. The *remote bulb* type is arranged so that the bulb and instrument head are connected by means of a flexible tube of the desired length. The bulb is inserted in the duct, and the head is located where it is accessible for adjustment and inspection.

4. An *immersion thermostat* is provided with fittings suitable for installation in a pipe line or tank where a fluid tight connection is required. Both insertion and remote bulb types are available. A union connection and separable socket when used permit removal of the bulb without draining the line or tank. The sockets may be of copper, stainless steel or other materials.

5. A *day-night* or *two-temperature thermostat* controls a heating or cooling source to maintain either of two selected temperatures. They may be indexed (set at desired control temperature) individually or in groups from a remote point by means of a manual or time switch.

6. A *summer-winter* or *heating-cooling thermostat* is similar to the Day-Night type except that both the temperature setting and action are changed by the indexing means. Such a thermostat could open a volume damper on a rise in temperature in summer and close the same damper on a rise in temperature in winter.

7. A *submaster thermostat* has its temperature setting raised or lowered a predetermined amount for a given change in some other variable. For example, the water temperature on a heating system may be raised as the outdoor temperature drops. A *master instrument* is used to reset a submaster thermostat and may be a switch, pressure controller, thermostat, or similar device. In the foregoing example the master thermostat would be located where it would respond to outdoor temperature and the submaster thermostat would be located in the pipe line of the heating system.

A *hygrostat* is a controller which is sensitive to changes in relative humidity, and is available in room and duct types. Where the controlled condition is below 20 per cent or above 80 per cent, or the temperature is above 100 F, selection of a suitable type and kind of hygroscopic element is essential.

A *pressure regulator* is a device which is sensitive to changes in pressure. It is available in types which control a *single* pressure or the *differential* type which maintains a pre-determined difference between two pressures. For pressures in duct work, *static pressure regulators* are available in the *differential* type. They are sensitive to changes of 1/100 in. of water.

ACTUATED CONTROLS

Thermostats, hygrostats, pressure regulators and other actuating devices obtain control of heating and cooling mediums, fuels, liquids, etc., by

actuating various control devices such as control valves, dampers, damper motors, relays, or controllers defined in the following paragraphs.

A *control valve* is designed to control the flow of fluids, and may be considered as a variable orifice which is repositioned by a motor operator as directed by a thermostat, or other controlling device.

1. A *normally open* or *direct acting valve* will assume an open position when all operating power is removed.

2. A *normally closed* or *reverse acting valve* will assume a closed position when all operating power is removed.

3. *Single seated valves* are designed for *tight* shut-off using appropriate disc materials for various pressure ranges.

4. *Pilot piston valves* serve a similar function on high pressure installations.

5. *Double seated* or *balanced valves* are designed for applications where tight shut-off is not required. They are not affected by varying inlet pressures or pressure differentials, and thus are widely used where these conditions exist.

6. A *three-way mixing valve* is fitted with a double faced disc, operating between two ports, and functioning to close one port as the other is opened. It is used to mix two fluids as required.

7. A *three-way diverting valve* is fitted with two discs which act to close one outlet port as another outlet port opens, and thus diverts the flow from the inlet port to either of the outlet ports.

Valve discs, poppets, and seats are available in various shapes to meet any desired flow characteristics with various materials as required by service conditions.

A *damper* is designed to control the flow of air or gases and is similar to a valve in this respect. *Single blade dampers* are generally restricted in size because of the difficulty of securing proper operation with high velocity air. *Multi-blade* or *louver dampers* can be furnished so that adjacent blades move in the same direction or in opposite directions. The *opposed blade type* gives better directional and flow characteristics than the parallel blade type.

For long life and trouble free operation, dampers should be constructed with heavy metal frames, blades of iron adequately braced, and ample bearing surfaces of non-corrosive materials. When fairly tight closing is desired felt may be glued and riveted on the edges and ends of the blades. Other materials for blades and frames are also used for special services.

A *damper motor* is repositioned by a controlling instrument, and is connected to the damper blades as required to give the desired movement. It can be mounted on the damper frames or mounted outside the duct and connected to an extended shaft on one or more damper blades. Suitable brackets are available for floor, wall, or duct mounting of the motor.

A *relay* is a device which uses an auxiliary source of energy to amplify or convert the force of a controller into available energy at a valve or damper motor. Various types of relays are designated as follows:

1. An *electro-pneumatic relay* when electrically energized starts or stops the flow of air as required.

2. A *pneumatic-electric relay* when affected by different air pressures starts or stops the flow of electrical energy as required.

3. A *switching relay* or *pilot valve* may be used to switch the operation of a controlled device from one controller to another; or to reverse the action of a controlled device in response to an impulse from a controller.

4. An *averaging relay* is affected by the forces from two or more controllers, and the resulting flow of energy is in accordance with the average of these forces.

5. A *positioning relay* has a direct connection to a valve or damper motor lever, and is affected by both valve or damper position and controller demand. It is re-

positioned by a thermostat or other controlling device, and is arranged to give a definite motor position for a given force from the thermostat without regard for motor hysteresis, friction, or pressure variations of the controlled fluid.

A *sequence controller* is used to operate two or more devices in a prearranged sequence. It is generally used in connection with refrigeration compressors, and may be arranged to prevent simultaneous starting in the event of temporary electrical shutdown or control medium failure.

Manual switches are available in the two-position or proportional types. *Two-position switches* change the flow of energy from one line to one or more other lines; or from one pair of lines to another pair of lines. *Proportioning switches* vary the flow of energy as determined by the manual setting of the switch.

RESIDENTIAL CONTROL SYSTEMS

The control equipment function in a residence may vary from the regulation of a coal-fired heating plant to the completely automatic control of an all year air conditioning system. Regardless of the type of heating or air conditioning system used, the control system should be selected carefully to insure safety and comfort of the occupants and also economy of operation.

Heating Unit Controls

Typical controls for the appliances used to supply heat in residences are as follows:

1. *Hand Fired Coal Burners.* The control of a hand fired coal burner for a boiler or furnace normally consists of a room thermostat operating a two position electric control motor which in turn opens the draft damper and closes the check damper on a demand for heat. The motor then closes the draft damper and opens the check damper when the thermostat is satisfied. A limit control on the boiler or furnace should be connected to the motor so that it may check the fire whenever a predetermined temperature or pressure has been exceeded. A manually operated basement switch is usually included on the motor so that the draft may be opened and the check closed when the boiler or furnace is being filled with coal.

2. *Coal Fired Stokers.* Domestic stokers are usually controlled by a room thermostat, a limit control, and a stoker relay. When the thermostat calls for heat, the relay causes the stoker motor to increase the flow of fuel and air to the burner to its maximum rate. When the thermostat is satisfied, the relay provides for a minimum flow of fuel and air to the burner to maintain the fire at its minimum rate. The limit control prevents the continuance of the maximum fuel rate if the temperature or pressure in the boiler or furnace exceeds a predetermined value and also stops the feeding of fuel if the fire goes out. Automatic ignition usually is not available and the firing of a stoker is normally on or off.

3. *Automatic Oil Burners.* Automatic oil burner controls normally consist of a room thermostat, a limit control, a combustion safety control, and a control relay. On a call for heat by the thermostat, the relay starts the oil burner motor which supplies oil and air to the burner. An ignition device consisting of an electric spark or a gas flame ignites the oil automatically. If for any reason the oil and gas mixture does not ignite, a time delay mechanism in the relay is operated by the combustion safety control after a predetermined length of time to cause the oil and air supply to be shut off. If the oil and air mixture ignites properly the burner continues to run until the thermostat is satisfied or until the limit control affected by the temperature or pressure in the boiler or furnace, stops the burner.

4. *Automatic Gas Burners.* The controls for an automatic gas burner usually include a room thermostat, a limit control, a safety pilot, gas pressure regulator and a gas valve (solenoid, motorized or diaphragm type). Upon a demand for heat at the thermostat, the gas valve is opened, admitting gas to the burner. The safety pilot ignites the gas which continues to burn until the thermostat is satisfied or until the limit control shuts off the gas valve. The limit control may also reduce (throttle) the gas flame as required to maintain a desired temperature or pressure of the heating

medium. If the pilot flame is extinguished for any reason either before or after the main gas valve is turned on, the safety pilot closes the gas valve, thus eliminating the danger of delivering gas to the burner without ignition.

5. *Electric Heating.* Electric heating has become popular in those areas where electric power is plentiful and inexpensive. The electric heating elements may be located in each individual room and turned on and off by thermostats in each room or the heat may be supplied by a central heating system. In the case of a central heating system the control is usually of the proportioning type which energizes from five to ten heating elements in sequence according to the demand for heat by means of a sequence controller consisting of a series of switches operated by a proportioning motor. A limit switch recycles the sequence controller if the furnace or boiler exceeds a predetermined temperature.

Limit Controls

A *high limit* control for steam consists of a pressure control having bellows, responsive to the boiler pressure, which breaks an electric contact when the steam pressure exceeds a predetermined point, thereby preventing the burner from delivering additional heat to the boiler. A low water cut-off should also be used to stop the burner if the water in the boiler drops to a dangerous level.

A high limit control for a hot water boiler consists of an immersion thermostat (usually equipped with a bi-metal helix) inserted in a well in the boiler. This control stops the burner when a predetermined water temperature has been reached in the boiler.

In a warm air system the high limit control is a thermostat including a bi-metal helix inserted in the bonnet of the furnace. It will shut off the source of heat when a predetermined furnace temperature is exceeded.

System Control

There are several types of system control in common use for residential applications. They are usually of the two position (on-off) type or of the proportioning type.

1. *Two Position (On-Off) Control.* The most simple type of domestic control is the type in which the room thermostat starts the burner or other source of heat when the temperature of the air at the thermostat falls below the thermostat setting and stops the source of heat when the air temperature rises above the setting. If forced warm air or forced hot water is used, the fan or circulator may be turned on and off at approximately the same time as the source of heat.

2. *Proportioning Control.* When a proportioning type of control is used, the flame of the burner may be varied or the burner may be cycled (started and stopped) frequently to provide for time modulation so that the heat input to the home is proportioned continuously to the heat loss from the home. The fan of a forced warm air system or the circulator of a forced hot water system may be run almost continuously, thereby providing for the constant flow of heat into the home. Such operation eliminates the cold drafts on the floor as caused by cold air dropping from cool walls and windows during the off period of an on-off system.

Room Thermostats

Room thermostats usually employ a bi-metal element or a vapor filled bellows as the temperature sensitive element. They may be of the plain or single temperature type, or of the day-night type providing for automatic night lowering and morning increase of the control point. The automatic setback type usually includes a clock mechanism which accomplishes this result. Opinions vary regarding the amount of fuel that can be saved by automatic setback. Tests made both in the Warm-Air Heating Research Residence and the I-B-R Research Home at the University of Illinois indicate that, on thermostatically controlled systems, a possible

fuel saving of from 7 to 10 per cent may be obtained by reducing the house temperature 6 to 10 degrees from about 10:00 p.m. to 5:30 a.m.*

In locating a room thermostat the following rules should be observed:

1. It should always be located towards the center of a relatively open room on the coolest rather than the warmest side of the building.
2. It should never be mounted on an outside wall or other cold surface, or where it is exposed to cold drafts from an outside door.
3. It should never be mounted where it will be affected by heat from a nearby warm surface such as chimneys, pipes or ducts in a wall, radiators; or by direct currents from a warm air register.
4. It should never be located where normal circulation of air is impeded by furniture or an opened door.
5. It should be located where it is safe from mechanical injury.

In a typical home, a satisfactory location for the thermostat can usually be found on an inside wall of the living room or dining room.

Air-Conditioning Systems

Year-round residential air-conditioning systems which provide for heating in winter and cooling in summer should be given the same consideration in selecting the control system as required for commercial air-conditioning systems described later in this chapter since the basic principles are the same and the final results must provide for the comfort of the occupants. Economy in first cost may result in both lack of economical operation and discomfort.

ZONE CONTROL

In residential heating, it is often desirable to divide the house into two or more zones for greater accuracy of control and comfort. Each zone may then be maintained individually at the desired temperature level. The division by zones should be based upon exposure and occupancy and the most common division is usually found to be:

1. Living section (Living room, dining room, den, etc.)
2. Sleeping section
3. Service section (Kitchen, pantry, servant's quarters, etc.)

Zone control for steam and hot water heating systems is employed where it is desired to control the heating effect of a multiplicity of radiators or convectors, located in various heated spaces, through the use of a single regulator. Under certain conditions, particularly in buildings of limited size, it is possible to consider the entire building as a single heating zone. In such cases, the zone regulator, or *master controller*, may operate directly the automatic firing equipment of the boiler or the reducing valve in the street steam main. In large buildings the demands of satisfactory temperature control, however, will make it necessary to sub-divide the heating system into suitable zones.

There are a number of factors to be considered in zoning, in order that heating requirements in a single zone will be approximately consistent throughout its extent.

1. Exposure may be a factor to be considered, with particular reference to prevailing winds, sun effect, and the shelter afforded by surrounding structures and topographical features.

*Save Fuel for Victory, University of Illinois, Engineering Experiment Station Circular Series No. 47, p. 31.

2. Occupancy may be a determining factor, in that the indoor temperature requirements for the activities carried on in various portions of the building may vary and the hours of occupancy likewise may differ.

3. The physical characteristics of the building will enter into the sub-dividing of the heating system into zones by reason of the fact that satisfactory temperature conditions throughout a single zone of given extent may not be enjoyed equally in buildings of dissimilar types of construction. Also, the height of the building and its horizontal extent and form are considerations which must be borne in mind.

4. The cost of the zone control equipment for such additional zones as might seem otherwise desirable often will influence the decision as to the final number of zones to be employed. In buildings of considerable size, accepted practice dictates that there shall be at least one zone for each exposure, although each exposure very possibly should be sub-divided vertically into two or more zones, for the higher structures. Also, the presence of two or more wings, having the same general exposure, may suggest the desirability of more restrictive zoning. In smaller buildings, and in those of larger extent where cost and other conditions limit the number of zones, a common compromise is to combine the North and West exposures in one zone, and the East and South exposures in a second zone. Frequently, when the street floor level is given over to public spaces or to activities which are markedly different from those carried on in the remainder of the building, it is advisable to provide a separate steam main, with conventional room thermostats in each individual area.

For steam heating systems, the radiator output may be varied proportionately to the changes in outdoor temperature, by any of a number of general methods. Those in most common use are:

1. Turning the steam on and off at appropriate intervals as dictated by temperature or time considerations, proportioned to the need for heating.

2. Varying the pressure of steam in the system in accordance with the demand for heat.

3. Throttling the steam pressure, at the demand of the controller, to allow flow through orifices in proportion to the heating requirements.

In hot water heating, the accepted practices are (1) to vary the temperature of the hot water supplied to the system or, (2) to vary the flow of hot water, both proportionately to the heating requirements.

The regulator for each zone usually is of a type which in some manner responds to the outdoor temperature and effect of sun and wind for the zone. Many of the available zone controllers are arranged in such a way as to be affected also by the temperature of the heating system and the indoor temperature in the zone. Whatever the mechanical features of the regulator, its function is to dictate the flow impulse, or rate of flow, in such a way as to maintain the desired indoor temperature in the zone, regardless of the fluctuations in the outdoor temperature. Provisions also may be made to maintain a predetermined low economy temperature in each zone during periods of non-occupancy; to facilitate quick warming-up following such periods; and to follow those portions of the daily cycle of control with normal heating effect during the occupancy period.

A control panel, at a central location, may be arranged so that manual switches for each zone may raise or lower the operating temperature, and time switches, if desired, may be provided for obtaining, automatically, any day-night or other predetermined control program which the operators of the building may desire. The characteristics of the regulators which are operated by the zone controllers depend upon which of the basic systems of zone control is employed in a given installation. Shut-off, mixing or throttling valves and various forms of devices to reset or pilot the action of reducing valves and to control firing means, are some of the more common regulators which are used to control the flow of steam or hot water,

under the command of zone controllers. The characteristics of these regulators usually are determined by the manufacturer of the type of controller which is selected.

AUTOMATIC CONTROL APPLICATION

Some of the considerations affecting the selection of automatic controls for applications are given in the following paragraphs which describe controls and operation for various types of units.

Unit Heater Control

Two-position (on-off) control by means of a room thermostat is the standard method of control for unit heaters. A limit control should be incorporated to prevent operation of the unit heater fan motor when steam or hot water is not available. The limit control can be a surface thermostat or pressure control. Where there is no possibility of drafts and continuous air circulation is required, the unit heater fan motor may operate continuously. In this case, a room thermostat controls a valve (two-position or proportioning) in the steam or in the hot water supply line.

Unit Ventilator Control

Various makes of unit ventilators are designed for different control cycles. Selection of automatic temperature control for unit ventilators is largely determined by the design of the particular unit. The choice of control cycle may also be determined by local and state ventilating codes, particularly where units are installed in school rooms. It is desirable to coordinate the selection of control cycle with the unit ventilator manufacturer since, in many cases, modifications of the unit are required for the installation of the control equipment; and, in some cases, it is desirable to have the equipment factory-mounted. Two typical control cycles are: (1) *Variable Outdoor Air with Fixed Minimum;* and (2) *Variable Outdoor Air without Fixed Minimum.*

Control Cycle No. 1. In full heating position, the outdoor air damper is closed, the recirculating air damper is open, and the supply valve is open.

In full cooling position the outdoor air damper is open, the recirculating air damper is closed and the supply valve is closed or at a minimum position to maintain a minimum discharge air temperature. The sequence of control operations is: On call for cooling, under control of a room thermostat, outdoor air damper opens to minimum setting. The supply valve then closes gradually, after which the outdoor air damper gradually opens to the maximum position and, simultaneously, the recirculating air damper closes. A low limit thermostat mounted above the coil prevents the discharge temperature from dropping below a predetermined minimum.

Control Cycle No. 2. In full heating position the outdoor air damper is closed, the recirculating air damper is open and the supply valve is open.

In full cooling position the supply valve is closed, the outdoor air damper is open and recirculating air damper is closed or at any position required to maintain a minimum discharge air temperature. The sequence of control operations is: On call for cooling, under control of a room thermostat, the valve gradually closes. As the valve leaves full-open position, control of the recirculated air and outdoor air dampers is transferred to a thermostat installed ahead of the heating coil to maintain a minimum air temperature. On continued call for cooling, the valve gradually closes.

If direct radiation is used, it is desirable that it be controlled in sequence with whatever control cycle is adapted for the unit ventilators.

Unit Coolers

Although most unit coolers can be adapted to any control cycle, continuous fan operation is recommended to avoid stratification and wide

fluctuations in space temperature. Should the unit be completely self-contained, control of the direct expansion refrigeration unit may be obtained from the temperature of the recirculated air and from suction pressure. In the case of multiple unit systems supplied with refrigerant or chilled water from a central source, a valve in the supply to each cooling coil may be controlled thermostatically from space temperature.

Refrigeration and Dehumidification Equipment

Typical control equipment and its functions are described in the following paragraphs.

Well Water. Where well water is used directly in air washers or cooling coils, control of temperature or humidity is usually obtained by thermostats or humidistats operating valves (two-position or proportioning). The two-position valve will provide better dehumidification since a lower coil temperature will be maintained but the temperature of the discharge air will fluctuate. With proportioning control, better control of discharge air temperatures will be maintained. In both cases the sensible-latent heat ratio is basically a matter of coil design rather than automatic control. Proportioning three-way valves may be used as mixing or diverting valves for better pump performance and may also be applied to an air washer used with a recirculating pump to control water temperature rather than volume.

Ice Bunkers. Where water is sprayed over the ice in bunkers and circulated to air washers or cooling coils control is obtained by a thermostat in the water line from the bunker. The thermostat proportions a three-way valve to by-pass enough return water around the ice bunker to maintain a constant discharge temperature.

Compressors. Compressors may supply refrigerant to direct expansion cooling coils in air conditioning units or to direct expansion coils in water-chilling units. In either case, the compressor motor may be started and stopped directly by a room or duct thermostat or a pressure controller may be used to regulate the suction pressure of the compressor. In the latter case, a room or duct thermostat may be used to control a solenoid valve in the refrigerant supply line to the cooling coil. A high and low pressure cut-out is standard safety equipment on most compressor installations. Reduced capacity of the refrigerating unit may be obtained by means of temperature or pressure controlled unloading devices which vary the capacity of the compressor in some proportion to variations of cooling load. Program or step controllers, actuated by temperature or pressure, are commonly used in multiple compressor installations. It is desirable in such installations to return the program or step controller to the *off* position when the system is shut down to prevent the full electrical load of multiple compressors from being thrown across the line at the same time. Thermostatic control of water supply to water-cooled condensers may be achieved by means of self-contained controllers or valve and thermostat application.

Steam Jet. A steam jet refrigeration system is commonly controlled by means of a thermostat in the chilled (secondary) water. The thermostat operates a two-position valve in the steam line to the jet. In the case of multiple jet units, program or step control can be achieved by controlling the jets in sequence. As in the case of direct expansion refrigeration units, a system of control is advisable for the water-cooled-condensing unit.

Centrifugal Units. The control of centrifugal refrigeration units or other types of vacuum systems is customarily achieved by means of a thermostat in the chilled water to control the operating cycle of the equipment at full or reduced capacity.

Adsorption Units. Control of adsorption units consists of a damper control which, in response to humidity, controls the air flow through or around the activated bed of adsorption material. Standard controls for cooling are used to reduce the dehumidified air to a desired drybulb.

Absorption Units. Since at constant density, the absorption solution will extract water from the treated air in an amount proportional to the solution temperature, the moisture content of the air leaving an absorption unit is regulated by solution temperature. Solution density is held constant by a combination of float control and steam valve controlling the solution regenerator. Two basic methods of control are standard:

1. Constant solution temperature where the solution temperature is set so that, at the full load for which the unit is designed, the discharge air will have the

desired moisture content. Proportioning control of the water and two-position control of the steam is recommended to maintain constant temperature.

2. Control by varying the solution temperature so that the moisture content of the discharge air remains constant regardless of load variation. Basically, this control is similar to the constant solution temperature control with the addition of a hygrostat or wetbulb controller controlling the water valve from space conditions. In order to secure a constant discharge drybulb temperature, a coil is provided with proportioning valve controlled by a proportioning thermostat in the unit discharge.

CONTROL FOR CENTRAL FAN SYSTEMS

Automatic temperature control for central fan heating, cooling, ventilating and air conditioning systems involves the proper application of various types of controlling instruments and associated regulators such as valves, dampers and damper operators, relays and other auxiliary equipment

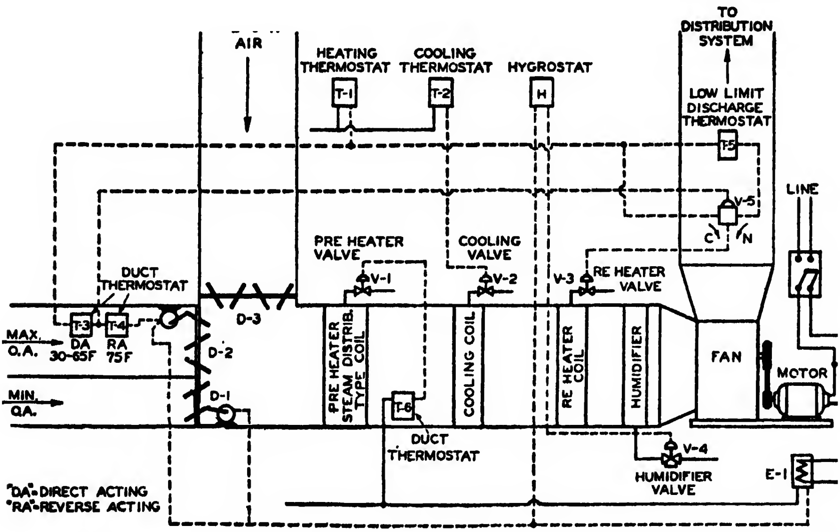

Fig. 1. Control Diagram for Year 'Round Air Conditioning System

which are described in earlier sections of this chapter. In central fan systems the conditions required dictate the type of built-up control system to be used. Otherwise, some arrangement of available *package* equipment probably would be used. For that reason, it is impossible to state in detail the control apparatus which will be required in even the most representative applications.

In general, in so far as automatic temperature and humidity control equipment is concerned, central fan systems may be divided into certain broad classifications, as follows:

1. Heating
2. Humidifying
3. Ventilating and Atmospheric Cooling
4. Cooling and dehumidifying
5. Control of Zone Temperatures
6. Year-round air conditioning with automatic change-over
7. Constant temperature and humidity

The apparatus which enters into the automatic maintenance of temperatures and humidities for central fan systems which are designed to produce each of the effects listed in items 1 to 7, is indicated in the following paragraphs:

1. In *heating control*, there are three considerations to be borne in mind: (1) to control space temperature, (2) to prevent drafts, (3) to guard against freezing. Usually in central fan systems, suitable thermostats operate valves in the steam or hot water supply to heating coils or face dampers across such coils and bypass dampers around them. If the heating coils are sub-divided into two or more groups, such as preheaters and reheaters, a duct thermostat following the preheaters, and located in the entrance to the chamber between the groups of coils, controls the preheaters, but it is essential that the preheater coils be of the steam distributing type. Similarly, a duct thermostat in the fan discharge, where any effect of stratification has been dissipated, operates the valves and dampers associated with the reheaters. In some cases, where there is more than one bank of preheaters, the practice is to place

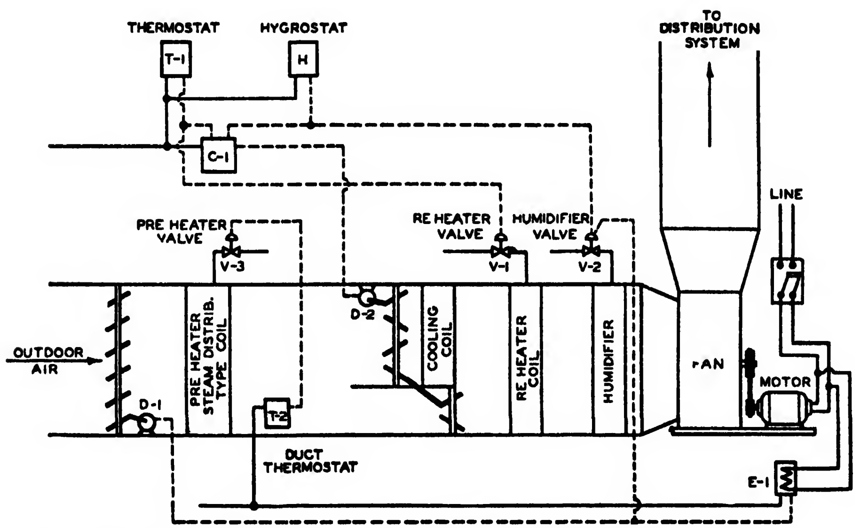

FIG. 2. CONSTANT TEMPERATURE AND HUMIDITY CONTROL FOR YEAR 'ROUND SYSTEM USING 100 PER CENT OUTDOOR AIR

a freeze protection thermostat in the outdoor air intake to control the valve on the first bank of heaters, which is designed so that the heat-rise through it will not cause overheating. A room thermostat in the heated space may serve as the controlling instrument, with a thermostat in the fan discharge serving to prevent the delivery of air at a temperature which might cause drafts. If desired, similar action may be obtained from a thermostat in the return air connection. When there is only one heating coil, a limit thermostat in the fan discharge accomplishes the control, in conjunction with a thermostat in the heated space or in the return air.

2. *Humidity control* may be obtained by means of a hygrostat, usually located in the conditioned space or in the return air. Such controlling instruments may operate steam supply valves to humidifiers, mixing valves to control the temperature of the water to sprays, or a system of dampers to regulate the quantity of air passed through the humidifying chamber or bypassed around it. The control of humidity according to dew-point temperature is sometimes accomplished by means of a thermostat in the outlet of the humidifying chamber. However, the setting of the dew-point thermostat may have to be changed as the humidity in the conditioned space varies.

3. The *control of ventilating* and *cooling* by the use of outdoor air consists of an arrangement of dampers, usually determining the relative quantities of outdoor air and return air which are to be delivered to the conditioned space. The damper positions are regulated by proper types of thermostats, located in the minimum outdoor

air intake, the fan discharge, the conditioned space or the return air. Instruments are available which, with an adequate arrangement of dampers, will cause a maximum quantity of outdoor air to be handled until it becomes more economical to utilize return air.

4. *Cooling and dehumidifying* may be controlled by means of thermostats, and hygrostats or dew-point thermostats, which regulate dampers and mixing valves to maintain air of the proper temperature and humidity in the discharge from the central fan plant. Such controlling instruments normally are located in the fan discharge or in the return air, or both, and they may be associated with thermostats or hygrostats in the conditioned spaces.

5. Where a *separate duct serves each zone* of an area with which a central fan system is associated, a room thermostat in each zone may operate mixing dampers in the inlet to each zone duct, determining the quantity of warm air which is required from that portion of a plenum chamber into which heated air is delivered and the quantity of cool air which should be taken from the other portion of the double plenum chamber. In many instances, separate zone heating and zone cooling coils are employed, instead of mixing dampers.

6. The control hook-up for a typical *year 'round air conditioning system*, including automatic change-over from heating to cooling, is indicated in Fig. 1 and described as follows:

Whenever the fan is started, solenoid air valve or relay E-1, actuated by the fan motor starter, opens minimum outdoor air damper D-1, places hygrostat H, in service, and allows duct thermostats T-3 and T-4 to control the maximum outdoor air damper D-2 and the return air damper D-3.

When the fans stop, E-1 is de-energized, to close the outdoor air dampers and also to close humidifier valve V-4.

Thermostat T-1 positions steam valve V-3 on the reheater coil, to maintain a constant space temperature. As the space temperature rises, T-1 positions reheater valve V-3 to a closed or to a minimum open position, as determined by low limit discharge thermostat T-5. Duct thermostat T-6, in the preheater discharge, positions preheater coil valve V-1, to maintain a constant preheater discharge temperature.

On rising outdoor temperature, between 30 F and 65 F, duct thermostat T-3, located in the outdoor air intake, moves maximum outdoor air damper D-2 toward the open position. At 65 F, outdoor, D-2 will be fully open and return air damper D-3 will be fully closed.

As the outdoor air temperature rises above 65 F, duct thermostat T-3 positions V-5 in such a way as to by-pass low limit thermostat T-5, so that reheater coil valve V-3 is operated directly from thermostat T-1. As outdoor air temperature rises from 65 F to 75 F, duct thermostat T-4 gradually closes maximum outdoor air damper D-2 and opens return air damper D-3.

Cooling thermostat T-2 positions cooling coil valve V-2, to admit more chilled water, as the space temperature rises.

Hygrostat H positions humidifier valve V-4 to maintain the desired humidity in the conditioned space.

7. The arrangement of automatic control for a *constant temperature and constant humidity air conditioning system*, using 100 per cent outdoor air, is shown in Fig. 2, and the control description follows:

Whenever the fan is running, relay or solenoid air valve E-1, actuated by the fan motor circuit, is energized, opens outdoor air damper D-1, and also permits hygrostat H, in the conditioned space, to control humidifier valve V-2.

When the fan stops, E-1 closes outdoor air damper D-1 and humidifier valve V-2.

Remote bulb thermostat T-2, with bulb located in preheater discharge, operates valve V-3 on the preheater coil, to maintain a constant preheater discharge temperature.

On rising temperature, thermostat T-1, in the conditioned space, closes reheater valve V-1 and, through relay C-1, opens face damper D-2, for cooling. On rising humidity in the conditioned space, hygrostat H closes humidifier valve V-2, and likewise, through C-1 may open face damper D-2 for dehumidification.

For closer control, the face and bypass dampers should be eliminated and cooling means continuously provided whenever the outdoor dew-point rises above a predetermined maximum. Reheating and humidifying may be required to provide the desired conditions. However, such a system will be less economical in operation.

PANEL HEATING CONTROL

Automatic controls for radiant and convective heating differ somewhat due to the thermal inertia characteristics of the panel heating surface, and the increase in the mean radiant temperature within the space under increasing loads for panel heating.

Effect of Inertia of Panel

If a panel has considerable heat storage capacity (as compared with a convector or conventional radiator) it will continue to emit heat for some time after the room thermostat has become satisfied and shut off the supply of heating medium. This will cause uncomfortably warm conditions to exist in a space. Also, there will be a considerable delay between the time the thermostat calls for heat and the time heat is actually delivered to the space (because of the large part of the heat that must first be stored in the thermally "heavy" radiant surface). Whenever inertia exists in the source of heat supply, uncomfortable cycling of space conditions will result unless means of anticipating load changes *before* they occur in the space, or means of setting the basic energy supply rate from load conditions, is provided.

If a thermally heavy radiant surface is used the primary control should be actuated by outdoor temperature (load) to determine the basic temperature of the heating medium supplied to the radiant surface. To allow for variations in internal load, an inside thermostat should be used as a high limit to reduce further the heat input if necessary. If a thermally *light* radiant surface is used, controls may be applied in the same manner as for typical convection heating.

The terms *thermally heavy* and *thermally light*, referring to capacity for heat storage, are comparative and descriptive rather than exact. For example, a concrete floor panel in a frame structure without insulation would represent a *heavy* panel in a *light* structure. A frame type (metal lath and plaster) panel in a concrete structure would represent a *light* panel in a *heavy* structure. As indicated previously a *heavy* panel in a *heavy* structure provides comfortable conditions if outside controls are used in addition to the inside thermostat. But if a *heavy* panel is used in a *light* structure, rapid changes in outdoor conditions may cause discomfort in spite of outdoor controls, because the structure reacts so much more rapidly than the radiant heating surface.

Compensation for Increase of MRT

In order to maintain comfort, the air temperature in a panel heated space should be lowered as the heating load increases. Usually the required reduction in air temperature is not great and a conventional fixed control point room thermostat may be used unless a rather large infiltration load exists, or if untempered mechanical ventilation is used. Because of the relationships existing between MRT (mean radiant temperature) and air temperature in the space (and the variable MRT from point to point in the space), a conventional type of room thermostat (either fixed or variable control point as previously determined) may provide simple and satisfactory control.

Lowered Night Temperature

In general, lowered night temperature control is not recommended with *heavy* panels though it may be satisfactory with *light* panels. Best practice

indicates continuous circulation of the heat in a medium with control provided to vary the temperature of the medium.

INDICATING AND RECORDING EQUIPMENT

In addition to the automatic control of temperature and humidity conditions, visual indication and permanent chart records of the variables involved are desirable. They provide an accurate check on the performance of the system both from the standpoint of conditions maintained and cost of operation. Instruments are available to provide accurate records of these variables, such as pressure, temperature, humidity, flow and CO_2, which go to make up a complete heating or air conditioning system. In some cases the control equipment is provided with indicating or recording mechanisms by means of which the performance of the controls may be observed or recorded and in other cases separate instruments are used for the purpose.

CHAPTER 35

MOTORS AND MOTOR CONTROLS

Motor Rating; Functions of Motor Control Equipment; Direct Current Motors, Types, Control Equipment and Specifications; Alternating Current Motors, Types, Control Equipment and Specifications; Gear Motors; Glossary of Motor Terms, Enclosures, Speed Classification and Mounting

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors are divided into two general classifications, *alternating-current* or *direct-current*, depending on the power source to be used.

In selecting a motor for a particular application consideration must first be given to the type of power supply available. All machinery has certain load characteristics which may vary with speed. Some types may have a constant torque over wide ranges of speed, while others may have changing torques with changing speed. Consideration should be given to selecting the motor and the motor control which best suit the requirements of the drive.

MOTOR RATING

The rating of an electric motor depends upon the total temperature which the motor attains under operating conditions. This total temperature depends on both the ambient temperature and the temperature rise of the motor. As motor temperature rise is in turn determined by the ability of the motor to dissipate heat, circulation to the motor should not be restricted. Improper selection of motors with regard to temperature ratings may result in high motor operating temperatures with accompanying reduction in motor life.

In general, the electrical insulation is the portion of the motor most susceptible to injury from high operating temperatures. Of the several types of insulation which are available, the most common type, specified as Class A by the *National Electrical Manufacturers Association*, consists of cotton, felt, paper or similar organic materials and permits a 55 C rise in temperature over a 40 C ambient temperature for totally enclosed motors. Class B insulation consists of mica, asbestos, fiber glass, or similar inorganic materials and permits a 75 C rise in temperature over the 40 C ambient for totally enclosed motors. Other types of insulation such as silicone resin are available and permit much higher operating temperatures.

The mechanical construction of the different types of motor enclosures and the rise in temperature with Class A insulation for each type are enumerated in the glossary at the end of this chapter. Since the difference between the hottest spot and the maximum observable temperature, as measured by a thermometer, is greater for an open machine than for an enclosed machine, the permissible temperature rise is 50 C for an open motor.

TABLE 1. CLASSIFICATION OF MOTORS

Power Supply	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote†
Constant Speed Drives						
d-c	1. Shunt	Constant	Normal (with controller)	Normal (with controller)	All	(a) Fans and (c) centrifugal pumps and centrifugal compressors
d-c	2. Compound	Variable	High (with controller)	Normal (with controller)	All	(b) (c) (e) Reciprocating pumps and frequent or hard starting
d-c	3. Series	Variable	High (with controller)	Normal (with controller)	Small	(d) Fans direct connected
Polyphase a-c	4. Squirrel-cage general purpose Design A	Constant	Normal 1–2.5 times	High 6–8 times	All	(a) Fans and (c) centrifugal pumps and centrifugal compressors
Polyphase a-c	5. Squirrel-cage Design B	Constant	Normal 1–2.5 times	Normal 5–6 times	Medium Small	(a) Fans and centrifugal pumps and centrifugal compressors
Polyphase a-c	6. Squirrel-cage Design C	Constant	High 2–2.5 times	Normal 5–6 times	Medium Small	(b) Reciprocating pumps (e) and compressors started loaded
Polyphase a-c	7. Wound rotor	Constant or variable	High 1–2.5 times (with secondary control)	Low 1–3 times (with secondary control)	All	(a) Hoists (b) reciprocating pumps and compressors (c) and frequent (e) or hard start
Polyphase a-c	8. Synchronous high speed	Exactly constant	Normal 0.75–1.75 times	Normal 5–7 times	Medium Large	(a) Fans and centrifugal pumps and centrifugal compressors
Polyphase a-c	9. Synchronous low speed	Exactly constant	Low 0.3–0.4 times	Low 3–4 times	Medium Large	(a) Reciprocating compressors starting unloaded
Single phase a-c	10. Two value capacitor	Constant	High	Normal	Small	(b) Pumps and compressors
Single phase a-c	11. Permanent split capacitor.	Constant	Low	Normal	Fractional	(a) Fans, Blowers
Single phase a-c	12. Capacitor start	Constant	Moderate	Normal	Small Fractional	(a) Fans and pumps
Single phase a-c	13. Repulsion Induction	Constant	High	Normal	Medium Small	(a) Fans (b) pumps and compressors
Single phase a-c	14. Split phase	Constant and adjustable	Normal	Normal	Fractional	(a) Fans (b) pumps and compressors (d) fans—direct

† Applications:

a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded.

b. Drives having high starting torques, such as reciprocating pumps and compressors started loaded.

c. Similar to (a) except where frequent or hard starting (large WR^2) requires a higher starting and accelerating torque.

d. Fans direct connected.

e. Stoker drives.

TABLE 1. CLASSIFICATION OF MOTORS—(Concluded)

Power Supply	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote†
			Adjustable Speed Drives			
d-c	15. Shunt field adjustment	Constant	Normal (with controller)	Normal (with controller)	All	(a) Fans and (e) centrifugal pumps
d-c	16. Armature resistance adjustment	Variable	Normal (with controller)	Normal (with controller)	All	(a) Fans and (e) centrifugal pumps
d-c	17. Variable voltage control	Constant	Normal (with controller)	Normal (with controller)	All	(d) Fans and centrifugal pumps
Polyphase a c	18. Squirrel-cage high slip. Transformer adjustment	Variable	Normal	Normal	Medium small	(a) Fans
Polyphase a c	19. Squirrel-cage separate winding or regrouped poles	Constant multispeed	Normal or high	Normal or low	All	(a) Fans (b) pumps and (c) compressors
Polyphase a c	20. Wound rotor	Variable	High (with secondary control)	Low (with secondary control)	All	(a) Fans (b) centrifugal pumps and compressors
Single Phase a-c	21. Repulsion	Variable	High	Normal	Low and Fractional	(a) Fans, centrifugal pumps (b) compressors
Single Phase a-c	22. Capacitor low torque tapped winding	Variable two speed	Low	Normal	Fractional	(d) Fans, direct
Single Phase a-c	23. Capacitor low torque transformer adjustment	Variable	Low	Low	Fractional	(d) Fans
Single Phase a-c	24. Split phase regrouped poles	Constant	Normal	Normal	Fractional	(d) Fans

FUNCTIONS OF CONTROL EQUIPMENT FOR MOTORS

In general, control equipment for all types of motors should provide: (1) means of disconnecting the motor from the power supply; (2) means for starting the motor; (3) overload protection for the motor; (4) protection against low voltage; and (5) means for varying the motor speed.

Full voltage starting for motors is preferable because of its lower first cost and simplicity of control. Except for d-c machines, most motors are mechanically and electrically designed for full voltage starting. The starting inrush current, however, is limited in many cases by regulations of power companies because of the voltage fluctuations which may be caused by heavy current surges. It is therefore often necessary to reduce the starting current below that obtained by across-the-line starting. The power supplier should be consulted to determine the allowable inrush current for any given location.

The choice between full voltage and reduced voltage starting is governed almost entirely by inrush current limitations. The starting torque of all

motors varies with the starting current and it is therefore necessary to insure that the motor is supplied with sufficient current to develop enough torque to accelerate the load.

In present practice overload protection of motors is obtained by use of thermal overload inverse time limit type protection. The usual setting of such protection devices is not to exceed 125 per cent of rated full load current for open 40 C rise motors and not to exceed 115 per cent of rated full load current for all other motors, the element tripping after a definite interval of time. The National Electrical Code requires the addition of fuses or circuit breakers to protect the overload elements from severe short circuit currents.

Two types of protection are available against low voltage at the motor terminals. One type, called low voltage release, permits the motor line contactor to drop out on low voltage and to close again when the voltage returns to normal, thereby restarting the motor when the abnormal condition is ended. The second type, called low voltage protection, causes the motor line contactor to drop out on low voltage but prevents restarting when the voltage returns to normal except by the action of an operator. This latter type of protection is desirable where it is necessary for the operator to make initial starting adjustments on the machine.

Manual control for an alternating or a direct current motor is usually located near the motor. When so located an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Manual control is sometimes employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system. In other cases manual control in the form of drums, when used with multi-speed motors, is only used as a speed setting device while the starting and stopping functions operate automatically through thermostats and pressure switches.

Because of the increasing complexity of air conditioning systems, the equipment is operated preferably by automatic control and less dependence is placed on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator, or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere.

DIRECT CURRENT MOTORS

Direct current motors are classified (see Table 1) according to type of winding as: *shunt wound*, *compound wound*, and *series wound*.

Shunt Wound motors, being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of direct current applications in the field of heating, ventilating, and air conditioning. They may be used on reciprocating pumps and compressors if started under unloaded conditions.

Without auxiliary control the shunt wound motor is designated as *constant speed.** Fig. 1 illustrates the characteristics of direct current

* Refer to Glossary at end of chapter.

motors, showing speed, horsepower, and torque as a function of current. The speed regulation* of small size shunt wound motors from ¾ hp to 5 hp is 12 per cent as specified by the *NEMA* while on larger motors it is 10 per cent.

Compound Wound motors are required for application to reciprocating compressors, stokers, reciprocating pumps when started under loaded conditions, and other similar equipment requiring high starting torque. The characteristics of this type of motor are such that for starting torques above full-load torque the starting current required is somewhat less than for the shunt wound motor. Compound wound direct current motors are normally used whenever frequent starting makes high starting and

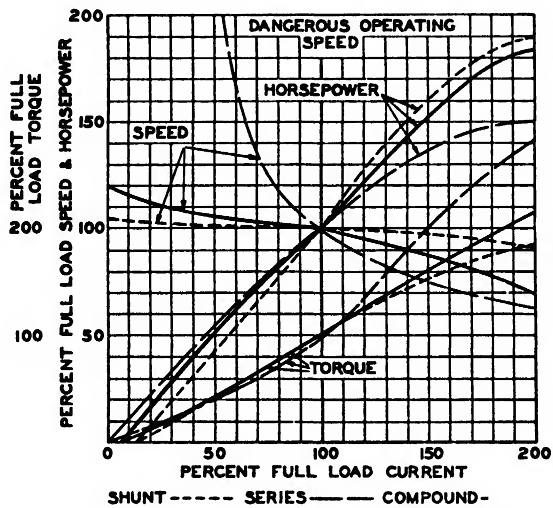

Fig. 1. Characteristics of Direct Current Motors

accelerating torque desirable. Without auxiliary control, compound wound motors are designated as *varying speed,** and have a speed regulation of 25 per cent.

Series Wound motors find only limited application in a few special cases and are available in a limited range of sizes. The motors are used where extremely high starting torques are required and must be applied only to direct coupled continuous loads due to the fact that the speed of the motor becomes dangerously high when the motor is operated at a light load.

Typical d-c motor specifications are shown in Table 2.

DIRECT CURRENT MOTOR CONTROL

Direct current motors are usually started through starting controllers employing a resistance in series with the motor armature which is gradually cut out as the motor comes up to speed. Motors up to 2 hp may be line-started providing the inrush current causes no serious voltage fluctuations in the power supply line.

Constant Speed and Varying Speed Motors. As shown in Fig. 2, the recommended practice for manual starting of motors over ½ hp requires

* Refer to Glossary at end of chapter.

Table 2. Typical Specifications for d-c Motors

..... Hp, ...RPM, $\left\{\begin{matrix}\text{Constant}\\ \text{Adjustable}\\ \text{Varying}\end{matrix}\right\}$ Speed, $\left\{\begin{matrix}\text{Shunt}\\ \text{Compound}\\ \text{Series}\end{matrix}\right\}$ Wound, $\left\{\begin{matrix}115\\ 230\\ 600\end{matrix}\right\}$ Volts d-c Motors For Driving .... (Application) Motor Shall Be Arranged For $\left\{\begin{matrix}\text{Horizontal}\\ \text{Vertical}\end{matrix}\right\}$ Mounting And Shall Be Provided With A .... (Open, Splashproof, etc.) Type Of Enclosure, *NEMA* $\left\{\begin{matrix}\text{Class } A\\ \text{Class } B\end{matrix}\right\}$ Insulation, And $\left\{\begin{matrix}\text{Ball}\\ \text{Sleeve}\end{matrix}\right\}$ Bearings.

Table 3. Typical Specifications for d-c Motor Control

Control For $\left\{\begin{matrix}\text{Constant}\\ \text{Adjustable}\\ \text{Adjustable Varying}\end{matrix}\right\}$ Speed d-c Motors Shall Be $\left\{\begin{matrix}\text{Manual}\\ \text{Magnetic}\end{matrix}\right\}$ And Shall Consist of a $\left\{\begin{matrix}\text{Safety Switch}\\ \text{Circuit Breaker}\end{matrix}\right\}$ And an Enclosed Controller Providing Overload Protection [And Low Voltage $\left\{\begin{matrix}\text{Protection}\\ \text{Release}\end{matrix}\right\}$].

Table 4. Typical Specifications for Squirrel-Cage Motors

..................Hp, ...RPM, $\left\{\begin{matrix}25\\ 60\end{matrix}\right\}$ Cycles, $\left\{\begin{matrix}208\\ 220\\ 440\\ 550\end{matrix}\right\}$ Volts, $\left\{\begin{matrix}2\\ 3\end{matrix}\right\}$ Phase, Squirrel-Cage Induction Motors of the *NEMA*

$\left\{\begin{matrix}\text{Design A—Normal Starting Torque, Starting Current above } \textit{NEMA} \text{ value}\\ \text{Design B—Normal Starting Torque, } \textit{NEMA} \text{ Starting Current}\\ \text{Design C—High Starting Torque, } \textit{NEMA} \text{ Starting Current}\\ \text{Design D—High Slip High Starting Torque}\end{matrix}\right\}$ Type**

Driving.................... (Application) Motor Shall Be Arranged For $\left\{\begin{matrix}\text{Horizontal}\\ \text{Vertical}\end{matrix}\right\}$ Mounting And Shall Be Provided With A .... (Open, Splashproof, etc.) Type of Enclosure, *NEMA* $\left\{\begin{matrix}\text{Class A}\\ \text{Class B}\end{matrix}\right\}$ Insulation, And $\left\{\begin{matrix}\text{Sleeve}\\ \text{Ball}\end{matrix}\right\}$ Bearings.

** Refer to following section on Alternating Current Motors for explanation of types.

Table 5. Typical Specifications for Wound Rotor Motors

...........Hp, ...RPM, $\left\{\begin{matrix}25\\ 60\end{matrix}\right\}$ Cycles, $\left\{\begin{matrix}208\\ 220\\ 440\\ 550\end{matrix}\right\}$ Volts, $\left\{\begin{matrix}2\\ 3\end{matrix}\right\}$ Phase, Wound Rotor Induction Motors For Driving .... (Application) Motor Shall Be Arranged for $\left\{\begin{matrix}\text{Horizontal}\\ \text{Vertical}\end{matrix}\right\}$ Mounting And Shall Be Provided With A............................ (Open, Splashproof, etc.) Type of Enclosure, *NEMA* $\left\{\begin{matrix}\text{Class A}\\ \text{Class B}\end{matrix}\right\}$ Insulation and $\left\{\begin{matrix}\text{Ball}\\ \text{Sleeve}\end{matrix}\right\}$ Bearings.

the use of a fused safety switch or circuit breaker and a face-plate type starter. For automatic push button starting a safety switch or circuit breaker and an automatic starter are recommended

Adjustable Speed Motors are normally shunt wound and are operated at various speeds by varying a resistance connected in series with the motor field. A maximum range of speed of about 5 to 1 can be obtained by this means. Rated speed regulation is 22 per cent for motors of this type from 2 to 5 hp; 15 per cent is standard in larger sizes. The *NEMA* practice on rating adjustable speed d-c motors is defined in the Glossary at the end of the chapter.

The control for the adjustable speed motor consists of the addition of

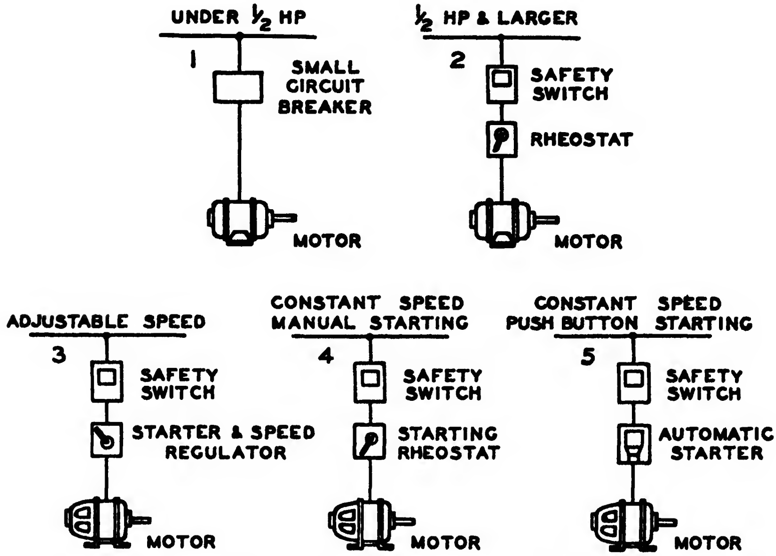

FIG. 2. RECOMMENDED CONTROLS FOR d-c MOTORS

a field rheostat for speed control to the equipment specified for the constant speed motor.

*Adjustable Varying Speed** motors are d-c motors in which the speed is varied by the addition of resistance in series with the armature. The speed of the motor by this means is always less than the rated full field speed and varies widely with a change in load, especially with high series resistance. Fig. 3 illustrates typical speed characteristics of this type of motor for different values of armature resistance.

The addition of series resistance in the armature circuit reduces the motor speed by lowering the voltage on the armature. At one-half speed the voltage is approximately one-half of line voltage. Consequently, with rated full load current the power delivered by the motor will be only one-half of the maximum, *e.g.*, it will be 5 hp from a 10 hp motor, because the other 5 hp will be lost in the resistance. It is, therefore, evident that the efficiency of the motor is reduced at reduced speeds due to the loss in the resistor.

Control for the adjustable varying speed motor is similar to that for

* Refer to Glossary at end of chapter.

the constant speed motor with the exception that the starting resistor must be designed for speed regulating duty and must therefore be capable of carrying the motor current continuously. Speed controllers are available both for constant torque applications and varying torque drives (such as required by fans) in which the torque is reduced considerably at reduced speed.

The *Adjustable Voltage* type of speed control is also often known as the variable voltage or Ward-Leonard system. For machines requiring a wide range in speed control and a large number of steps of control this type of system is used most extensively. The drive consists of one or more d-c motors, the armatures of which are supplied with power from a d-c generator and the fields of all machines are excited from a constant

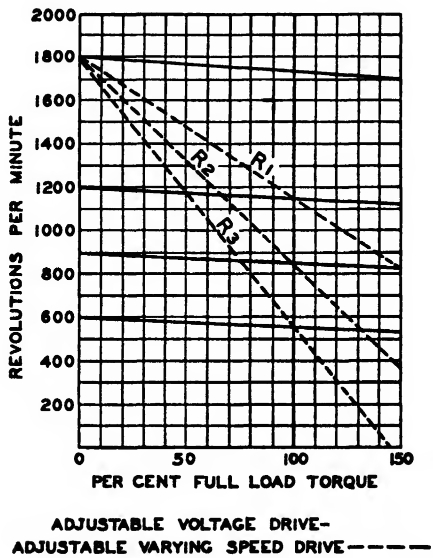

Fig. 3. Speed Torque Characteristics of Adjustable Varying Speed Drive and Adjustable Voltage Drive

voltage exciter. A schematic diagram of connections for an adjustable voltage drive is shown in Fig. 4. In most cases the d-c generator is a part of a three-unit set including a constant speed a-c driving motor, and a constant voltage exciter. As the voltage on the generator and consequently on the d-c motor or motors is adjusted by a rheostat in the generator field circuit, a great many steps are thus obtained in an efficient manner. With constant field excitation on the motor the speed of the motor will vary approximately as the voltage on the generator.

Extremely wide speed ranges are possible with the adjustable voltage type of drive. Ranges as high as 10 to 1 are common and, by the addition of field control on the motors, ranges as high as 40 to 1 are permissible. This type of drive provides the advantage of good speed regulation over the entire speed range, as shown in Fig. 3 in which this type of drive is compared with the adjustable varying speed drive.

Typical specifications for d-c motor control are shown in Table 3.

ALTERNATING CURRENT MOTORS

Alternating current motors are divided into two main classifications: *polyphase* and *single phase* (see Table 1), according to the type of power supply used. They are further subdivided as to the type of motor winding.

When polyphase power is available it is usually found more economical to apply polyphase motors in preference to single phase motors. A typical 5 hp, 1200 rpm capacitor start-induction run single phase motor, for instance, will cost approximately twice as much as the corresponding three phase *Design B* squirrel-cage motor. In addition, the polyphase motor has the advantages of higher power factor and higher efficiency.

Polyphase Motors

The three types of polyphase motors are: *squirrel-cage induction* motors, *wound rotor induction* motors, and *synchronous* motors.

Squirrel-cage motors are specified by NEMA standards providing a variety of speed and torque characteristics. *Design A* motors provide

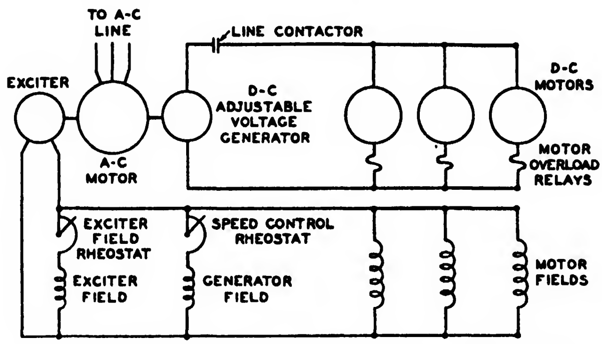

FIG. 4. COMPONENT PARTS OF AN ADJUSTABLE VOLTAGE DRIVE

normal starting torque at starting current in excess of *Design B* motors and are suitable for constant speed application, to equipment such as fans and blowers. *Design B* motors provide normal starting torque with *NEMA* starting current values which are acceptable by many power companies for full voltage starting. They are used for the same type of application as *Design A*. *Design C* motors provide high starting torque with starting current same as *Design B* and are used on compressors started without unloaders, and on reciprocating pumps. *Design D* motors have high slip* and are used with flywheels for widely pulsating loads on equipment such as reciprocating compressors and pumps where other motors would draw high peak currents.

Figs. 5, 6, 7, and 8 illustrate the characteristics of squirrel-cage motors. It will be noticed by inspection of Fig. 6 that both power factor and efficiency are improved if the motors are operating as near rated load as possible. In addition, as shown in Fig. 8, power factor and efficiency are better for higher speed motors.

Typical specifications for squirrel-cage motors are shown in Table 4.

Wound Rotor motors are used for applications requiring high starting

* Refer to Glossary at end of chapter.

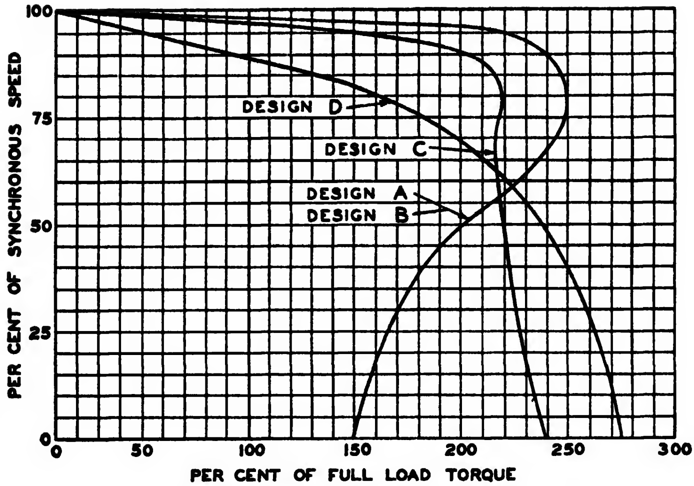

Fig. 5. Speed Torque Characteristics of Squirrel-Cage Motors

torque at low starting current, because a wound rotor motor with its controller and resistance can develop full load torque when starting with about full load current. For comparison, a squirrel-cage motor would require from 3 to 5 times as much current to develop full load torque at starting. The wound rotor motor is also used for varying speed service to drive fans, blowers, and other continuous duty apparatus. Typical specifications for wound rotor motors are shown in Table 5.

The addition of resistance to the secondary winding of the wound rotor motor changes the speed torque characteristics as indicated in Fig. 9. The motor speed, with the resistance added, is dependent on load and

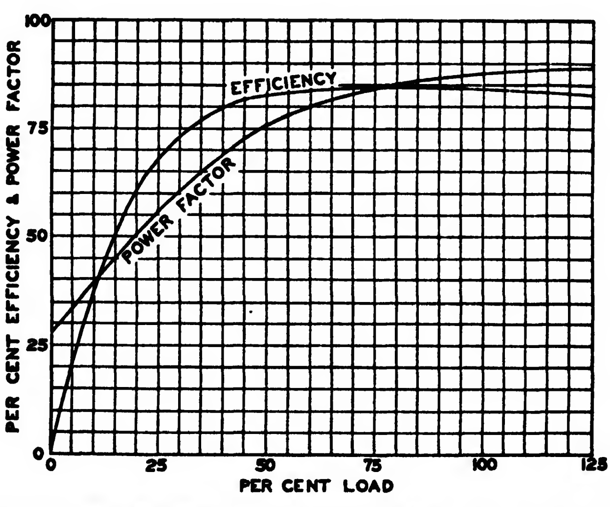

Fig. 6. Variation of Efficiency and Power Factor with Load for a Typical 5 Hp, 4 Pole 60 Cycle, Design B, Squirrel-Cage Motor

consequently the motor has very poor speed regulation when secondary resistance is added to reduce the speed to values below 50 per cent.

Synchronous motors are used for continuous duty applications at constant speed where efficiency and power factor are important. Another advantage of these motors is that of lower initial cost in large sizes and for low speeds when compared with squirrel-cage type motors.

The outstanding advantage of the synchronous motor is that its power factor can be changed to compensate for the low power factor of other drives in the same location. Lagging power factor is an inherent characteristic of all induction apparatus, such as induction motors and neon signs. Unless synchronous motors or capacitors are installed, the plant power factor may be comparatively low. This does not necessarily mean that corrective equipment must always be installed, but in most cases it is desirable to determine what advantages may be gained by improving

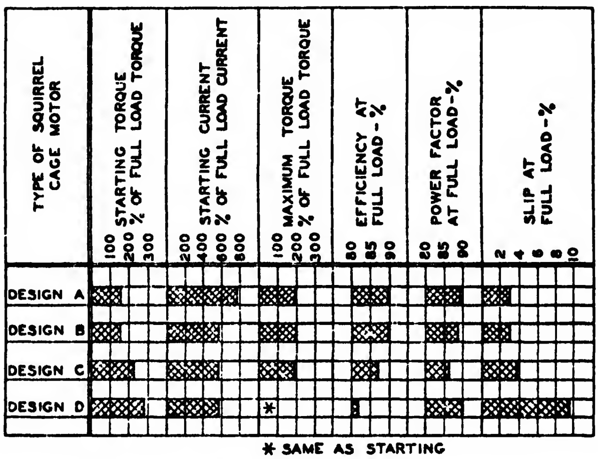

FIG. 7. COMPARATIVE PERFORMANCE OF SQUIRREL-CAGE MOTORS OF 30 HP AND SMALLER SIZES

the power factor. With purchased power, if the rates include a clause embodying a penalty for low power factor, or a bonus for high power factor, the saving in power costs may often make a very good return on the investment required for the corrective equipment.

Synchronous motors are used to drive fans, blowers, pumps, compressors and other applications. Compressor applications having a high peak torque require the use of flywheels to smooth out power peaks; and should always be referred to the electrical manufacturer for recommendations.

Synchronous motors are provided with built-in damper windings on the rotor and operate during the starting period similarly to squirrel-cage motors. After the motor is nearly up to speed, field excitation is applied and the motor draws into step at synchronous speed. After excitation is applied the motor runs at exactly constant speed and will remain at this speed until a load approaching the pull-out load is reached, whereupon the motor pulls out of synchronism and stops.

In applying synchronous motors consideration must be given to the torque the motor can develop on pull-in, that is, at the instant when field excitation is applied. Table 6 tabulates typical application requirements

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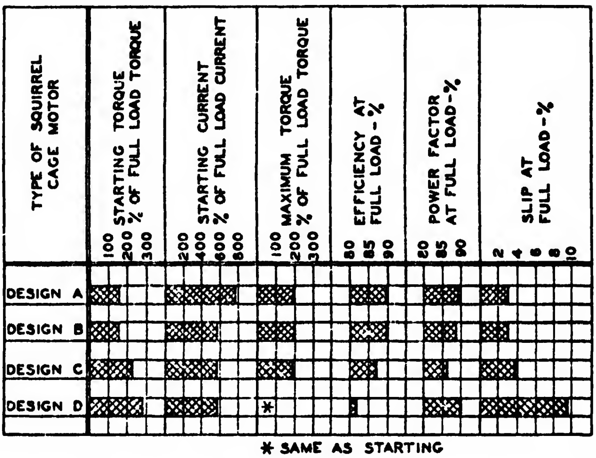

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of synchronous motor drives, listing starting, pull-in, and pull-out torques.

Typical specifications for synchronous motors are shown in Table 7.

Multi-Speed motors provide flexibility in many types of drives. Synchronous motors can be furnished only with a 2 to 1 ratio in speed, single winding. Squirrel-cage induction motors may be 2, 3 or 4 speed. Two-speed induction motors are usually of single winding type, having a 2 to 1 speed ratio such as 600 rpm and 1200 rpm, or may be double winding. Three-speed induction motors are always two winding, and four-speed motors are usually two winding with a 2 to 1 speed ratio in each winding.

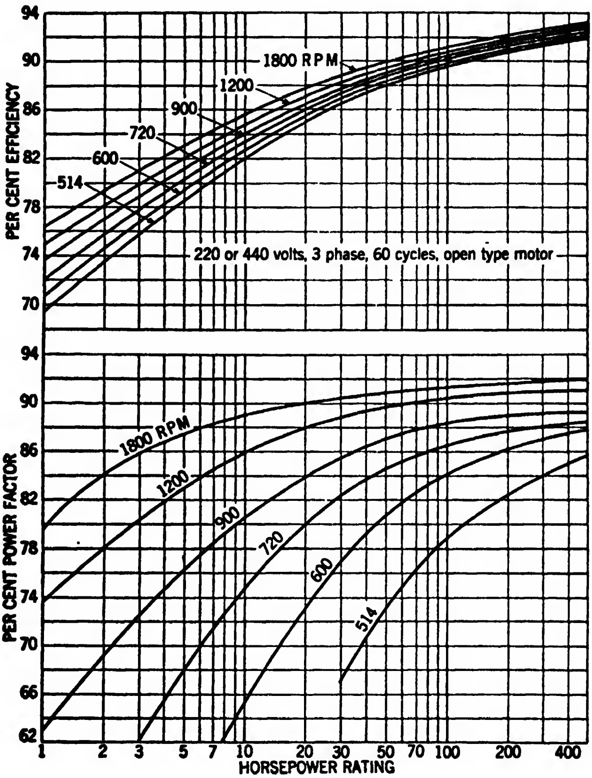

Fig. 8. Efficiencies and Power Factors for Squirrel-Cage Induction Motors

Motors can be provided in constant torque, varying torque, or constant horsepower ratings. The constant horsepower type of motor is considerably larger than the constant torque motor due to the fact that the same horsepower must be developed at either reduced speed or high speed.

In selecting two-speed motors for fan, pump, blower, or compressor applications, it is usually found that two winding motors are more expensive than the single winding type. The control cost for two-speed, two winding motors, however, is more economical, and therefore the combined price of both motor and control for the two winding motor is only slightly higher. Because of the improved performance of the two winding motors and because of the factor of safety provided by two independent windings, the increased cost is frequently worth the difference.

Single Phase Motors

Single phase induction motors have auxiliary windings or devices for starting and are classified by the method used.

Capacitor start motors develop high starting torque in fractional hp ratings and moderate starting torque in larger ratings. They are used for constant speed drive such as fans, blowers and centrifugal pumps. During the starting period, a winding with a capacitor in series is connected to the motor circuit and when the motor comes up to speed a centrifugal switch cuts the capacitor and second winding out of the circuit.

Two-value capacitor motors develop high starting torque employing a starting capacitor and a running capacitor. The starting capacitor gives high starting ability but is suited for short time operation only and is cut out for the running condition by a centrifugal switch. The running capacitor gives high efficiency at full speed. These motors are used on compressors, reciprocating pumps and similar equipment which may start under heavy load.

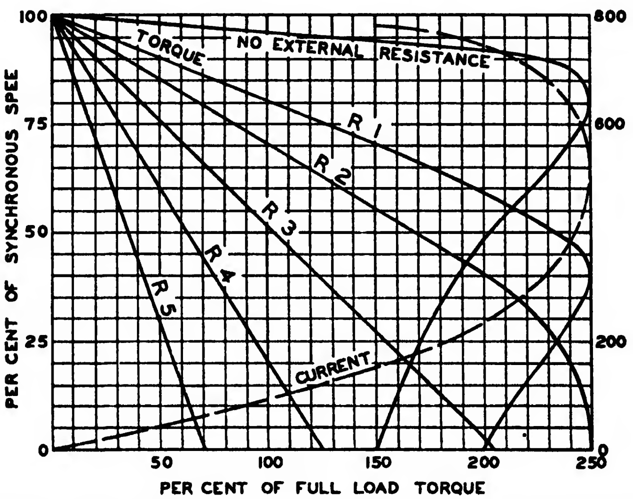

FIG. 9. PERFORMANCE CHARACTERISTICS OF A WOUND ROTOR MOTOR WITH EXTERNAL RESISTANCE

Permanent split capacitor motors have low starting torque and are ideally suited for small fan drives. Operation is similar to the capacitor start motor except that the capacitor is not cut out when running.

Repulsion-Induction motors develop high starting torque. The motors have two rotor windings—a squirrel cage for running and a wound rotor connected to a commutator for starting. No switching device is required to change from starting to running winding as this is accomplished by a gradual shift with speed in the magnetic flux path so that near rated speed the squirrel cage winding has taken over completely.

Repulsion Start-Induction Run motors are similar to the repulsion-induction motor except they have only the commutator winding. They are supplied with a centrifugal short circuiting switch which shorts the commutator bars when the motor comes up to speed to obtain a winding approximately like the squirrel cage in its function.

Repulsion-Induction and *Repulsion-Start-Induction Run* motors are suitable for applications such as industrial compressors where high breakaway torque is required and where commutator and brush noise are not factors.

Split Phase motors have a high resistance auxiliary winding which is in the circuit during starting but is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions it operates as a single phase induction motor with one winding in the circuit. These units are available for the small horsepower ratings and when equipped with a high slip rotor may be used for adjustable

Table 6. Typical Application Requirements of Synchronous Motor Drives Showing Starting, Pull-In and Pull-Out Torques

	Application	Method of Connecting Motor to Load	Starting Conditions	Torques			Remarks
				Starting	Pull-in	Pull-Out	
Fans	Exhaust and ventilating	Coupled or belted	Usually loaded	50	60–125	150	WR^2 of fan must be considered
Blowers	Cycloidal positive	Coupled or engine type	Unloaded	40–60	40–60	150	Two-speed motors sometimes used
Blowers	Blowing engines reciprocating	Engine type	Unloaded	40	40–60	150	
Blowers	Turbo high speed	Direct connected or step up gear	Unloaded (intake closed)	30	50	150	WR^2 of blower must be considered
Compressors	Air	Engine type	Unloaded	40	30	150	Flywheel effect important
Compressors	Ammonia and ammonia booster	High speed—belted Low speed—engine type occasionally coupled	Unloaded (by by-pass)	40	30	150	Flywheel effect important
Compressors	Freon	High speed—belted Low speed—engine	Unloaded (by by-pass)	45	50	150	Flywheel effect important
Compressors	Gas reciprocating	High speed—belted Low speed—engine	Unloaded (by by-pass)	40	30	150	Flywheel effect important

varying speeds through line voltage control. The motors are ideally suited for fan duty.

Speed-torque characteristics of single phase motors are shown in Fig. 10. Typical specifications for single phase motors are shown in Table 8.

CONTROL FOR ALTERNATING CURRENT MOTORS

Squirrel-Cage motors are usually linestarted where power company limitations permit. In sizes up to 2 hp the motors are started by means of manual switches with an overload current element for motor protection. In larger ratings a linestarter is usually provided with either an additional safety switch or circuit breaker for disconnecting and short circuit protection. Reduced voltage starting may be either of manual or push button controlled magnetic type. In specifying this type of starter consideration should be given to the fact that starting torque of squirrel-cage motors varies as the square of the applied voltage. For example, a motor developing 100 lb-ft starting torque on full voltage would produce only 25 lb-ft torque on starting on half rated voltage. Fig. 11 illustrates

recommended control practice for squirrel-cage motors. Typical specifications for squirrel-cage motor controls are shown in Table 9.

Wound Rotor motors require control of both primary and secondary circuits. The primary* control may be the same as for squirrel-cage motors, manual or magnetic, at full voltage. Secondary* control provides means of varying secondary resistance for starting and speed control. The secondary controller should be specified for starting duty only or for speed regulating duty. Fig. 12 illustrates recommended control practice for wound rotor motors. Typical specifications for wound rotor motor control are shown in Table 10.

Synchronous motor starters should provide pull-out protection, auto-

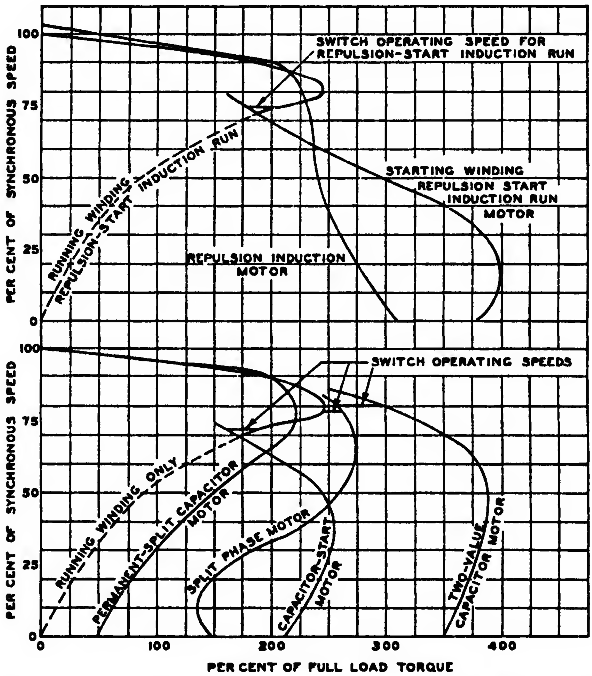

FIG. 10. SPEED-TORQUE CHARACTERISTICS OF SINGLE PHASE MOTORS

matic synchronization or automatic stopping of the motor after pull-out, and insurance of complete starting sequence, as well as overload and low voltage protection. The control may be either magnetic or semimagnetic at full or reduced voltage. Semi-magnetic starters provide automatic field control but require hand operation for closing the line contactors to start and transfer to full voltage.

In applying reduced voltage starters to synchronous motors it should be remembered that, since these motors are started on damper windings and during the acceleration period function similarly to squirrel-cage motors, the starting torque varies as the square of the applied voltage. Consideration should be given to insure development of sufficient motor torque to accelerate the load. Typical specifications for synchronous motor control are shown in Table 11.

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*Refer to Glossary at end of chapter.

TABLE 7. TYPICAL SPECIFICATIONS FOR SYNCHRONOUS MOTORS

..........Hp, ...RPM, ...Per Cent P.F., ...Volts, $\begin{Bmatrix}2\\3\end{Bmatrix}$ Phase, $\begin{Bmatrix}25\\60\end{Bmatrix}$ Cycles,

Synchronous Motors of the $\begin{Bmatrix}\text{Belted}\\\text{Coupled}\\\text{Engine}\end{Bmatrix}$ Type for Driving........................
(Application)

Motor Shall be Arranged for $\begin{Bmatrix}\text{Horizontal}\\\text{Vertical}\end{Bmatrix}$ Operation and Shall Be Provided With

A .... Type of Enclosure. Motor Shall be Capable
(Open, Splashproof, etc.)

of Developing A Starting Torque of ...... Per Cent Full Load Torque, A Pull-In Torque of ...... Per Cent Full Load Torque, And A Pull-Out Torque of ...... Per Cent Full Load Torque. The Motor Field Shall Be Excited From A

$\begin{Bmatrix}\text{Direct Connected Exciter}\\\text{Belted Exciter}\\\text{M-G Set Exciter}\\\text{d-c Bus}\end{Bmatrix}$ Which $\begin{Bmatrix}\text{Shall}\\\text{Shall Not}\end{Bmatrix}$ Be Included With The Motor.

TABLE 8. TYPICAL SPECIFICATIONS FOR SINGLE PHASE MOTORS

..........Hp, ...RPM, $\begin{Bmatrix}25\\60\end{Bmatrix}$ Cycles, $\begin{Bmatrix}115\\230\end{Bmatrix}$ Volts, Single Phase Motor of

the .... Type for Driving .... Motor Shall
(Capacitor, Split Phase, etc.) (Application)

Be Arranged for $\begin{Bmatrix}\text{Horizontal}\\\text{Vertical}\end{Bmatrix}$ Mounting And Shall Be Provided With

a .... Type of Enclosure, *NEMA* $\begin{Bmatrix}\text{Class A}\\\text{Class B}\end{Bmatrix}$ Insulation, And
(Open, Splashproof, etc.)

$\begin{Bmatrix}\text{Ball}\\\text{Sleeve}\end{Bmatrix}$ Bearings.

TABLE 9. TYPICAL SPECIFICATIONS FOR SQUIRREL-CAGE MOTOR CONTROL

Control for Squirrel-Cage .... Motors Shall Consist of An Enclosed
(Fan, Pump, etc.)

$\begin{Bmatrix}\text{Manual}\\\text{Magnetic}\end{Bmatrix}$ Type $\begin{Bmatrix}\text{Across The Line}\\\text{Reduced Voltage}\end{Bmatrix}$ Starter Providing Overload Protection And

Low Voltage $\begin{Bmatrix}\text{Protection}\\\text{Release}\end{Bmatrix}$. Control Shall Include A Safety Disconnect Switch

$\begin{Bmatrix}\text{Separately Mounted}\\\text{In a Common Enclosure}\end{Bmatrix}$ With the Starting Controller.

Multi-Speed control may be either manual or magnetic, and at full or reduced voltage. When using automatic magnetic control with two-, three-, and four-speed separate winding or consequent pole motors, control may be obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button. This is

known as selective speed control. It is commonly used in the smaller theater installations where the fan and motor are located backstage and the speed control is located in the lobby.

Multi-speed motor controllers may be provided with compelling relays which make it necessary for the operator to press the first speed button before regulating the motor to the desired speed. This insures that the motor is always started at low speed before adjusting to a higher speed.

Timing relays which provide for automatic acceleration may be used for control. With this feature the motor will always start at low speed and automatically accelerate to the desired speed. Decelerating relays may be used to reduce the shock effect of the braking action to the motor and drive when the speed is reduced from a higher to a lower speed.

Single Phase motor control usually consists only of a linestarter, either

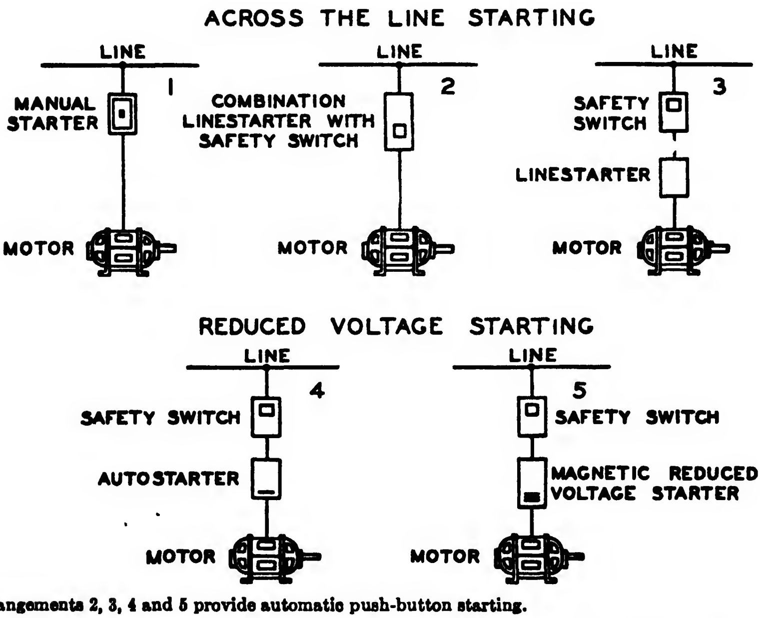


FIG. 11. RECOMMENDED CONTROLS FOR SQUIRREL-CAGE MOTORS

manual or magnetic. In some cases it is desirable also to provide a disconnect switch. Fig. 13 illustrates the recommended controls.

Typical specifications for single phase motor control are shown in Table 12.

GEAR MOTORS

A gear motor is a self-contained combination of any type of a-c or d-c motor and an enclosed speed-reducing gear, providing a more compact and readily adaptable unit than is obtained by using a motor coupled to a gear reducer. Gear motors are available in sizes up to 75 hp with output shaft speeds from about 4 to 1430 rpm, making it possible to couple or to connect by gear or chain to nearly any machine. High speed motors are used, generally 1800 rpm on 60 cycles, thus obtaining the advantages of

Table 10. Typical Specifications for Wound Rotor Motor Control

Control for Wound Rotor Motor Control for .. $\underset{\text{(Fan, Pump, etc.)}}{}$ Applications Shall Consist of a Safety Disconnect Switch, a $\left\{\begin{matrix}\text{Separately}\\ \text{Common}\end{matrix}\right\}$ Mounted Across the Line Starter Providing Overload Protection, Low Voltage $\left\{\begin{matrix}\text{Protection}\\ \text{Release}\end{matrix}\right\}$, and a Secondary $\left\{\begin{matrix}\text{Starting}\\ \text{Speed Regulating}\end{matrix}\right\}$ Controller for $\left\{\begin{matrix}\text{Separate Mounting}\\ \text{Mounting in Same Enclosure}\end{matrix}\right\}$. Primary and Secondary Control Shall Be Interlocked so as to Provide Complete Control from the Rheostat Handle.

Table 11. Typical Specifications for Synchronous Motor Control

Synchronous Motor Control for .... (Application) Motors Shall Provide for $\left\{\begin{matrix}\text{Full Voltage}\\ \text{Reduced Voltage}\end{matrix}\right\}$ Starting and Shall Be $\left\{\begin{matrix}\text{Full Magnetic}\\ \text{Semi-Magnetic}\end{matrix}\right\}$. (Reduced Voltage Starting Shall Be Obtained by Means of $\left\{\begin{matrix}\text{Autotransformers}\\ \text{Resistors}\\ \text{Reactors}\end{matrix}\right\}$ And Shall Limit The KVA Inrush to a Maximum of ... per cent of Full Load KVA.) The Control Panel Shall Be For $\left\{\begin{matrix}\text{Isolated}\\ \text{Switchboard}\end{matrix}\right\}$ Assembly And Shall Be of $\left\{\begin{matrix}\text{Open Switchboard}\\ \text{Enclosed Cubicle}\end{matrix}\right\}$ Construction. It Shall Provide Overload, Under Voltage Protection And After Pulling Out of Step Will $\left\{\begin{matrix}\text{Automatically Resynchronize}\\ \text{Automatically Stop}\end{matrix}\right\}$ The Motor.

Table 12. Typical Specifications for Single Phase Motor Control

Control For Single Phase .... (Fan, Pump, etc.) Motors Shall Consist of a $\left\{\begin{matrix}\text{Manual}\\ \text{Magnetic}\end{matrix}\right\}$ Type $\left\{\begin{matrix}\text{Across The Line}\\ \text{Reduced Voltage}\end{matrix}\right\}$ Starter Providing Overload Protection (And Low Voltage); And A Separate Safety Disconnecting Switch.

high motor power factor and efficiency. The gearing efficiency is also high, usually about 98 per cent for a single reduction of the helical or spur type, that is, a 2 per cent loss for one reduction or 4 per cent loss for a double reduction. Consequently, the over-all performance of the gear motors is much higher than a combination of open gearing, belting, countershaft, or other arrangement, which would otherwise be required. Gear motors are used extensively to drive numerous types of slow speed drives. Besides being more effective than other combination drives in saving space, they are important in reducing maintenance and operating hazards.

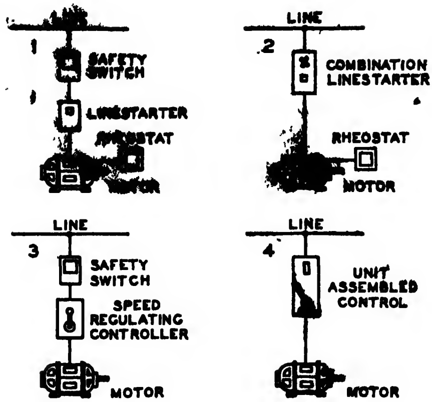
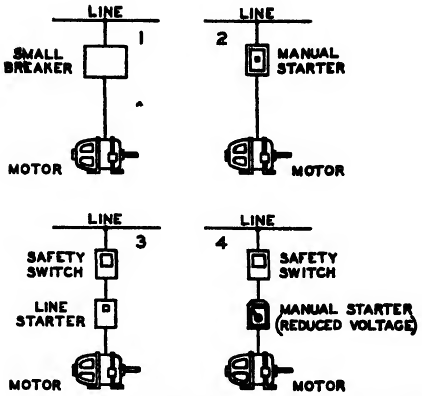


Fig. 12. Recommended Controls for Wound Rotor Motors



Arrangements 2, 3 and 4 are optional for motors up to 7½ hp, 220 volts.

Fig. 13. Recommended Controls for Single Phase Motors

GLOSSARY

General Definitions

NEMA is the abbreviation for the *National Electrical Manufacturers Association.*

Speed Regulation (d-c motors) is the change in speed between no-load and full-load, expressed in per cent of full-load speed; for example, a motor having a no-load speed of 1200 rpm and a full-load speed of 1140 rpm would have a speed regulation of 5.6 per cent.

Slip (a-c induction motors) is the difference between the motor speed and synchronous speed expressed in per cent of synchronous speed, *e.g.*, a 1200 rpm motor operating at 1140 rpm would have a slip of 5 per cent.

Torque is an expression of the turning effort developed by the motor at the shaft and is usually expressed in ounce-feet for fractional horsepower motors and in pound-feet for motors of larger ratings.

Primary is the term usually applied to the high voltage or line side of a transformer or motor. In the case of the wound rotor motor the primary is the stator winding.

Secondary is the term usually applied to the low voltage or load side of a transformer or motor. In the case of the wound rotor motor the secondary is the rotor winding.

NEMA Classification of Motor Enclosures

Open motors (40C rise, rated load, 50 C rise, service factor load) are self-ventilated machines having no restriction to ventilation other than that necessitated by mechanical construction.

Protected motors (50 C rise) have all ventilating openings in the frame protected by perforated covers.

Semi-Protected motors (50 C rise) have the ventilating openings in the top half of the frame only protected by perforated covers.

Drip Proof motors (50 C rise) are so constructed that drops of liquid or solid particles falling on the machine at any angle not greater than 15 deg from the vertical cannot enter the machine either directly or by striking and running along a horizontal or inclined surface.

Splash Proof motors (50 C rise) are so constructed that drops of liquid or solid particles falling on the machine or coming towards it in a straight line at any angle not greater than 100 deg from the vertical cannot enter the machine either directly or by striking and running along the surface.

Totally Enclosed Non-Ventilated motors (55 C rise) are so constructed as to prevent exchange of air between inside and outside of the case, but are not air tight and are not equipped with external cooling means.

Totally Enclosed Fan-Cooled Motors (55 C rise) are similar to totally enclosed, non-ventilated machines except that exterior cooling is provided by means of a fan or fans integral with the machine.

Explosion Proof motors (55 C rise) have an enclosing case designed to withstand an explosion of a specified gas or vapor which may occur within it, and to prevent the ignition of the gas or vapor surrounding the motor by sparks, flashes, or explosions of the gas or vapor which may occur within the machine casing.

Dust Explosion Proof motors (55 C rise) have an enclosing case designed and constructed so as not to cause the ignition or explosion of an atmosphere of the specific dust or to cause ignition of dust on or around the machine. (Proper overload protection and cleanliness are required for successful operation).

Water Proof motors (55 C rise) are so constructed as to exclude water applied in the form of a stream from a hose.

Dust Tight motors (55 C rise) are so constructed that the enclosing case will exclude dust.

Motor Speed Classifications

A Constant Speed Motor is one in which the speed remains practically constant with changes in load; *e.g.*, a d-c shunt wound motor or a-c squirrel-cage motor with low slip.

A Varying Speed Motor is one in which the speed varies with the load, usually decreasing when the load increases; *e.g.*, a d-c series motor or an induction motor with large slip.

An Adjustable Varying Speed Motor is one in which the speed can be adjusted gradually, but when once adjusted for a given load will vary in considerable degree with change in load; *e.g.*, a shunt wound d-c motor adjusted by armature resistance control.

An Adjustable Speed Motor is one in which the speed can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load; *e.g.*, a d-c shunt motor with field resistance control. The standard ratings for open type, adjustable speed motors, having a speed range of 3 to 1 and greater are in accordance with the following:

(1) A standard continuous horsepower rating at 150 per cent of minimum speed with a temperature rise of 40 C.

(2) The next higher standard continuous horsepower rating at 3 times minimum speed with a temperature rise of 40 C.

(3) Between 150 per cent of minimum speed and 3 times minimum speed, the standard continuous horsepower rating with a temperature rise of 40 C will vary with the speed along a straight line connecting these two horsepower ratings. No further increase in horsepower is recognized above 3 times minimum speed.

(4) Below 150 per cent of minimum speed the lower continuous horsepower rating (see preceding item 1) will apply with a temperature rise of 50 C.
Example: 20/25 hp, 400 to 1600 rpm. This motor may be rated 20 hp, 40 C at 600 rpm and 25 hp, 40 C from 1200 to 1600 rpm. Between 600 and 1200 rpm the rated horsepower increases directly with speed from 20 to 25 hp.

(5) Motors may also be rated 1 hour with temperature rise of 50 C with the higher horsepower rating (see preceding item 2) throughout the entire speed range.
Example: 20/25 hp, 400 to 1600 rpm. This motor may be rated 25 hp, 50 C. 400/1600 rpm; 1 hour.

Mechanical Modifications

Vertical Mountings are available for such applications as pumps, agitators, and so forth. This type of application may require a special umbrella-type hood to protect against dripping liquids.

Flanged Mountings are available for use where motors are built in as part of machines. Motors may also be supplied with flush plate mountings, suitable for close coupled pump and similar applications.

CHAPTER 36

UNIT AIR CONDITIONERS, UNIT AIR COOLERS

Definitions, Classification of Unit Type Equipment, Component Parts of Unit Type Equipment, Sound Isolation, Modification of Remote Units, Rating of Unit Air Conditioners, Application of Unitary Equipment, Unit Air Coolers

THIS chapter presents the physical characteristics of air cooling units and air conditioning units; a suggested procedure for selection of units; and some of the factors involved in the application of unitary equipment. In general, factory produced unit equipment can be obtained to accomplish all of the functions possible from field assemblies, but the advantages of unit equipment are most apparent in small and moderate capacities. Above 12,000 cfm capacity, or approximately 40 tons of refrigeration capacity, handling and assembly costs generally favor the use of field assembled units. Multiple application of unitary equipment is frequently justified for large gross tonnage installations where zoning or a minimum amount of air distributing ducts is desirable.

DEFINITION

The term *air conditioning unit* has been loosely used as a name for all types of factory produced air handling, cooling, or heating units. The code, Standard Method of Rating and Testing Air Conditioning Equipment[1], defines the various types of unitary equipment:

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.
2. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer with control means for maintaining temperature and humidity within prescribed limits.
3. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer with control means for cooling and maintaining temperature and humidity within prescribed limits.
4. A *Self Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified[2] according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).
5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.
6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.
7. A *Forced-Circulation Air Cooler* is a factory encased assembly of elements by which heat is transferred from air to refrigerants[3].

CLASSIFICATION OF UNIT TYPE EQUIPMENT

Field assembled apparatus as described in Chapter 43 can be designed in shape, size, and capacity for any application, with the refrigeration and

heating system exactly balanced to load conditions. To obtain the economies of mass production, factory built units must be standardized in a few models per manufacturer. Each model covers a range of capacities within the capacity of its fan to deliver air against the resistance of the unit and against the system resistance. For this reason, the unit performance will usually represent a compromise between actual load requirements and the rated capacity. Within the range of accuracy of most load calculations, this compromise is not objectionable.

If the condensing unit and cooling and heating coil surfaces are carefully selected, and if proper consideration is given to reduction of piping losses, the performance of the combined system will compare favorably with field assembled apparatus. A system, in which the air handling unit is separated from the condensing unit, is called a remote system, and the conditioning unit is designated as a remote unit. The economical capacities of remote units usually range from 10 to 40 tons.

For applications where load calculations are subject to considerable variance and where close control is not considered essential, further economies of factory assembly can be obtained by combining the air handling and condensing equipment in one unit. This effects another compromise between load calculations and equipment selection, since the capacity of the combined unit is then dependent on the predetermined balance between a particular coil and condensing unit. These combination units are called self-contained units and, under the aptly descriptive name *store conditioners*, find economical application in 3, 5, 7½, 10, and 15 ton refrigeration capacities. These capacities, or limited multiples thereof, meet the load requirements of the majority of small and medium sized commercial establishments. With some modifications, these units can also be adapted to light industrial work.

To meet the requirements of individual comfort in small rooms and offices, where load calculations are subject to the indefinite design condition of *feeling cool*, self-contained units, called room coolers, find extensive and economical application. These units are usually restricted to summer and intermediate season operation, and range from ⅓ to 1½ tons of refrigeration capacity.

A special application of remote units is found in the unit air cooler which is used extensively in refrigeration work. Its primary function is to reduce temperatures in insulated and sealed storage spaces, and humidity control is a secondary consideration. Because of the small temperature differences between the coil and room temperatures, unit coolers handle three to five times as much air per ton as remote units used in air conditioning.

The attic fan or exhaust fan is sometimes referred to as a cooling unit, but since it contains no element of heat transfer, it is treated in Chapter 32, Fans.

COMPONENT PARTS OF UNIT TYPE EQUIPMENT

Units can be obtained for producing any of the required effects on air. As they function most satisfactorily when doing the work for which they were designed, field modifications are usually unadvisable because of expense involved as well as the possibility of causing unexpected difficulties in operation. The basic design considerations of unitary equipment are discussed in the next following paragraphs.

Remote Units. Remote units can be obtained in two general classes, horizontal as shown in Fig. 1 and vertical as in Fig. 2. Their construction is essentially the same, except for the drain pan and filter locations.

Casings. Casings are generally constructed of sheet metal with angle iron frames and with removable panels for access to coil connections, blower bearings, filters, and drain. Casings should be air tight. Panels should be tight fitting with cam or similar fastenings for easy opening. Panel openings at coils for heavy units should be large enough to receive coils after the casing is suspended. Frames should be fitted with lugs strong enough to suspend horizontal units. Non-metallic casings are of advantage in small remote units for reducing sound, particularly when propeller type fans are used.

Insulation. Remote units are available with waterproof and vermin-proof sound and heat absorbing insulation on the inside of the casing. They are also available with flanges and flanged access doors to permit insulation after installation.

Drain Pans. Because of the corrosive effect of mild picric, carbonic, and sulfurous acids absorbed by condensate, drain pans are usually made of 14 gage or heavier metal. They should be hot dipped galvanized after fabrication or otherwise treated to resist corrosion. Some manufacturers extend the drain pan under the entire unit, but in any case it should extend far enough to catch any condensate carried over from the coils. The drain connection should be readily accessible for cleaning and, in air conditioning work, should be generously sized and trapped. Some municipal codes require a minimum size of 1¼ I.P.S.

Blowers. The usual practice among manufacturers is to use light construction in the blowers in remote units, although a few are available with heavy duty blowers in the larger sizes. These blowers work under almost constant conditions without overload or shock and will usually last as long as the unit with reasonable maintenance. As lubrication of bearings is very important, it is good practice to locate the oil cups conveniently outside of the unit. In any application where considerable dehumidification or humidification is involved, such as in a system where the unit is handling 100 per cent outside air, it is important that the blowers be painted with asphaltum or other corrosion resistant paint to prevent excessive oxidation and corrosion of the blowers.

Some of the smaller suspended type units, Fig. 3, use propeller fans with a trailing edge blade in order to obtain required pressure characteristics with quiet operation. Since most of the motors driving these fans are direct connected and use brushes for starting, adequate access should be provided for maintenance and inspection.

Cooling Coils. The cooling and dehumidifying coils used in unit air conditioners are essentially the same as those used in central station units. The face area of the coil is usually fixed and the number of rows deep in the direction of air flow is the variable that determines capacity. It should be remembered, as noted in Chapter 39, that adequate coil surface is important for efficient performance of any system and that there is little economy in reducing coil depth to less than four rows.

Where multiple circuits are used in the larger coils, equal distribution of the cooling medium to the various circuits is vital in order to develop full capacity of the coil. The cooling coils also perform the function of dehumidification. To prevent carryover of condensate, eliminator plates

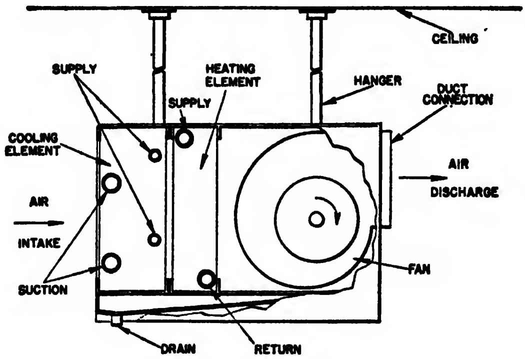


Fig. 1. Horizontal Remote Type Unit Air Conditioner

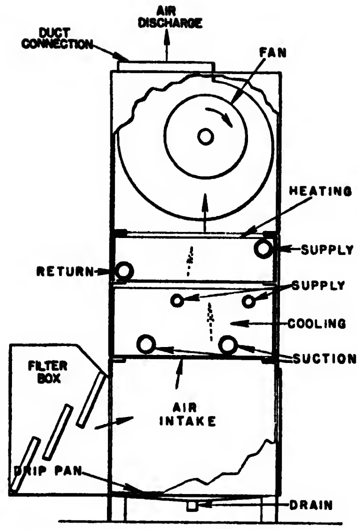


Fig. 2. Vertical Remote Type Unit Air Conditioner

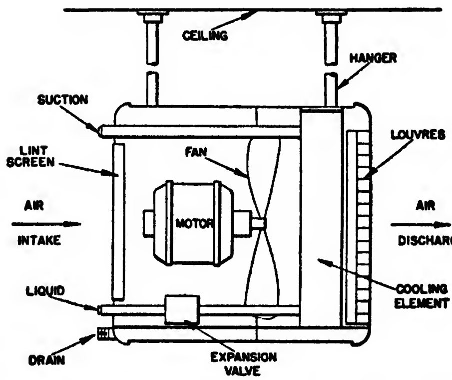


Fig. 3. Suspended Propeller Fan Type Unit Air Conditioner

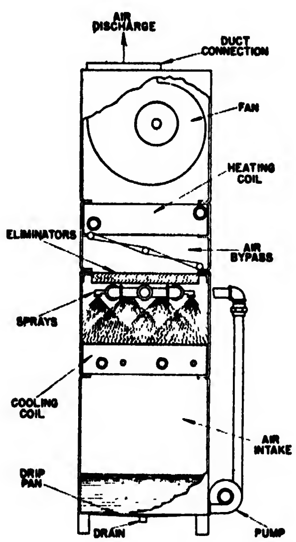


Fig. 4. Spray Type Remote Unit Air Conditioner

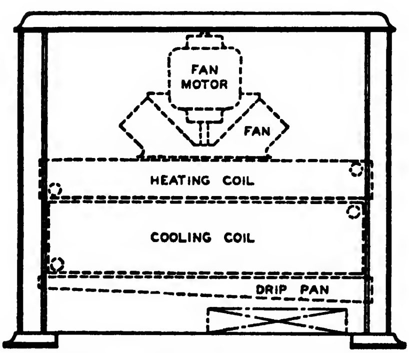


Fig. 5. Remote Floor Type Room Unit Air Conditioner

should be used if face velocities exceed 500 to 530 fpm, unless adequate means of catching the droplets are provided.

When air is drawn upward through dehumidifying coils as in some vertical units, water is entrained within the fins and held in suspension. This increases the resistance pressure against which the blower operates and results in wide variation in air volumes handled between dry and wet coil conditions. Some unit manufacturers have so designed vertical units that the air passes through the coils horizontally in order to overcome this difficulty.

Heating Coils. Heating coils of unit air conditioners are usually conventional blast coils and can be obtained with or without non-freeze steam distribution features. They usually match the cooling coils in face area and are one or two rows deep depending on the heating requirements. Where coils are selected for hot water and have more than two rows of tubes, the air resistance and the space requirements of the total number of rows of cooling and heating coils should be carefully checked.

Humidification. Spray type humidifiers are usually used in remote systems, but in some cases pan type humidifiers or steam humidifiers are also used. Some condensation on the inside of the unit casing may occur with possible water damage if the unit is located in a cold space without adequate insulation. Spray type humidifiers should be located so that no carryover of moisture occurs.

Filters. It is almost axiomatic that all units should have filters. Some small suspended units of 1 ton capacity or less with low coil face velocities and propeller fans are equipped only with lint screens or operate without filters, but in them the coils must be periodically cleaned and there is constant danger of clogging of the drain with the possibility of water damage. Filters used are usually of the throw-away type, although cleanable filters are available for most of the larger units. Care should be taken to insure adequate filter surface, since the cross-sectional area of the unit is seldom adequate for filter area. V-shaped or staggered filter arrangements are quite commonly used to increase filter area.

Motors. In some units where the motor is mounted inside, adequate access for maintenance and clearance for tightening belts are imperative. When motors are mounted in this manner, all of the heat equivalent motor input must be added to the heat load to be absorbed by the system and this requires a lower exit air temperature at the coils. Usually, however, the motor is located outside of the casing where it is readily accessible for service. In this case, only the brake horsepower required by the fan is transformed into heat to be included in the load calculations.

In either application, motors should be selected with adequate horsepower to handle the design volume of air against the resistance of the system when the coils are wet and then checked against the possible horsepower requirements for the increased volume of air when the coils are dry.

SOUND ISOLATION

Both suspended and vertical floor mounted units can transmit vibration through the supports. Wherever such transmission of sound might be objectionable, the supports should be isolated through rubber-in-shear or other sound deadeners (for design of suitable sound deadeners see section Controlling Vibration from Machine Mountings in Chapter 42).

MODIFICATIONS OF REMOTE UNITS

Features of various modifications of remote air conditioners are given in the following paragraphs.

Spray Type Unit

Fig. 4 shows a spray type unit used by designers who prefer air washing and coil wetting features. These units are equipped with a pump that sprays water or brine over the coils. Due to the direct mixing of the condensate and the spray, provision must be made for overflow in summer and replacement of water evaporated in winter.

Dehumidifying Units

In a further modification of spray type units, absorbent brine solutions such as lithium chloride are used to remove moisture from the air. As explained in Chapter 38, the latent heat of the moisture removed is changed to sensible heat, so that coils must be used as after-coolers to obtain the right dry-bulb temperatures. Factory produced units are also available for use with solid adsorbents such as silica gel.

Remote Room Units

For individual rooms, with cooling load requirements of ½ to 1½ tons, remote units are available in attractive casings for installation within the room. A suspended type is shown in Fig. 3 and a floor type, such as is usually installed in place of an existing radiator, is shown in Fig. 5. Furnished with chilled water from a central plant, these units offer a satisfactory method of conditioning existing offices, hotel, and apartment rooms. These units may be obtained with filters and outside air connections, but for most satisfactory application are used as supplements to central systems that supply properly conditioned and filtered air to the areas served.

Induction type units, using primary conditioned air under pressure to induce local circulation, are described in Chapter 43.

Self-Contained Units

A typical large self-contained unit is shown in Fig. 6. It is essentially a remote vertical type conditioner mounted on top of a sound insulated enclosure containing the condensing unit. Air distribution is obtained by means of grilles mounted in the discharge plenum when the unit is located in the conditioned area. Duct distribution of conditioned air can be obtained by removing the plenum connecting directly to the blower discharge, and safing the top of the unit.

The heat generated by the compression of refrigerant gases and that given off by the electric motor is removed from the compressor compartment in four ways: by the use of a water coil in the compressor compartment; by utilizing the cold suction gases; by drawing part of the return air through the compressor compartment, and finally by circulating room air through the compressor compartment by means of a fan attached to the motor shaft.

The 7½, 10, and 15 ton self-contained units usually have horizontal type conditioners. The condensing unit enclosures are not completely sound insulated, since they are not usually installed in the conditioned area. Most units can be divided into two or three sections for ease of

handling and installation. Although most large self-contained units have water-cooled condensers, the 7½, 10, and 15 ton units can be obtained for operation with evaporative condensers.

Self-Contained Room Cooling Units

Small self-contained units can be obtained with water-cooled condensers but they are generally air cooled.

The air-cooled types are small in capacity, ranging from ⅓ to 1½ hp. Their principal application is for conditioning such spaces as hotel rooms, offices and residential living quarters. A duct connection between the unit and an outside window or ventilated air shaft is required to permit

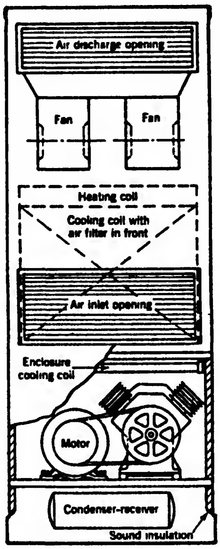


FIG. 6. SELF-CONTAINED WATER-COOLED AIR CONDITIONER

disposal of the heat extracted from the conditioned area. The unit may stand in front of the window or be mounted on the window sill. Various styles and types of windows are encountered which increase the difficulty of making the window connections. The evaporation of condensate on the condenser coils, as a means of disposing of this moisture, tends to increase the condensing capacity and reduce the operating head pressure. Some units add supplementary water so that increased capacity may be obtained from constantly wetted condenser coil surface. Connections to an electrical outlet may be by means of a conventional cord and plug or a permanent electrical connection, depending on local code rulings pertaining to the installation of small motors. The exterior finish of the unit in metal, wood or fabric is decorated to harmonize with office or bedroom furnishings.

A unit of the air-cooled condenser type for floor mounting is shown in Fig. 7. Of the two fans shown, the lower one acts as condenser air fan, and in some units this fan is arranged with slingers for discharging condensate on the condenser coil while the upper fan discharges air into the conditioned area. A feature of the design shown in Fig. 7 is that the

condensate from the cooling coil is sprayed over the condenser surface and vaporized, thus eliminating the need for drain connections. A simple dampering arrangement is generally provided for exhausting some air from the room, in addition to introducing outside air and recirculating required amounts of air. It is possible to remove the equipment for winter storage or utilize the ventilating features for winter operation.

Controls for Room Cooling Units

Control devices for self-contained cooling units are generally provided to include all necessary means for automatic operation. Provision is also made for adding auxiliary external controls when desired. Remote units are not generally equipped with controls. Control systems for remote units, and auxiliary controls for self-contained units, are covered in the general treatment of Controls in Chapter 34.

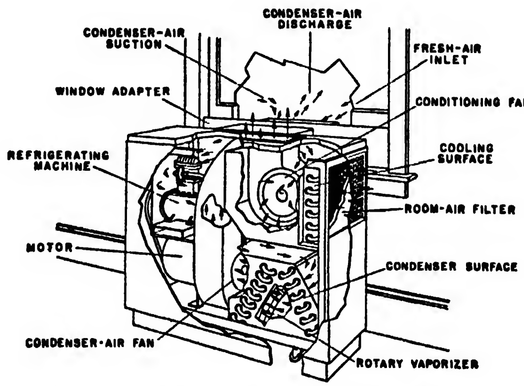


Fig. 7. Self-Contained Air-Cooled Unit Air Conditioner

RATINGS OF UNIT AIR CONDITIONERS

There are two codes governing the rating and testing of unit air conditioners. The first code, Standard Method of Rating and Testing Air Conditioning Equipment[1], covers all types of air conditioning units except the self-contained type. The latter is covered by the second code, The Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling[2]. The two codes are necessary because of the basic difference caused by the heat given up by the self-contained units. The standard rating conditions for self-contained unit air conditioners, as given in the code, are set forth in Table 1.

The standard rating of a self-contained unit for the conditions specified in Table 1 includes all items which apply to the function of a unit as: (1) name of unit, (2) functions which unit performs, (3) data on cooling, (4) data on heating, (5) data on air flow, and (6) data on humidification.

The standard rating conditions for unit air conditioners, other than the self-contained type, are identical with those in Table 1 except the entering wet-bulb temperature for cooling is expressed as 50 per cent relative humidity (66.7 F wet-bulb) instead of 67 F wet-bulb temperature. In

addition, the saturated suction refrigerant temperature for comfort cooling is specified at 40 F. This condition is omitted from Table 1 for self-contained units as immaterial in the rating of a unit that includes the evaporator and condensing unit.

TABLE 1. STANDARD RATING BASIS FOR SELF-CONTAINED AIR CONDITIONING UNITS

FUNCTIONS	TYPES OF UNITS	RATING CONDITION		
		Item	Description	Value
All	All	a	Barometric Pressure	29.92 in. Hg.
Cooling	Water-Cooled, Air-Cooled and Evaporatively-Cooled Condensers	b	Unit Ambient and Air Entering Room—Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 80 F 67 F
		c	Ventilation Air	See Note
	Water-Cooled Condensers	d	Water Temperature Entering Unit	75 F
		e	Water Temperature Leaving Unit	95 F
	Air-Cooled and Evaporatively-Cooled Condensers	f	Air Entering Outside Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 95 F 75 F
Heating	All Types Provided with Heating Function	g	Unit Ambient and Total Air Entering Unit	70 F
		h	Heating Medium, Pressure or Temperature (1) Dry Saturated Steam (2) Water In (3) Water Out	 16.7 lb per sq in. abs 180 F 160 F
Humidifying	All Types Provided with Humidifying Function	i	Unit Ambient	70 F
		j	Total Air Entering Unit (1) Dry-Bulb (2) Wet-Bulb	 70 F. 53 F
Air Circulation	All	k	Filters	New and Clean

Note: Rating shall be based on both ventilation and recirculated room air entering at 80 F dry-bulb and 67 F wet-bulb temperature. (The *Note* as given in the code has been condensed in order to remove material not pertinent to this chapter).

APPLICATION OF UNITARY EQUIPMENT

One of the chief advantages resulting from use of factory produced units is the saving in installation and field assembly labor, a factor that should always be kept in mind when selecting a location for these units. Because of their compactness, the tendency exists to put them in closets, storage rooms, and other inaccessible places, where installation is so difficult that much of this cost advantage is lost.

Access panels are provided on units for the proper servicing and maintenance of the equipment. Adequate outside clearance at these panels is

essential. Wherever there is danger of freezing of coils or clogging with dirt, sufficient clearance should be available for replacing them without removing parts of the building.

The outstanding source of difficulties in unitary systems is usually dirty filters. The characteristics of the light weight fans used are such that air volume drops off rapidly with increase in static resistance. Since changing filters is an unpleasant duty likely to be neglected unless it can be done easily, the operator should be at the same level as the filters, rather than under them, when they are being removed.

Fan speeds should be selected accurately for the system resistance.

Variable pitch motor pulleys are often provided in a unit for minor field adjustment of air volume. Any such field adjustment should be made when filters are dirty, to simulate average operating conditions.

Because varying sizes of coils are used within the same casing, it is very important that coil safing be carefully installed to prevent by-passing of unconditioned air. If coils are not equipped with individual casings, additional safing may be required on top to prevent short circuiting air down through the upper edges of the fins. In this same category is the need for careful installation of the various sections of sectionalized units, using a sealing compound if necessary to prevent air leakage into the fan section of the unit.

Since the drain connection is usually made on the exit side of the coil, it is important that the drain line be properly sealed. This seal should be at least twice as deep as the suction on the fan in inches of water, to prevent gurgling sounds and to insure a positive seal against infiltration of odors and moisture laden air. Drain pans should not be used to support the coils unless they are designed to hold this weight without sagging. As the movement of air draws the condensate or excess humidification water toward the fan, drain connections are usually located on the exit side of the coil. The advantages of quick drainage are lost if improperly supported coils distort drain pans and cause water to accumulate in the center or back of the pan.

When the fans of vertical units are stopped, condensate that has been held up in the coils by fan suction drops into the drain pan and splashes against the casing. If water damage is to be avoided, flashings should be provided to prevent this water from running out of the unit at the seams.

When locating unitary equipment, floor and beam loadings should be carefully checked. Suspended horizontal units can add 50 to 100 lb per square foot to the loading on the floor above. Should this floor be already heavily loaded, or be a roof structure designed for a 40 lb per square foot snow load, excess beam deflection may occur and cause cracking of plaster or concrete fire-proofing. A small fire, normally of little consequence, may cause a rupture of a heavily loaded structure and permit the equipment to drop with extensive property damage. Self-contained units should be carefully installed since their weights run as high as 200 lb per square foot. When they are installed in street floor shops, the extra precaution of placing a column beneath them in the basement is an inexpensive method of reducing vibration as well as providing insurance against overloaded floor beams.

The services required for operation of unitary equipment should conform to the many restrictive, but necessary, local municipal codes. Existing buildings seldom are wired adequately for the electrical load imposed by the starting of an air conditioning compressor on any branch

circuit. Even the smallest room cooler can draw enough current to reduce the voltage of a lighting circuit to the point where it is visibly apparent. This voltage drop may even affect the life of the unit due to the relatively slow starting. The cost of a separate electrical circuit of adequate capacity from the main panel is more than justified; it is a necessary expense in the majority of installations.

A water supply of adequate capacity and pressure is necessary to prevent overloading of electrical equipment by high head pressures. The average city water supply pressure is adequate for installations up to the third floor. Since most water cooled units require about 20 lb pressure, including control valve losses, it is important that any units served by gravity from roof tanks be checked carefully if located less than 40 ft below the tank.

Drain connections from condensers should flow to an open and properly trapped sink as required by most city codes. This prevents back pressures on the city water system in the event of condenser failure. A check valve should also be installed in the water supply as a further precaution against contamination.

When installing small remote or self-contained units with outside air connections in buildings more than 6 stories high, the effect of wintertime stack action in elevator and stairwells requires special attention. This stack action is the cause of negative pressures on the lower floors, tending to draw cold air through the units, and positive pressures on the upper floors preventing adequate ventilation and disrupting air distribution. It can also cause annoying whistling at door openings that is a serious source of complaint in hotels and offices. Wherever the removal of such units is impracticable, it is important that carefully fitted, felt-edged dampers be installed in the outside air intakes with adequate locking devices.

One further consideration when installing self-contained units in conditioned areas is that any maintenance or repairs to be required in future years must be carried on in occupied space. If this is kept in mind in locating units, much inconvenience can be avoided.

UNIT AIR COOLERS

This type of unit is primarily intended to perform the main function of cooling air, with humidity control a secondary function within the limitations of the design. The main application of this equipment is in process and product refrigeration, such as cold storage warehousing, fruit and vegetable packing, in breweries, and in wholesale and retail food markets.

Application of the unit method of air cooling with mechanical circulation is comparatively recent, being an improvement over the pipe or finned coil, which depended on gravity for circulation. Bunkers were sometimes constructed around the coils to direct the air flow and sometimes fans were used for forcing air over the coils. The location of the unit air cooler is usually within the refrigerated area, but the larger, blower type models may be remotely located.

Design and Performance. Greater application and use of commercial refrigeration have resulted from the development of the unit air cooler. Flexibility of design has permitted almost any condition to be met. Finned type coils are usually employed, with continuous fan operation. By varying such physical features as tube size, fin spacing, refrigerant circuiting, the depth of coil rows, and air volume over the coils, the designer is able to produce a wide range of performances and to offer many desirable

features not obtainable with the coil and bunker method. Higher suction temperature operation, more uniform temperatures, higher relative humidities with the defrosting cycle, moderate first cost, and a minimum of installation expense are likewise factors in their development.

New uses have appeared for unit air cooler application in industrial and commercial processes involving both the raw materials and finished product, where the maintenance of low temperatures is a necessary part of these processes. Of particular interest is the new field of extreme low temperature application where many new uses for refrigeration are being found.

Types of Units. The two standard types are the suspended or ceiling type, and the vertical or floor mounted type. There are variations of these such as the panel type which is wall mounted and arranged to take in air from the lower section and discharge it from the upper section.

The ceiling type has the appearance of a unit heater, with its propeller type fan blowing air through a bank of coils. Singly or in combination, they are easily installed and occupy little or no useful space. Alterations may be accomplished with little cost by relocating units or adding additional ones for increased capacity.

The floor mounted types employ blower type fans, as their air deliveries are higher and their locations may be remote from the space to be refrigerated. Air velocities and volumes must be designed for the individual application. Due to the small temperature difference between coil and air, the air volumes handled are many times greater than in comfort air conditioning work. Where a defrosting cycle is not practicable, this type of unit may employ a pump to spray a eutectic solution over the coils for the purpose of avoiding frosting.

Ratings. In order to rate and test equipment of this kind which normally operates below the frost temperature, a proposed code, Standard Methods of Rating and Testing Forced-Circulation and Natural Convection Air Coolers for Refrigeration[3], has been issued. In this standard, the gross cooling effects are taken since the motor power input equivalent is to be computed as part of the load.

REFERENCES

[1] Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association* (*A.S.R.E. Circular* No. 13-42).

[2] Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association* (*A.S.R.E. Circular* No. 16).

[3] Proposed A.S.R.E. Standard Methods of Rating and Testing Forced-Circulation and Natural Convection Air Coolers for Refrigeration (*A.S.R.E. Circular* No. 25-44).

CHAPTER 37

SPRAY APPARATUS

Air Washers, Humidification with Air Washer, Dehumidification and Cooling with Air Washers, Well and Water Main Temperatures, Apparatus for Direct Humidification, Water-Cooling Towers, Water Use and Conservation, Rivers and Lakes, Spray Cooling Ponds, Atmospheric and Mechanical Draft Cooling Towers, Mechanics of Atmospheric Water-Cooling, Design Conditions, Water-Cooling Tower Design, Selection of Water-Cooling Towers, Operation and Maintenance

AIR humidification is effected by the vaporization of water and always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 3. The removal of moisture from air may or may not involve the removal of heat from the air-vapor mixture. With spray equipment, dehumidification of air always necessitates the removal of heat.

AIR WASHERS

An air washer consists essentially of a chamber or casing in which is provided a spray nozzle system, a tank at the bottom of the chamber for collecting the spray water as it falls, and an eliminator section at the leaving end of the chamber for removal of drops of entrained moisture from the delivered air. Air is drawn through the casing of the washer, where it comes into intimate contact with the spray water. A heat transfer takes place between the air and water, resulting in either humidification or dehumidification of the air, depending upon the method of operation and the relative temperatures of air and spray water.

To prevent backlash of spray ahead of the washer chamber and to aid in more uniform air distribution, inlet diffusion plates or eliminator baffles, where necessary, are provided in the air entrance end of the air washer. Inlet diffusion plates are used when the air flow and water spray are in the same direction; eliminator baffles of special design are used where one or more of the water sprays opposes the air flow. At the outlet end of the washer suitable flooded eliminator plates are used. These plates, for the removal of entrained moisture, usually cause four to six changes in direction of the air flow.

Figs. 1 and 2 show the essential construction features of conventional air washers. Intimate contact between the air and the water is secured; (1) by breaking the water into fine drops, (2) by passing the air over surfaces continuously wetted by water, or (3) by a combination of the two.

The wetted surfaces in an air washer may be of fibre glass, metal or scrubber plate construction. Scrubber plate types of washers are generally used to wash reclaimable products from the air and are composed of several baffle type plates located across the air stream. Water is supplied at the top of the washer to spray over these plates. In the case of the fiber glass or metal surfaces the water spray is usually rather coarse and at

low pressure. In many cases these sprays are set at an angle with the air flow. Air washers of this type not only perform necessary heat transfer functions, but also are effective removers of dust and dirt from the air stream.

Essential requirements in the air washer operation are: uniform distribution of the air across the chamber section, moderate air velocity of from 250 to 600 fpm in the washer chamber, an adequate amount of spray water broken up into fine droplets throughout the air stream, at pressures of from 15 to 30 psig, sufficient length of travel through the

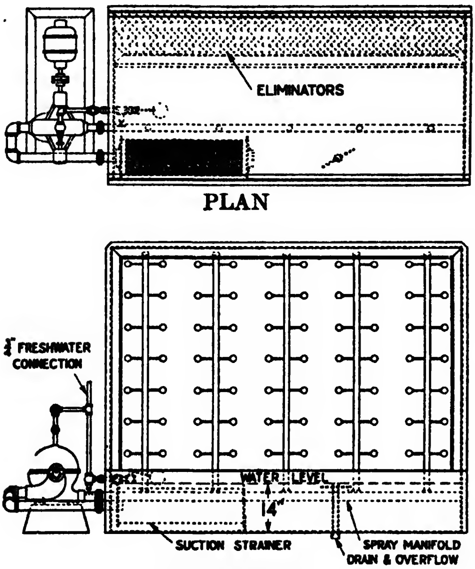


FIG. 1. TYPICAL SINGLE-BANK AIR WASHER

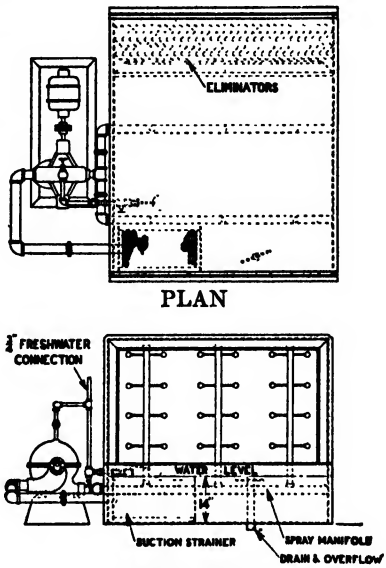


FIG. 2. TYPICAL TWO-BANK AIR WASHER

water spray and wetted surfaces, and the elimination of entrained moisture from the outlet air.

Expected performances, physical size, length, number of sprays, etc., vary greatly, depending upon the functions of the installation. In general, the width and height of an air washer are dictated by the space available. Washers of nearly equal height and width are desirable from an air flow and economic standpoint although not necessary. The length of washers varies considerably. A space of approximately 2½ ft between spray banks is used and the first and last banks of sprays are located about 1 ft to 1½ ft from the entering or leaving end of the washer. In addition air washers are very often furnished with cooling coils or heating coils within the washer chamber and the use of these coils affects the overall length of the washer.

Where increase of overall heat transfer between the air and water is required, multistage washers are used. These washers are equivalent to a number of washers in series and the water is often pumped from one stage to the other where conditions permit.

The resistance to air flow through an air washer varies with the type of eliminator and wetted surfaces, number of banks of spray and their

direction, air velocity, size and type of other resistances such as cooling and heating coils, and other factors such as air density. Resistances vary from as low as ¼ in. to higher than 1 in. water column and it is therefore necessary that the manufacturer be consulted in regard to the resistance of any particular washer design involved.

HUMIDIFICATION WITH AIR WASHER

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air, (2) preheating the air and washing it with recirculated spray water, and (3) using heated spray water. In any air washing installation the air should not enter the washer with a dry-bulb temperature less than 35 F in order to eliminate danger of freezing the spray water.

Method 1. Except for the small amount of energy added from outside by the recirculating pump in the form of shaft work, and for the small amount of heat leak from outside into the apparatus, including the pump and its connecting piping, the process would be strictly adiabatic. Evaporation from the liquid spray would therefore be expected to bring the air immediately in contact with it to saturation adiabatically; and, since the liquid is recirculated, its temperature would be expected to adjust to the thermodynamic wet-bulb temperature of the entering air.

It does not follow from the foregoing reasoning that the whole air stream is brought to complete saturation, but merely that its state point should move along a line of constant thermodynamic wet-bulb temperature as explained in Chapter 3. The extent to which the final temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached is conveniently expressed by a ratio known as *humidifying effectiveness* or *saturating effectiveness* and is defined:

$$e_h = \frac{t_1 - t_2}{t_1 - t'} \tag{1}$$

where

e_h = humidifying effectiveness, per cent.
t_1 = dry-bulb temperature of the entering air, Fahrenheit degrees.
t_2 = dry-bulb temperature of the leaving air, Fahrenheit degrees.
t' = thermodynamic wet-bulb temperature of the entering air, Fahrenheit degrees.

The following may be taken as representative humidifying or saturating effectiveness of an air washer for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The humidifying or saturating effectiveness of a washer is dependent upon the essential items of design mentioned under Air Washers. Other conditions being the same, low velocity of air flow is more conducive to higher humidification effectiveness.

Method 2. The preheating of the air increases both the dry- and wet-bulb temperatures, lowers the relative humidity, but does not alter the humidity ratio (pound water vapor per pound dry air). At a higher wet-

Table 1. Average Maximum Water Main Temperatures[a]

State	City	Temp. F	State	City	Temp. F	State	City	Temp. F
Ala.	Birmingham	84	Mass.	Boston	80		Tulsa	86
	Mobile	80		Cambridge	70	Ore.	Eugene	60
Ark.	Little Rock	83		Fall River	76		Portland	65
Ariz.	Phoenix	82		Lowell	50	Pa.	Altoona	74
	Tucson	80		Lynn	68		Erie	75
Calif.	Anaheim	60		New Bedford	70		Johnstown	74
	Berkeley	69		Salem	68		McKeesport	82
	Fresno	80		Worcester	76		Philadelphia	85
	Fullerton	75	Mich.	Detroit	77		Pittsburgh	86
	Glendale	68		Flint	70	R. I.	Providence	68
	Los Angeles	80		Grand Rapids	84	S. C.	Charleston	83
	Oakland	69		Highland Park	77		Greenville	81
	Ontario	70		Jackson	56		Spartanburg	78
	Pasadena	82		Kalamazoo	53	S. D.	Rapid City	65
	Pomona	75		Lansing	64	Tenn.	Chattanooga	84
	Riverside	78		Saginaw	82		Knoxville	90
	Sacramento	72	Minn.	Duluth	55		Memphis	85
	San Bernardino	65		Minneapolis	85		Nashville	90
	San Diego	84		St. Paul	80	Texas	Amarillo	70
	San Francisco	71	Mo.	Jefferson City	82		Austin	90
	Whittier	75		Kansas City	84		Beaumont	86
Colo.	Denver	75		Springfield	82		Dallas	86
Conn.	Bridgeport	66		St. Joseph	84		El Paso	85
	Hartford	73		St. Louis	85		Fort Worth	85
	New Haven	76		Springfield	74		Galveston	90
	Waterbury	72	Nebr.	Lincoln	70		Houston	83
D.C.	Washington	84		Omaha	85		Port Arthur	83
Del.	Wilmington	83	Nev.	Reno	70		San Antonio	78
Fla.	Jacksonville	86	N.H.	Manchester	76		Wichita Falls	85
	Miami	82	N. J.	Jersey City	63	Utah	Logan	44
	Tampa	77		Newark	75		Salt Lake City	65
Ga.	Atlanta	85		Paterson	78	Va.	Fredericksburg	75
	Macon	80		Trenton	79		Lynchburg	73
Idaho	Boise	60	N. Y.	Albany	68		Norfolk	80
Ill.	Chicago	78		Buffalo	78		Richmond	85
	Cicero	76		Jamaica	56	Wash.	Olympia	58
	Evanston	73		Mt. Vernon	74		Seattle	62
	Moline	83		New Rochelle	75		Spokane	51
	Peoria	67		New York	72		Tacoma	57
	Rockford	59		Rochester	70	W.Va.	Charleston	85
	Springfield	82		Schenectady	60		Huntington	78
Ind.	Evansville	88		Syracuse	74		Wheeling	78
	Gary	75		Utica	69	Wis.	LaCrosse	54
	Indianapolis	84		Yonkers	70		Madison	58
	South Bend	61	N. C.	Asheville	74		Milwaukee	75
	Terre Haute	82		Charlotte	85		Racine	68
Iowa	Cedar Rapids	78		Raleigh	92			
	Des Moines	77		Winston-Salem	82			
	Sioux City	62	N. M.	Albuquerque	65	Province		
Kans.	Concordia	57	Ohio	Akron	76			
	Kansas City	86		Canton	50			
	Topeka	88		Cincinnati	84			
	Wichita	72		Cleveland	77	Alta.	Calgary	64
Ky.	Louisville	85		Columbus	84	B. C.	Vancouver	60
La.	Baton Rouge	85		Dayton	60	Ont.	London	50
	New Orleans	90		Lakewood	82		Toronto	63
	Shreveport	88		Springfield	72	P.E.I.	Charlottetown	48
Me.	Augusta	60		Toledo	83	Que.	Montreal	78
Md.	Baltimore	75	Okla.	Oklahoma City	82		Quebec	68

[a] These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown. Some values were supplied by H. E. Degler, Marley Company. Some were obtained from City Water Department records. The highest values given by the various authorities are usually those listed.

bulb temperature but the same humidity ratio, more water can be absorbed per pound of dry air in passing through the washer, assuming that the humidifying effectiveness of the washer is not adversely affected by operation at the higher wet-bulb temperature. The analysis of the process occurring in the washer itself is the same as that explained under *Method 1*. The final desired conditions are secured by adjusting the amount of preheating to give the required wet-bulb temperature at entrance to the washer.

Method 3. Even if heat is added to the spray water, the mixing occurring in the washer itself may still be regarded as adiabatic. The state point of the mixture should move in a direction determined by the specific enthalpy of the heated spray as explained in Chapter 3. It is possible, by elevating the water temperature, to raise the air temperature, both dry-bulb and wet-bulb, above the dry-bulb temperature of the entering air.

In each of the methods, 1, 2 or 3, the air leaving the air washer may require reheating to produce in the conditioned space the required dry-bulb temperature and relative humidity.

DEHUMIDIFICATION AND COOLING WITH AIR WASHERS

Cooling of the wet-bulb temperature of an air vapor mixture can be accomplished by an air washer if the temperature of the spray water is lower than the wet-bulb temperature of the air. Moisture removal is obtained when the spray water temperature is lower than the dew-point of the entering air. In these cases the final dry-bulb temperature and relative humidity of the leaving air are dependent upon the design factors of the air washer.

Both sensible and latent heat are removed in the process of dehumidification by cold spray water. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat removal takes place as condensation occurs. Therefore, the lower the spray temperature the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same.

Washers with two or more banks of spray are usually selected for dehumidifying installations, whether for comfort or industrial installations. Generally such air washers cool the air to within one or two degrees (Fahrenheit) of the leaving spray water temperature; this differential will increase somewhat when the difference between the entering wet-bulb and leaving dew-point is relatively large.

Where a limited supply of cold water is available multiple stage washers may be used to great advantage. In such washers the cool water is pumped through the multiple spray systems in series and counterflow to the air flow. Such an arrangement brings the delivery air in contact with the coldest water, securing a maximum amount of cooling with economy of water.

When using cold well water or water from city water mains care should be used to secure accurate data on the water temperatures. Table 1 lists some approximate water main averages which may be used as a guide but they should be verified from local records. This is particularly true with city water main temperatures. In the case of well water temperatures Fig. 3 shows the approximate temperatures of water to be expected from wells at depths of 30 to 60 ft.

Air washers for dehumidifying and cooling usually have separate recirculating pumps. These pumps deliver a mixture of cold and recirculated water under the control of a three-way valve. The valve may be actuated either by a thermostat in the washer outlet or by a humidity or other controller in the space being conditioned.

Air washers for dehumidifying are very often furnished with direct expansion or water cooling coils within the washer space, in which case water for the washer sprays is entirely recirculated.

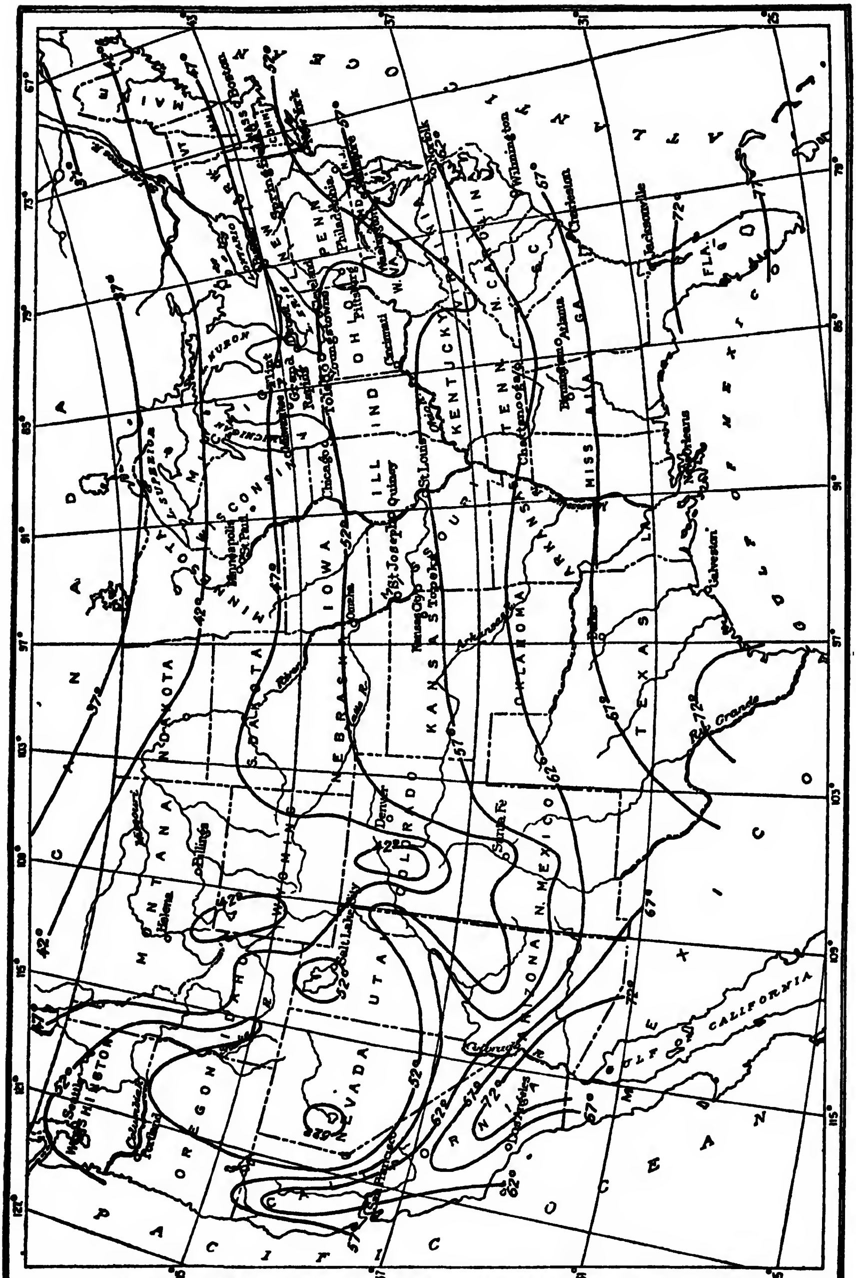


FIG. 3. APPROXIMATE WELL WATER TEMPERATURES AT DEPTHS OF 30 TO 60 FT[1]

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into the following general types, according to the method of operation: (1) indirect, such as the air washer, which introduces moistened air; and (2) direct, which sprays moisture into the room or introduces moisture by means of steam jets.

As in the cases of humidification by use of an air washer, the heat necessary for the vaporization of the moisture added to the air by direct humidification is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the enthalpy of the air remains constant but the dry-bulb temperature of the air is reduced.

Direct humidification is usually preferable where high relative humidities must be maintained, but where there is little cooling or ventilation required. In comfort air conditioning, where both humidification and ventilation are required, the indirect humidifier is preferable. In industrial applications, where the cooling or ventilation load is large and where very high relative humidities must be maintained, a combined system employing both direct and indirect humidifiers is sometimes used.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers which employ nozzles placed within the room and rely upon compressed air to effect complete atomization of the water so that it can be converted to vapor by the heat of the room air. Some of these nozzles depend upon an aspirating effect to draw the water into the nozzle and atomize it, others operate on a combined air and water pressure. It is usual for nozzles of the water pressure

type to be controlled by a diaphragm valve actuated by the pressure of the atomizing air.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor diffused. This distribution of fine spray over the maximum possible area promotes complete and rapid vaporization.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

WATER-COOLING TOWERS

The removal and dissipation of heat from a compressed refrigerant or from exhaust steam are important factors in the efficient operation of a refrigerating plant or an electric steam-generating station. This heat removal is generally accomplished by first transferring the heat of the *gas* to cooling water in a heat exchanger. The water, if cheap or plentiful, may be wasted to the nearest sewer or open waterway such as a river or lake. Where water usage is restricted or expensive or where the available water contains dissolved salts which would form scale on the heat-exchange apparatus, it is necessary to recirculate the water, and to cool it, after each passage through the heat-exchanger, by contact with moving air in some type of water-cooling apparatus.

Water Use and Conservation

Many communities have found that present water systems are not sufficiently large to satisfy the increasing demands of domestic and industrial users. The reasons for such shortages are primarily: (a) inadequate purification and water-distribution systems, (b) inadequate sanitary and storm-sewer disposal systems, or (c) inadequate sources of water.

Even when an adequate supply of water is available from the water

mains or private wells, many cities do not have sufficient sanitary or storm sewer facilities to handle increasing demands. The sanitary systems are usually limited because of the capacity of the filtration plants, and therefore many cities restrict the use of the sanitary system to sewage.

Rivers and Lakes

Until the year 1920, large generating stations were usually located on the banks of rivers, lakes, or artificial ponds. The removal and dissipation of the heat from the Diesel cylinder or the exhaust steam of a turbine was accomplished by taking in the circulating water at a considerable distance from the discharge, thus preventing mixing of the heated discharge with the inlet water. The use of water from streams for this purpose has the following disadvantages: the site may be far removed from the fuel source or from power consumers; water supply may limit plant expansion; municipal restrictions on use of water may hamper operation; costly intake structures with screens and sediment basins may be required; drastic flood or drouth conditions, the vagaries of most rivers, upstream pollution, scale-forming constituents, debris, sand, algae, and formation of troublesome ice may cause operating difficulties.

When lakes and cooling ponds have been used as a source of circulating water, the hot water is discharged close to the surface at the shore line. Natural air movement over the surface of the water causes evaporation over that area, thus carrying the heat away at a rate of about 4 Btu per (hr) (sq ft) (F deg. temp. difference between air and water). Increased density of the water due to loss of heat causes the cooled water to sink to the bottom of the pond. The suction connection is therefore located as far below the surface as possible and at as great a distance from the discharge as practicable. The area required by such cooling ponds is about 50 times that of a spray pond or about 1000 times that of a water-cooling tower to dissipate the same quantity of heat and achieve equal operating costs. If the surfaces of such ponds were below the level of surrounding terrain and the shore were wind-sheltered by trees or other vegetation, so that natural air movement across the surface of the water would be retarded, the use of a spray pond or water-cooling tower would be indicated.

SPRAY COOLING PONDS

The spray pond consists of a water collecting basin, above which spray nozzles are located in an arrangement such as shown in Fig. 4 to spray the water upwards into the air. Properly designed spray nozzles break the water into small drops, but not into a mist. Since the objective is to cool the pond water, the individual drops must be heavy enough to fall back into the pond and must not float away in the air. The water surface exposed to the air passing over the pond becomes the integrated area of all the small drops. The spray pond requires about one-fiftieth of the space required by the cooling pond to dissipate the same quantity of heat with equal results, due to four factors: (1) the speed with which the drops are propelled into the air and fall back into the water basin; (2) the increased wind velocity at a point above the surrounding obstruction; (3) the increased volume of air delivery due to the greater vertical cross section of air permissible; and (4) the vastly increased area of contact between water and air.[2]

Spray pond effectiveness is increased by: (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity

nozzles to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area.

It is usual practice to locate the nozzles from 5 to 12 feet above the surface of the water (dependent also upon depth of water and curb level) with water supply at 5 to 7 psig pressure at the nozzles. Nozzles spray from 25 to 60 gpm each and the nozzles are spaced so that the average water delivered to the surface varies from 0.1 gpm (small ponds) to 0.4 gpm (large ponds) per square foot. See Table 2 for additional spray pond design data. Best results are obtained by placing the nozzles in a long, relatively narrow area, located broadside to the wind.

Louver fences to prevent the carrying of entrained water beyond the edge of a spray pond by the air on the leeward side are required for all roof locations and for ground locations where space is restricted; the outer nozzles should be located at least 20 ft from the edge of the basin. Such fences up to 12 ft in height usually are constructed of horizontal overlapping

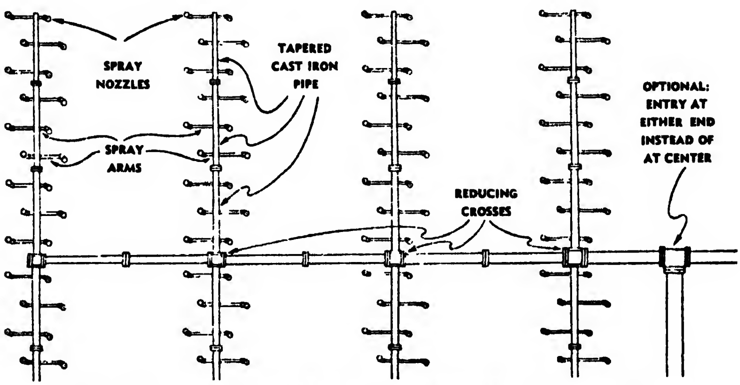


Fig. 4. Typical Nozzle Arrangement for a Spray Pond

louvers supported between vertical posts. The air, in passing between these louvers, tends to be freed of the larger drops of water. The louvers also restrict the flow of air, particularly at the higher wind velocities, thus reducing the possibility of water being carried from the spray cloud. The height of an effective fence should be equal to the height of the spray cloud. Algae formations may be a nuisance in a spray pond. Such growths are minimized by the periodic addition of bromine, chlorine, chlorinated lime, copper sulfate, or various blends of chlorophenates (see Chapter 51).

The performance of a spray pond is limited because of space requirements and the probable high cost of piping and pumping. Water-cooling towers, however, allow the designer a wider range of performance within a given space because of the possibility of altering the smaller physical dimensions or varying the water concentration, measured in gallons per (minute) (square foot of tower area). In most cooling towers the water is broken up into drops many times whereas with the spray pond it is broken up only once and consequently in the latter the rate of cooling diminishes rapidly as the temperature of the surface of the drop approaches the wet-bulb temperature of the ambient air.

ATMOSPHERIC COOLING TOWERS

Spray-filled atmospheric cooling towers are used for open-area installations because of their dependence upon the velocity and direction of the wind. Operation is not so limited as with spray ponds, but the design is generally based on a 3 mph wind and the performance falls off rapidly as the ambient air velocity decreases. These towers require less basin area, less piping, and no more mechanical equipment than spray ponds, but these savings may be largely offset by the extra cost of the structure. The drift nuisance is similar to that of spray ponds. The word *tower* used in this connection is a misnomer, as the design simulates a narrow spray pond with length twice the width, or more, having elevated nozzles and a high louver fence. As usually built, the nozzles spray downward from the top of the structure, and the distance from the center of the nozzle system to the louvers on either side is not more than half the distance that the nozzles are elevated

Table 2. Spray Pond Design Data

Conventional Up-Spray System

	Units	Standard	Minimum	Maximum
Water Capacity per nozzle	gpm	35 to 50	25	60
Nozzles per 12 ft length of pipe		6	4	6
Height of nozzles above water level	ft	6	5	12
Nozzle pressure	psig	6	5	7
Size of nozzles and nozzle arms	in.	2	1½	2
Distance between spray lateral piping	ft	25	13	38
Distance nozzles from pond side unfenced	ft	25 to 35	20	50
Distance nozzles from pond side fenced	ft	15 to 20	15	25
Height of louver fence	ft	12	12	12
Depth pond basin	ft	4 to 5	2	—
Friction loss allowed per 100 ft pipe	ft	1 to 3	—	—
Design wind velocity	mph	5	3	—

above the water-collecting basin. Heights range from 6 to 15 ft, with the total width of the structure usually not greater than the height. Loadings range from 0.6 to 1.5 gpm per sq ft of tower area, and hence require about one-fourth the area of an equivalent spray pond. As the louvers are wetted continuously they add to the surface of water exposed to the cooling air. The spray-filled atmospheric tower is shown in Fig. 5.

Much of the atmospheric water cooling for refrigeration work during the past 30 years has been done with natural-draft deck type towers, also referred to as *atmospheric deck towers*, see Fig. 6. These towers consist of a sturdy wooden or steel frame 20 to 50 ft high and 8 to 16 ft wide, carrying open horizontal wooden latticework or *decks* at regular intervals from top to bottom. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and drops from deck to deck enroute to the basin. The purpose of the decks is primarily to arrest the fall of the water, to break and re-break it into drops so as to present the most efficient cooling surface to the air, which is passing through the tower transversely to the decks. The wooden decks also add to the area of water surface exposed to the air, but since they offer resistance to the flow of air, the number and arrangement of the decks depend upon basic tests and operating experience.

To prevent loss of water on the leeward side of the tower, wide louvers

(drift eliminators) are attached at regular intervals from top to bottom, these louvers extend outward and upward at an angle of 45 to 50 deg. In most designs the top edge of each louver extends above the bottom edge of the one above. These louvers serve the same function as a louver fence around a spray pond, namely to stop the water drops carried by the air beyond the open area of the tower, and to control the quantity of air permitted to pass through the tower.

The efficiency of a deck tower is improved primarily by increasing length or height, or both, within limits; the length and height increase the area of tower exposed to the wind. The improvement is not directly proportional to the change made in either case. Neither does a certain percentage

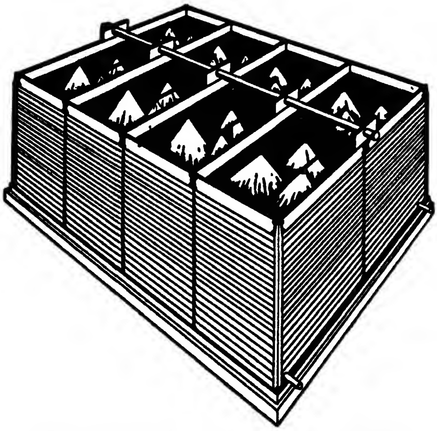


Fig. 5. Spray Filled Atmospheric Cooling Tower

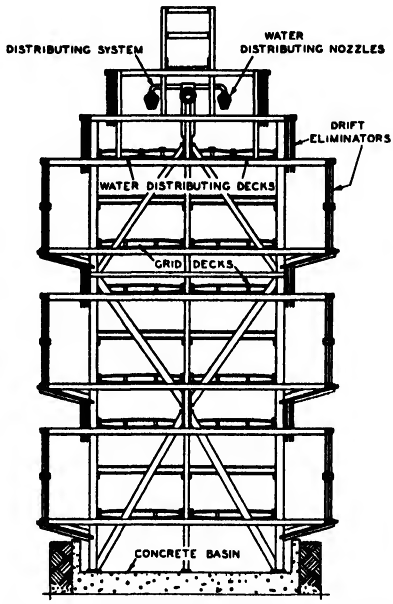


Fig. 6. Atmospheric Deck Tower

change of one dimension make an equal improvement in efficiency on two equal towers of different original lengths or heights. Since the range of efficiency varies through wide limits, it is impracticable to attempt to list data here on the area required per unit quantity of water. Improved efficiency due to added height is obtained at the expense of additional pumping head and increased weight per unit of area, whereas improvement gained by greater length or width will increase the area and consequently the foundation required.

Drift loss in a properly designed deck tower is considerably less than in the spray pond, but the drift nuisance may be considerable and for this reason atmospheric deck towers are unsuitable for downtown building roofs, locations adjacent to buildings, or near expensive mechanical equipment in industrial plants. They must be located in an open area, broadside to the prevailing wind. They are inefficient with less than 3 mph wind velocity and with wind directions other than broadside. These

towers are long and high in proportion to width and must be securely anchored to prevent uplift or overturning during high winds. High pumping requirements (30 to 60 ft) and total dependence upon atmospheric caprice, especially wind (quantity and direction), are disadvantages.

Due to new uses and growth of demand in recent years requirements for water-cooling equipment have become increasingly varied and exacting, necessitating refinements and specialized adaptations. The principal demand for large water-cooling systems in recent years has come from the petroleum industry and steam power plants. Refrigeration, air conditioning, and engine-jacket cooling service today employ a large percentage of the medium sized and small water-cooling towers installed.

MECHANICAL DRAFT TOWERS

The mechanical draft tower consists usually of a vertical shell constructed of wood, metal, transite, or masonry. Water is distributed near the top,

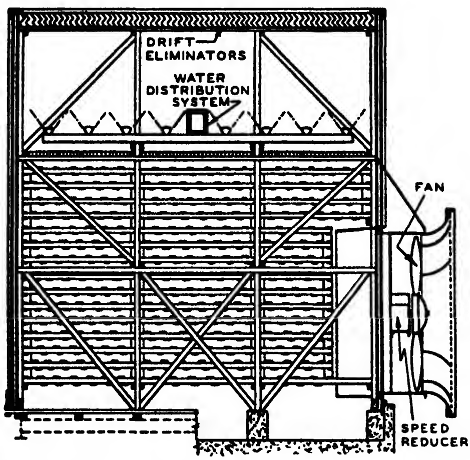


FIG. 7. FORCED DRAFT COOLING TOWER

uniformly over the area, and falls to the collecting basin in the bottom, passing through air which is being circulated in the tower from bottom to top by forced or induced draft fans, or which is circulated horizontally in crossflow towers by induced draft fans.

In vertical towers the air passes counterflow to the water and is in contact with the hottest water just before leaving the tower; hence a given quantity of air picks up more heat than the average equal quantity of air on natural draft equipment. This permits the water to be cooled with the least quantity of air required by any type of cooling equipment. As movement of air through the towers is obtained by power-consuming fans it is essential that this air quantity and the draft loss be reduced to a minimum so as to secure low operating cost.

The inside of a mechanical draft tower may be spray filled, i.e., the water surface is presented to the air by filling the entire inside of the structure with water droplets from the spray nozzles, or it may be packed with wood

filling over which the water cascades from top to bottom. In many cases, a combination of the spray-filled and wood-filled design is used.

The *forced draft* type of *tower* (Fig. 7), has the advantages of being suitable for corrosive waters and having the fan mounted near the ground level on a rigid foundation where it is easily accessible.

The heated air leaves the top of a forced draft tower at a low velocity and may be subject to *recirculation* to the fan inlet, with consequent reduction in performance. This reduction could be as much as 20 per cent under certain conditions. During cold weather, recirculation may cause ice formation on adjacent equipment and buildings as well as in the tower

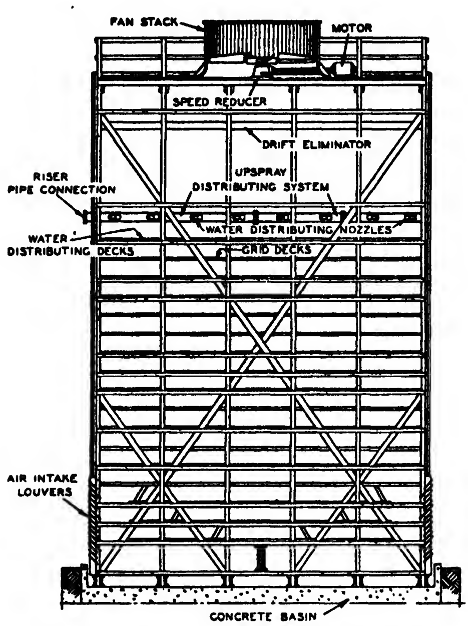


Fig. 8. Counterflow Induced Draft Cooling Tower

fan ring with possible resultant fan breakage. Fan sizes are limited to 12 ft or less, and therefore more fans, motors, starters, and wiring are needed than for induced draft towers. Induced draft towers, since fans and motors are not visible, are therefore somewhat more adaptable to architectural treatment.

In the *spray-filled mechanical draft tower*, the area presented to the air is the combined surface area of the small drops present in the tower at any one time. The net free cross-sectional area of the air spaces in a spray-filled tower is greater than that of the wood-filled tower for the same plan area. Before discharging to the atmosphere, the water-laden exhaust air passes through a drift eliminator to remove entrained moisture. This type of tower is particularly applicable for installations in restricted areas where city ordinances require fire-proof construction.

In the wood-filled tower, lumber of various cross sections is laid horizontally across the space on as close centers, horizontally and vertically,

as required, without introducing too great a resistance to air flow. The water is distributed over the top layer by means of spray nozzles, troughs, splash heads, or through evenly spaced nozzles located in the floor of an overhead open-type water distribution basin, and drops from piece to piece of the wood filling as it progresses downward. As the air moves upward or across the wood filling, the latter presents a large wetted surface, repeatedly breaks up the falling drops of water, and continuously provides new drop surfaces whose integrated areas are several times that of the wood-fill area.

The efficiency of a mechanical draft tower is improved by increasing the amount of filling, height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water, without affecting seriously the fan power required, but increases the pumping power. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocity increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but since a greater quantity of air is passing through, the average differential between the water temperature and wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air handled by fans of the disc type.

The performance of mechanical draft towers is independent of wind velocity, hence it is possible to design them for more exacting performance. They require less space and less piping than atmospheric deck towers, and the pumping head varies from 11 to 26 feet, depending upon the design. Overall plant economy due to colder water temperature usually more than offsets the additional operating expense and initial cost as compared with those of atmospheric towers.

The *counterflow* (conventional) type of *induced draft tower* has the fan located at the top, Fig. 8, to provide vertical air movement across the filling. Air is discharged upward at a high velocity to prevent recirculation. Another type, for small requirements, has the induced draft fan in one end (see Fig. 9) to provide horizontal flow.

Another induced draft tower, developed for the purpose of obtaining compactness, larger capacity, increased flexibility and improved performance, is the *cross flow* type. This type of tower employs multiple fans centered along the top, each fan drawing air through two cells paired to the suction chamber which is partitioned midway beneath the fans and fitted with *drift eliminators* that turn the air upward toward the fan outlet. This tower obtains a horizontal air movement as water falls in a cascade of small drops over the filling and across the air stream with less resistance to air flow. The air travel is longer than with the conventional design.

Air velocities through mechanical draft towers vary from 250 to 400 fpm over the gross area of the structure. The air requirements are approximately 300 to 400 cfm of air per ton of mechanical refrigeration and about 100 to 150 cfm of air per gallon of water passing through the tower. Cooling tower calculations are based upon the fact that mechanical refrigeration requires approximately 30 gallon-deg of cooling water per minute per ton of refrigeration. In atmospheric cooling towers, if 5 gpm were circulated, the water-cooling range would be 6 F; with mechanical draft towers, 3 or 4 gpm are usually circulated for a desired water-cooling range of 10 or 7½ F. Some designs of mechanical draft towers are limited to 6

or 7 gpm per sq ft because of blanketing effect while the capacity of the most efficient types range up to 9 or 10 gpm.

When an inside cooling tower is required some adaptation of a spray filled or wood filled induced draft tower is often used and occasionally an air washer is converted to this service. In this type of application precautions must be taken to prevent the discharged air from short circuiting to the intake.

MECHANICS OF ATMOSPHERIC WATER-COOLING

The heat exchange in atmospheric water-cooling equipment is accomplished partially by a transfer of sensible heat which raises the wet-bulb temperature of the moving air, but most of the cooling is due to an exchange of latent heat resulting from the evaporation of a small part of the water.

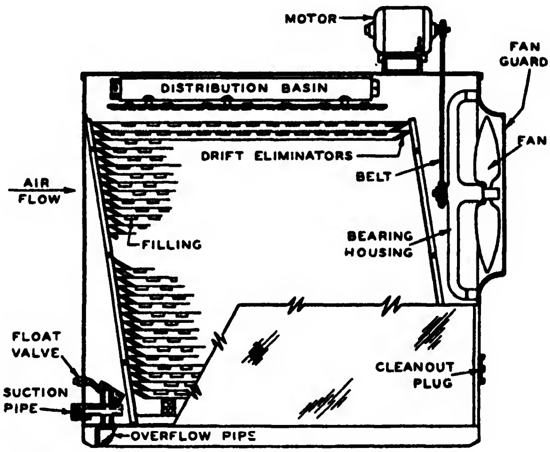


FIG. 9. SMALL HORIZONTAL INDUCED DRAFT COOLING TOWER FOR 3 TO 50-TON REFRIGERATING UNITS

If all of the water were cooled by evaporation, the rate of evaporation would be approximately one per cent for each 10 deg of cooling. In practice, the loss of circulating water by evaporation will approximate 1 per cent for 12 to 14 deg of actual cooling due to the additional amount of cooling by sensible heat transfer, and the rate of evaporation will vary from about 0.64 per cent of the water circulated in the winter to 0.88 per cent in the summer for a water-cooling range of 10 deg.

The lowest temperature to which water may be cooled in atmospheric cooling equipment is to the temperature of adiabatic saturation, which is at the wet-bulb temperature of the air. Performance is measured in terms of *approach* (5 to 10 F deg, with 7 F deg average) of the cooled water to the wet-bulb temperature of the ambient air when cooling the water through some desired range. The water-*cooling range* in some installations will vary from 10 to 12 F deg when a spray pond is used, and from 5 to 17 F deg (with 10 F deg average) for a mechanical draft cooling tower.

Heat absorption by the moving air in an atmospheric water-cooling tower continues as long as the wet-bulb temperature of the air is lower than the temperature of the water. The rate of heat transfer depends upon: (1) the area of water in contact with the air, (2) the relative velocity of

the air and water during contact, (3) the difference between the wet-bulb temperature of the air and the initial temperature of the water, and (4) the time of contact of the air with the water. The rate of heat dissipation is also influenced by many other lesser factors[3] which further complicate the cooling tower design. Ultimate selection of water-cooling equipment for any specified service depends on over-all economic considerations established from correlated performance data. As the enthalpy of the moving air increases, its wet-bulb temperature rises (see Chapter 3). Since it is impracticable to allow the air to be in contact with the water for a long enough time to permit the wet-bulb temperature of the moving air and the temperature of the water to reach equilibrium, atmospheric water-cooling equipment aims to circulate only enough air to cool the water to the desired temperature with least expenditure of power.

DESIGN CONDITIONS FOR WATER-COOLING

The maximum wet-bulb (design) temperature at which the total quantity of circulating water must be cooled through a specified range by water-

TABLE 3. EFFECTIVENESS OF WATER COOLING EQUIPMENT

Cooling Equipment	Water Cooling Effectiveness—Per Cent		
	Minimum	Typical	Maximum
Spray Ponds....	30	40 to 50	60
Spray Filled Atmospheric Towers	40	45 to 55	60
Atmospheric Deck Towers........	50	50 to 60	90
Mechanical Draft Towers......	50	55 to 75	93

cooling equipment is never selected as the highest wet-bulb temperature ever known to have occurred for some locality nor the average wet-bulb temperature over any period of time. The *maximum* basis would require cooling equipment several times larger than normal capacity, and the *average* basis would result, for a large part of the time, in higher condenser temperatures than those for which the plant was designed.

Accepted design practice for water-cooling towers, evaporative condensers, and spray ponds, is to use the maximum hourly outdoor dry-bulb temperature which will be exceeded no more than 2½ per cent of the time for the months of June to September, also to use the maximum hourly wet-bulb temperature which will be exceeded no more than 5 per cent of the total hours for the same period. Tabulation of these data has not been completed. The limited portion of such data as are available is given in Table 3, Chapter 15 for airport weather stations; for other localities design dry-bulb and wet-bulb temperatures in use locally are tabulated as a guide to design temperatures. More complete summer weather data, statistics, charts, maps, and technical analysis have been prepared by Albright.[4]

Equipment for steam turbine condensers and internal combustion engines is usually based upon somewhat lower design temperatures if peak loads occur at night or during winter months when outdoor temperatures are lower.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the effectiveness range of the

type of atmospheric water cooling apparatus to be used. This effectiveness is expressed as the percentage ratio of the actual cooling effect to the maximum possible cooling effect. Since the wet-bulb temperature of the entering air is the equilibrium temperature to which the water could be cooled, the effectiveness of water cooling apparatus can be indicated thus:

$$E_1 = \frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}} \quad (2)$$

where

E_1 = water cooling effectiveness, per cent.

Magnitudes of this effectiveness ratio will vary through wide limits in accordance with construction and conditions of operation. Values indicative of the commercial range of the effectiveness ratio are given in Table 3, although unusual designs may operate outside these ranges.

Example 1. A mechanical refrigeration installation of 100-ton capacity requirés 3 gpm of cooling water with the hot-water temperature at 95 F and the cold-water at 84 F with a design wet-bulb temperature of 78 F. Find the water-cooling effectiveness of a mechanical draft tower for the above conditions.

Solution. Substituting known conditions in Equation 2,

$$\text{Water-cooling effectiveness} \quad \frac{95 - 84}{} \times 100 = 64.7 \text{ per cent (typical)}$$

From a consideration of the factors which include the water-cooling range and the design wet-bulb temperature of the ambient air, the quantity of water required can be calculated from the amount of heat to be rejected. The average quantities of heat to be removed from various types of mechanical equipment that require cooling are listed in Table 4.

WATER-COOLING TOWER DESIGN

Because of the many variables[3] in water-cooling tower calculations and performance, it is difficult to provide simple handbook equations and tables whereby an engineer can readily select the type and size of unit for a definite requirement. Each manufacturer has a semi-confidential method of sizing a tower, based largely upon research and actual performance correlated with definite requirements; selection of water-cooling equipment

TABLE 4. HEAT ABSORBED BY COOLING WATER

MECHANICAL EQUIPMENT	BTU PER MIN PER TON	BTU PER LB OF STEAM	BTU PER BHP-HR
Refrigeration Compressor	250	—	—
Refrigeration, Absorption System	550	—	—
Steam Turbine Condenser	—	1000	—
Steam Jet Refrigerating Condenser	550	1100	—
Diesel Engine Jacket & Lube Oil:			
Four-cycle, Supercharged	—	—	2600
Four-cycle, Non-supercharged	—	—	3000
Two-cycle, Crank-case Compressor	—	—	2000
Two-cycle, Pump Scavenging, Large Unit	—	—	2500
Two-cycle, Pump Scavenging, High Speed	—	—	2200
Natural Gas Engine:			
Four-cycle	—	—	4500
Two-cycle	—	—	4000

for any specified service must ultimately depend upon overall considerations established from reliable design and performance data.

Some of the variables encountered in water-cooling tower work are: continuously changing air and water temperatures throughout the structure; varying moisture content, pressure, and volume of the moving air; caprice of the weather, ambient air changes in temperature, humidity, wind velocity and direction, and the amount of sunshine. Other less important physical properties of the air and water affecting tower performance are: density, specific heat, conductivity, viscosity, vapor pressure, surface tension, latent heat, coefficient of expansion, vapor diffusivity, emissivity, and molecular weight. The air velocity, overall and in different parts of the tower, is an important heat transfer factor; also the type of air movement provided, whether natural draft, forced draft, induced draft, counterflow, or crossflow design. Different details of construction will produce dissimilar velocities of water, its distribution and diffusion, size of the drops, jets, sprays, and sheets as affected by the pressure and elevation of the water supply system, as well as the adsorption and interfacial surface tension of the wetted tower areas.

Dissolved gases and other impurities in the water influence the water-cooling process. Additional cooling tower variables affecting its performance include: location (ground, roof, nearby obstructions, wind orientation), dimensions, relative proportions (contour) of tower structure, materials, type and arrangement of interior surfaces; the louver and drift-eliminator designs as they facilitate the air flow to and from the tower; temperature of the structure at different points as influenced by the external and internal conditions. Other cooling tower factors to be considered are: loss of water by entrainment (drift loss), design and location of water-collecting basin, and surface evaporation therefrom, also the noise generated by the air, water, fan, and structure vibration.

Basically, a water-cooling tower is a heat exchanger in which heat flows from the water to the air: (1) by a flow of sensible heat from the warm water to the cooler air, and (2) by an exchange of latent heat resulting from the evaporation of a small part of the circulating water to increase the humidity ratio of the air by a corresponding amount. The general principles involved are similar to those encountered in the processes of diffusion in absorption and extraction equipment.[5,6]

Details of the application of the process to water-cooling tower performance have been published by various authorities,[7,8,9,10,11,12] and those interested in the derivation of the various equations should refer to these references, as listed at the end of this chapter. The approach in each case is based on a heat balance in which the total heat given up by the water equals the total heat absorbed by the air. These derivations are also based on certain assumptions, viz: that the specific heat of water is unity at the temperatures encountered; that there is no loss in weight of the circulating water as a result of evaporation; that the water suspended in the tower is surrounded by a film of air which is saturated with water vapor and at the temperature of the water surrounded; and that the basic theory of cooling tower operation proposed by Lewis[13] and developed by Merkel[14] is applicable. This theory refers to the fact that the numerical value of the coefficient of sensible heat transfer when divided by the numerical value of the coefficient of diffusion equals the specific heat (at constant pressure) of air. The reader should observe that this relationship refers to the numerical values of three distinct constants, the units for each being different. The above relationship makes it possible to simplify the heat

transfer equation by combining the two driving forces into one potential represented as the difference between the enthalpy of the air film (at water temperature) surrounding the water and the enthalpy of the main air stream.

Tower Performance Factor

The operations taking place in a typical water-cooling tower are shown in Fig. 10. If the reduction in water flow rate due to evaporation within the volume is neglected, and the usual concepts of heat flow and mass heat transfer are applied the equations typifying cooling tower operation are:

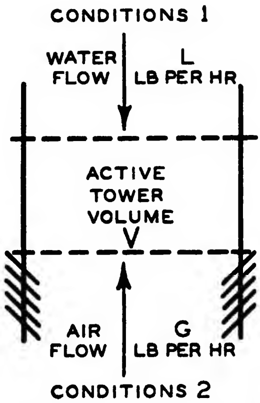


FIG. 10. OPERATIONS IN A TYPICAL WATER COOLING TOWER

$$\frac{KaV}{G} = \int_2^1 \frac{dh}{h'' - h_a} \tag{3}$$

and

$$\frac{KaV}{L} = \int_2^1 \frac{d\theta}{h'' - h_a} \tag{4}$$

where

a = over-all average wetted area (surface of water drops plus wetted tower surface) square feet per cubic foot of active tower volume.
G = weight rate of flow of air, pounds of dry air per hour.
h = enthalpy, Btu per pound of dry air.
h_a = enthalpy of air-vapor mixture, Btu per pound of dry air.
h'' = enthalpy of saturated air-vapor mixture at water temperature, Btu per pound of dry air.
K = over-all energy unit conductance, Btu per (hour) (square foot over-all average wetted area) (Btu enthalpy difference per pound of dry air).
L = water rate, pounds per hour.
θ = temperature of water in tower, Fahrenheit
θ_1 = temperature of inlet water, Fahrenheit.
θ_2 = temperature of outlet water, Fahrenheit.
V = active tower volume, cubic feet.

Either term $\frac{KaV}{G}$ or $\frac{KaV}{L}$ may be called the *Tower Performance Factor* or *Number of Tower Units* (NTU).

These equations indicate that the rate of heat transfer from the water to the air depends primarily upon the enthalpy of the air, the latter being dependent only on the wet-bulb temperature of the air; this explains the common observation that cooling tower performance is independent of inlet dry-bulb air temperature and that adiabatic conditions exist.

The integration of Equations 3 and 4 must be performed by mechanical or graphical means, because direct mathematical integration would be

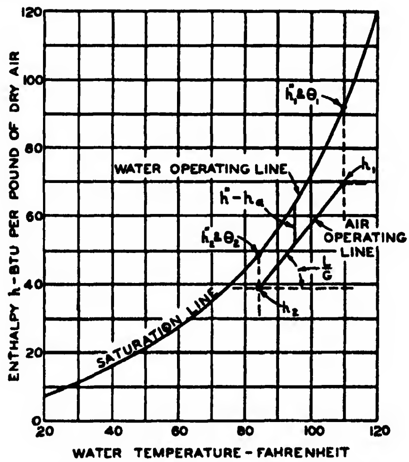


FIG. 11. TEMPERATURE-ENTHALPY DIAGRAM FOR AIR-WATER VAPOR MIXTURE SHOWING OPERATING LINES FOR EXAMPLE 2

accurate only within narrow temperature limits. The temperature enthalpy diagram in Fig. 11 represents the conditions for either of the above equations. The water is cooled from the temperature θ_1 to θ_2, and the enthalpy of the air film surrounding it follows the saturation line h''. Air enters the tower at a wet-bulb temperature of t_2' and an enthalpy of h_2, it is heated to an outlet wet-bulb temperature of t_1', having an enthalpy of h_1. Since the heat rejected by the water equals the heat absorbed by the air, the heat absorbed per pound of air is a function of the pounds of water per pound of air going through the tower, and the slope of the air operating line is the $\frac{L}{G}$ ratio.

Example 2. It is desired to cool 150,000 lb of water per hour (about 100 tons of mechanical refrigeration) from 110 to 84 F with 125,000 lb of dry air per hour with a design wet-bulb air temperature of 75 F. These conditions could prevail with a steam-turbine driven centrifugal refrigeration compressor. Determine the *Tower Performance Factor*; show in tabular form the successive steps for this mechanical integration by selecting two-degree intervals of the water-temperature range.

Solution. The accompanying Table 5 shows the sequence of mechanical integration for the given water and air temperatures. The first column shows the water temperature θ in increments of two degrees ($\Delta\theta = 2$ F deg). Column 2 gives the enthalpies

of the saturated air-vapor mixture at the water temperature, Btu per pound of dry air. The enthalpy of air, h_a in column 3, has an original value of 38.61 Btu per lb corresponding to the 75 F entering wet-bulb temperature of the ambient air; this value of h_a increases in equal increments of 1.2 $\left(\frac{L}{G}\text{ratio}\right)$ Btu per F deg, hence

$\Delta\theta \times \frac{L}{G} = 2 \times \frac{150{,}000 \text{ lb}}{125{,}000 \text{ lb}}$ 2.4. The potential for mass heat transfer is $(h'' - h_a)$ as shown in column 4, this is frequently called the tower driving force potential. The values in column 5 for each increment are determined by dividing 2.4 Btu per F deg by the average value of $(h'' - h_a)$; and column 6 is calculated in a similar manner, except that the increments are two degrees instead of 2.4 Btu.

TABLE 5. SEQUENCE OF MECHANICAL INTEGRATION TOWER PERFORMANCE FACTOR

1 WATER TEMP. θ	2 ENTHALPY OF FILM h''	3 ENTHALPY OF AIR h_a	4 ENTHALPY DIFFERENCE $(h' - h_a)$	5 $\frac{\Delta h}{(h' - h_a)}$ (avg.)	6 $\frac{\Delta\theta}{(h' - h_a)}$ (avg.)
84	48.22	38.61	9.61	0.249	0.208
86	50.66	41.01	9.65	0.247	0.206
88	53.23	43.41	9.82	0.241	0.201
90	55.93	45.81	10.12	0.232	0.194
92	58.78	48.21	10.57	0.221	0.184
94	61.77	50.61	11.16	0.208	0.173
96	64.92	53.01	11.91	0.194	0.162
98	68.23	55.41	12.82	0.180	0.150
100	71.73	57.81	13.92	0.165	0.137
102	75.42	60.21	15.21	0.150	0.125
104	79.31	62.61	16.70	0.137	0.114
106	83.42	65.01	18.41	0.124	0.103
108	87.76	67.41	20.35	0.112	0.093
110	92.34	69.81	22.53		
			Tower Performance Factor =	2.460 or	2.050

Hence, the mechanical integration for the above conditions gives two results:

$$\frac{KaV}{G} = \sum \frac{dh}{h'' - h_a} = 2.46, \textit{ Tower Performance Factor}$$

and

$$\frac{KaV}{L} = \sum \frac{d\theta}{h'' - h_a} = 2.05, \textit{ Tower Performance Factor}$$

The results obtained in Example 2 are designated as the *Tower Performance Factor* (*TPF*) or the *Number of Tower Units* (*NTU*); these figures represent correlated values that are directly proportional to the performance being considered. Similar calculations could be made for other quantities and temperatures of air and water. It should be noted that this factor is not related to the equipment doing the cooling, that any numerical value may represent an infinite number of possible performance conditions, and that any cooling tower arrangement may give almost any performance under certain conditions. Also the mechanical integration procedure used above applies only to counterflow apparatus. However, the same principles may be applied to crossflow atmospheric water-cooling towers although the method is more involved.

The basic mathematical theory for water-cooling towers is now well established and recognized, but each manufacturer relies upon experimental

results and practical experience with his own tower designs to establish a system for rating each unit that he builds. The problem of cooling tower design or selection is based on a knowledge of the characteristics of the equipment being considered. The *Tower Performance Factor* is a variable which is a function of the design; it also varies with the water loading and air velocity. Experimental data indicate that it varies with the heat load; although this variation may be due to deviations from the theoretical calculations which become more pronounced at the higher temperatures. The reference literature contains *Tower Performance Factors* which have been reported by various investigators, but the reader should be warned that the use of such factors without a full understanding of the source may lead to erroneous results.

SELECTION OF WATER-COOLING TOWERS

The correct type and size of water-cooling equipment for a given service cannot be determined intelligently without considering the characteristics of the various types, together with the many correlated requirement factors. Very few installations are exactly alike in details of requirements, hence conditions affecting performance and operation of the several types of water-cooling equipment vary widely because of the many diversified applications and wide-spread geographical locations.

Before the characteristics of a specific water-cooling apparatus can be judged desirable or undesirable for a given heat load and wet-bulb temperature, a survey should consider the importance of each of the following items: first cost including all necessary auxiliaries, area, height, weight, effect of wind velocity and direction, rigidity of structure to withstand high winds, safety, conformity to building codes, drift nuisance, make-up water requirements and cost of chemical treatment if needed, total power for pumping (plus fan operation in the case of mechanical draft), maintenance, available locations (with due thought to possible future expansion, wind restrictions, space cost, proximity and accessibility, etc.), appearance, the equipment's operating flexibility for the most economical conformance to varying loads or seasonal changes, and other considerations occurring with regard to a specific application.

For a definite heat-load dissipation, the type and size of a water-cooling tower is primarily affected by the following conditions:

1. Gallons per minute of cooling water.
2. Geographical location of the tower installation.
3. Wet-bulb design temperature of ambient air; see Table 3, Chapter 15.
4. Temperature of the hot water entering the tower at normal rating.
5. Temperature of the cold water leaving the tower at normal rating.
6. Ground, roof, or sub-structure installation.
7. Area available for cooling tower.
8. Proximity to other structures.
9. Surface of water exposed to each unit quantity of air.
10. Time of contact of the air with the water; this depends upon height (or length) of tower, and upon the relative velocity of air and water.

The selection of a proper water-cooling range depends upon: (1) type of service,—refrigeration, internal-combustion engine, or steam condenser, (2) wet-bulb air temperature at which the equipment must operate, and (3) type of condenser or heat exchanger employed.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 psig head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 psig and the power consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot-water temperature within 3 or 4 deg of the ammonia temperature corresponding

TABLE 6. CONDENSER DESIGN DATA

Gas	Desired Pressure in Condenser	Gas Temperature in Condenser, Fahrenheit	Leaving Hot-Water Temperature, Fahrenheit	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.4	120	114
Ammonia	185 psi*	96.0	92	88
Carbon dioxide	1030 psig*	86.0	83	80
Methyl chloride	102 psig*	100.0	96	92
Freon, F-12	117 psig*	100.0	96	93
Freon, F-12	126 psig*	105.0	100	97
Freon, F-12	136 psig*	110.0	104	101

* Head pressure.

to the head pressure, while a small condenser may require a 10 deg difference.

Table 6 lists several gases with data for the temperatures and pressures for which commercial condensers are designed. Careful evaluation of costs of water and electrical power should be made before deciding to use city water for jacket water and condensers. Economy of operation generally indicates the use of either a water-cooling tower or an evaporative condenser for most refrigeration installations of five tons or more capacity. Refer to Chapter 39, for information on Evaporative Condensers. Internal-combustion engines have limiting hot-water temperatures of 140 to 180 F for closed systems, and 110 to 130 F for open systems, depending upon the quality of the cooling water. The cooling of such fluids as milk or wort has variable requirements and is usually accomplished in counterflow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

OPERATION AND MAINTENANCE

Water Treatment. The amount of make-up water required by a cooling tower depends upon evaporation loss, drift loss, and blow-down. Evapora-

tion losses average 0.80 per cent of the water circulated for each 10 F deg range. Drift loss is the water carried out of the tower by the air currents in the form of droplets or mist. In properly designed induced draft towers this loss normally approximates one-tenth of one per cent, and most cooling tower manufacturers will guarantee a drift loss not to exceed two-tenths of one per cent. The amount of blow-down water wasted depends upon the hardness of the circulating water, type of water softening used and the amount of drift loss. Blow-down is normally controlled to maintain the concentration of soluble and scale-forming solids below the point where the formation of scale would occur or would be caused by corrosion.

Algae formations will plug nozzles and prevent proper distribution of the water over the tower filling. This growth may also collect on equipment served by the cooling tower, and thereby reduce the heat transfer rate. Algae should be held at a minimum or eliminated by use of bromine, chlorine, chlorinated lime, copper sulfate, or various blends of chlorophenates (see Chapter 51).

Although some scale-forming materials are found in practically all water, those which cause trouble in water-cooling systems are normally calcium and magnesium carbonates. Scale formation in equipment served also reduces heat transfer rates. Scale can be reduced materially or prevented by softening the make-up water with lime and soda ash, zeolite, or sulfuric acid, or by use of small amounts of sodium hexametaphosphate. Water softening or treatment requires close regulation and control by a competent chemist. Too high a concentration of soluble solids in cooling tower water may raise the temperature of the water leaving the tower and may cause sludge deposits or corrosion in the system. Concentration of solids is normally controlled by either blowing down or by a continuous overflow to the sewer. Refer also to Chapter 51.

Delignification. The presence of sodium carbonate in the circulating water results in delignification of any wood with which water comes in contact. This chemical dissolves lignin which binds the wood fibers together and leaves the wood surface in a white fibrous condition. Prolonged exposure reduces the structural strength of the wood. Delignification first appears on parts of the tower that are alternately wet and dry, since evaporation at such points rapidly increases the concentration of dissolved solids. The presence of sodium carbonate in harmful amounts is generally indicated by a high pH of 9 to 11. The effect of the sodium carbonate may be neutralized by the use of sulfuric acid. It is desirable to have the pH value of the water at 7 to 7.5 (7.2 pH value is neutral for redwood).

Two-speed motors. For readily adapting tower performance to temporary or seasonal decreases in heat load, and especially for winter operation, the use of two-speed motors (for fan drives) is recommended. The chief advantage is that when operated at half-speed, fans require only about 15 per cent of the power used at full speed. Particularly in multi-fan towers, the ready flexibility provided by two-speed motors results in considerable savings even though load reductions may sometimes call for only one or a few fans to be operated at half speed.

Cold-weather Operation. Extremely cold water normally does not increase performance to any great extent, but increases operating hazards considerably. Water-cooling towers operated in sub-freezing weather are subject to ice formation on the louvers and the outer portion of the filling.

To prevent icing in cold-weather operation, the cold raw water (tower circulating water) temperature should be maintained as high as practicable,

taking into consideration the effect upon the economy of the equipment served. One or more of the following procedures are recommended for induced draft towers: (a) Run two-speed motors on low speed, or shut off some of the fans; (b) Shut down some cells completely and put all of the water over the remaining cells; (c) Reduce water flow to the tower and shut off some of the cells; (d) By-pass the cooling tower with part of the water and shut off some of the fans or cells of the tower.

If ice should form on the louvers and filling, one of the following methods of removal can be used: (a) Reversing (for not more than 10 minutes) the rotation of the motor driving the fan and thus blowing the warm air out through the louvers; (b) Shutting down fans on some sections temporarily, but not the water. When these cells have thawed out, use the same procedure on other cells.

Where intermittent operation of a system is employed, water in outside basins may cause considerable damage due to freezing. To prevent this, such basins are drained when out of service and therefore in some small roof installations a tank large enough to hold all the water in the system may be installed inside the building.

Maintenance. Well-maintained equipment provides the best operating results and the least overall maintenance cost. A regular schedule should be set up for the structural and mechanical upkeep of water-cooling towers. The life and continued utility of any cooling tower is directly dependent upon its inherent qualities, climatic environment, type of service, severity of operation, and general care and maintenance.

REFERENCES

[1] Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S. Geological Survey*, Water Supply Paper No. 520 F).

[2] A.S.H.V.E. Research Paper—Design of Spray Cooling Ponds, by S. Hori, U. A. Patchett and L. M. K. Boelter (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1942, p. 624).

[3] Cooling Tower Performance Studies, by L. M. K. Boelter (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 615).

[4] Summer Weather Data; Statistics, Charts, Maps, and Analysis, by J. C. Albright, 1939. (The Marley Company, Inc., 1944).

[5] Principles of Chemical Engineering, by W. H. Walker, W. K. Lewis, W. H. McAdams and E. R. Gilliland (McGraw-Hill Co., 1937, p. 480).

[6] Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937, p. 91).

[7] Heat Transmission, by W. H. McAdams (McGraw-Hill Co., 1933, p. 157).

[8] Performance Characteristics of a Mechanically Induced Draft, Counterflow, Packed Cooling Tower, by A. L. London, W. E. Mason and L. M. K. Boelter (*A.S.M.E. Transactions*, January, 1940, Vol. 62, p. 41).

[9] Determination of Unit Conductances for Heat and Mass Transfer by the Transient Method, by A. L. London, H. B. Nottage and L. M. K. Boelter (*Industrial and Engineering Chemistry*, April, 1941, Vol. 33, p. 467).

[10] Graphical Method of Determining Number Transfer Units, by T. Baker (*Industrial and Engineering Chemistry*, August, 1935, Vol. 27, p. 977).

[11] Performance and Selection of Mechanical-Draft Cooling Towers, by Joseph Lichtenstein (*A.S.M.E. Transactions*, October, 1943, Vol. 65, No. 7, p. 779).

[12] Performance of Small Mechanical Draft Cooling Towers, by W. M. Simpson and T. K. Sherwood (*Refrigerating Engineering*, December, 1946, p. 535).

[13] The Evaporation of a Liquid into a Gas, by W. K. Lewis (*A.S.M.E. Transactions*, Vol. 44, 1922, p. 325).

[14] Verdustungs Kühlung, by H. Merkel (Forschungsarbeiten, No. 275, 1925).

CHAPTER 38

DEHUMIDIFICATION BY SORBENT MATERIALS

Definitions and Principles, Adsorbents, Dehumidification by Solid Adsorbents, Dehumidification Equipment Using Solid Adsorbents, Absorbents, Dehumidification by Liquid Absorbents, Dehumidification Equipment Using Liquid Absorbents, Calculation of Moisture Load, Vapor Transfer to Dehumidified Space

DEHUMIDIFICATION as used herein is the reduction of the water vapor content of a given volume of air or other gas. The term thus describes a special case of dehydration which covers the removal of moisture in any form from matter. The degree of dehumidification required varies greatly with different applications and is one of the prime considerations influencing the choice of a method. Dehumidification may be accomplished by latent heat removal together with sensible heat removal, as described in Chapters 7, 37, and 43, or by the use of sorbents.

Sorbents are substances which have the property of extracting and holding other substances (usually gases or vapors, *e.g.* water vapor), brought into contact with them. All materials are sorbents to a greater or lesser degree. The weight of water held by a substance will increase or decrease depending upon whether the vapor pressure of the water held by the substance is less or greater respectively than the partial pressure of water vapor in the surrounding atmosphere. As generally used, however, the term, sorbents, refers to those materials having a capacity for moisture which is large compared to their volume and weight. Such materials are divided into two general classifications:

1. *Adsorbent*—A sorbent which does not change physically or chemically during the sorption process. Certain solid materials, such as activated alumina, silica gel, activated bauxites, and activated charcoal have this property. The action of adsorbents, most of which adsorb some gases and condensible vapors besides water vapor, is selective. Thus, in the case of a mixture containing both water and organic vapors, silica gel would remove the water vapor in preference to the organic vapors, while the reverse would be true in the case of activated carbon. The selective property of adsorbents is made use of in some instances for the removal of objectionable and contaminating vapors from an air or gas mixture. (See Chapter 10.)

2. *Absorbent*—A sorbent which changes either physically, chemically, or both, during the sorption process. Calcium chloride is an example of a solid absorbent, while liquid absorbents include solutions of lithium chloride, calcium chloride, lithium bromide, and the ethylene glycols.

ADSORBENTS

The ability of an adsorbent to remove water vapor from a gas is explained by the fact that the vapor pressure of the water in the adsorbent (when in the reactivated condition) is less than the partial pressure of the water vapor in the surrounding atmosphere. For instance, when an active adsorbent is brought into contact with a gas of high humidity, there is a tendency for the vapor pressure of the water in the adsorbent to reach equilibrium with the partial pressure of the water in the surrounding gas, with

the result that water is extracted by the adsorbent and its weight increased, while the moisture content of the gas is correspondingly reduced. (The adsorbent is said to be saturated for a given condition when equilibrium is attained.) The weight of water a given adsorbent will extract is dependent upon the relative humidity (ratio of the partial pressure in the gas to the saturation pressure at a given temperature) and the temperature of the adsorbent. The process is reversible: if the temperature of the adsorbent is raised until the vapor pressure of the adsorbed water becomes greater than the partial pressure of the vapor in the surrounding atmosphere, water will be released by the adsorbent. After the adsorbent cools to room temperature, for instance, the vapor pressure of the water in the adsorbent falls below the partial pressure of the vapor in the atmosphere, and the adsorbent will again start extracting water. The elimination of water by the addition of heat is known as reactivation and is a means of regenerating the adsorbent so that it may be used repeatedly.

Adsorption is proportional to the amount of surface (internal and external) of the sorbent. The materials that are used commercially as solid adsorbents have a porous structure of sub-microscopic dimensions, which gives them extensive surface area. An adsorbent should meet the following requirements in order to be satisfactory for dehumidification purposes:

1. Have a high adsorptive capacity under normal atmospheric conditions.
2. Be chemically stable, resisting contamination from impurities.
3. Be physically rugged to resist breakdown from handling and use.
4. Be capable of reactivation at temperatures generally obtainable.
5. Be heat-stable at reactivation temperatures.
6. Have a weight per unit volume such as to avoid excessive bad dimensions.
7. Be available at reasonable cost.

Activated Alumina

Activated alumina is a granular porous material which removes by adsorption substantially 100 per cent of the moisture from gases, vapors, and certain liquids. Regeneration or reactivation may be accomplished by employing a heating medium at temperatures ranging from 350 to 600 F. After many cycles of adsorption and reactivation it is substantially as effective as originally and retains its original size and shape.

Activated alumina is produced by chemically controlled precipitation from a sodium aluminate solution resulting from the extraction of alumina from bauxite by the Bayer process. By subsequent processes this precipitate is converted into a highly porous adsorptive material. It is low in iron and silica, each normally less than 1/10 of 1 per cent. Commercially produced material is uniform in analysis and physical form.

Activated alumina is a partially dehydrated aluminum trihydrate containing about 7 per cent water and small amounts of soda, oxides of iron, silicon, and titanium, as well as very minor amounts of other elements indicated spectrographically. Substantially all of the soda is combined with silica and alumina as an insoluble constituent.

Activated alumina is inert chemically to most gases and vapors, is nontoxic, and will not soften, swell or disintegrate when immersed in water. High resistance to shock and abrasion are two of its more important physical characteristics. Commercial sizes range from a powder passing through 300 mesh screen to particles 1 in. in diameter. The sizes commonly used are 8–14 mesh and $\frac{1}{4}$ in. to 8 mesh. The average weight for most forms is

50 lb per cubic foot. Its high degree of purity warrants classification among commercially pure chemicals.

Silica Gel

Silica gel is a prepared form of silicon dioxide (silica) having an extremely porous structure which makes it an efficient adsorbent. It is made by mixing predetermined concentrations of an acid, such as sulfuric acid, and a soluble silicate, usually sodium silicate, and allowing the mixture to set to a jelly-like mass called hydrogel. The product takes its name from its condition as a colloid at this stage of its manufacture. After setting, the hydrogel is broken into small lumps, washed, dried, crushed, and screened to the desired particle sizes and then given a final heat treatment or activation. The surface area of silica gel has been found to be in excess of 50,000 sq ft per cubic inch of product. Silica gel has high adsorptive capacity per unit weight and may be reactivated repeatedly at temperatures up to 600 F. Reactivation is generally accomplished by blowing gases through the silica gel at approximately 300 F, or by heating in a well-vented oven maintained at this temperature until no more moisture is given off. Silica gel is a high purity, rugged, non-toxic, heat-stable material, having a specific heat of 0.2 and is most inert. There is no change in the size or shape of the particles as it becomes saturated and no corrosive or injurious compounds are given off. It is available commercially in a number of grades, ranging in particle size from a 3 to 8 mesh product to an impalpable powder passing through a 325 mesh screen. The product generally used for dehumidification applications has a particle size of 6 to 12 mesh and a bulk density of between 40 and 45 lb per cubic foot.

Activated Bauxite

Activated bauxites are certain natural products which after controlled heat treatment have properties which make them suitable for use as solid adsorbents. They are marketed under different trade names by several processors. The activated bauxites consist primarily of Al_2O_3. Fe_2O_3, SiO_2, TiO_2, and H_2O in varying percentages. The surface area, adsorptive capacity, and other properties of the several products differ to some extent depending upon the source of the original material and its subsequent treatment. The available activated bauxites are durable products having a specific heat of about 0.24 and can be regenerated at temperatures between 300 F and 500 F. They are usually supplied in a number of particle sizes and have a bulk density of between 55 and 65 lb per cubic foot.

There are other solid substances having marked adsorbent properties, but details concerning them are not available.

DEHUMIDIFICATION BY SOLID ADSORBENTS

Since adsorption is primarily a condensation process, heat equivalent to the latent heat of evaporation of the vapor plus the heat of wetting, which is an additional amount of heat depending upon the vapor being adsorbed and the adsorbent used, is liberated. The sum of the latent heat of evaporation and the heat of wetting is known as the heat of adsorption. During adsorption it might be said that latent heat is transformed into sensible heat, which is dissipated into the adsorbent, into the metal of the adsorbent container, and into the gas mixture, resulting in a rise in temperature. Fig. 1 shows the relationship between temperature, vapor pressure, and moisture content of a solid adsorbent. These curves indicate the general performance of solid adsorbents, although the exact values vary for the

different adsorbents and may vary even for different types of the same compound. The effects of vapor pressure and temperature upon the moisture content of an adsorbent may be observed by referring to Fig. 1. When the given type of solid adsorbent is in equilibrium with air having a dry-bulb temperature of 70 F and 70 per cent relative humidity, that is, having a dew-point of 60 F or a water vapor pressure of 13.2 mm Hg, the water content of the adsorbent is 33 per cent. With air having the same dry-bulb

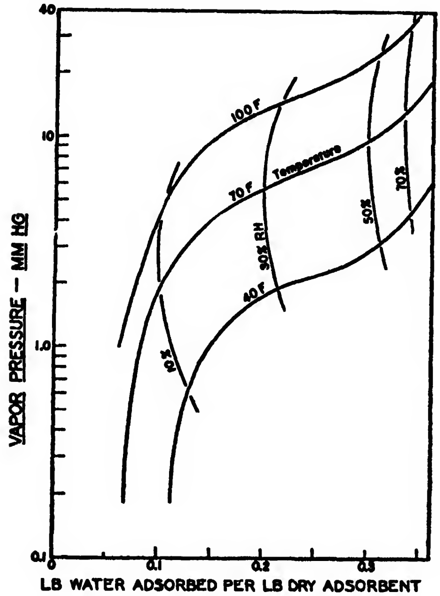


Fig. 1. Temperature—Vapor Pressure—Concentration Characteristics for a Typical Solid Adsorbent

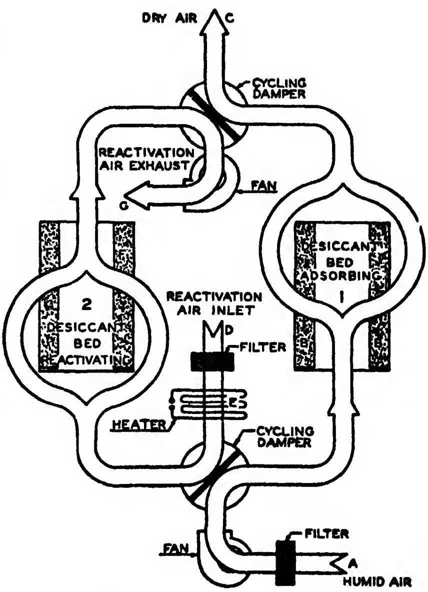


Fig. 2. A Typical Solid Adsorbent Dehumidification Unit Air Flow Diagram

temperature and a dew-point of 37 F, or a vapor pressure of 5.6 mm Hg, the water content is 20 per cent. The increase in weight for an activated solid adsorbent after it reaches equilibrium with a gas of any given water vapor content may be found by subtracting the residual water content (for example 6 per cent) from the equilibrium value. In the case of the two examples cited, the actual water gain would be 27 per cent and 14 per cent respectively. The effect of temperature upon the adsorptive capacity may be observed by following the 5.6 mm Hg vapor pressure line. At a temperature of 70 F the moisture content of the adsorbent is 20 per cent, while at 100 F the equilibrium water content is 11 per cent.

In practice the temperature rise in the dehumidified air caused by the adsorption heat is approximately 10 deg F for each grain of moisture removed per cubic foot of air at atmospheric pressure. This temperature rise occurs progressively through the adsorbent bed and is an important consideration in predetermining the performance of a given design of apparatus. Data such as these, together with information covering other characteristics such as specific heat, resistance to air flow, etc., are of value in the basic design of adsorption apparatus. In the solution of air conditioning prob-

lems, however, reference must be made to performance data on established apparatus designs.

DEHUMIDIFICATION EQUIPMENT USING SOLID ADSORBENTS

A typical solid adsorbent dehumidification unit air flow diagram is shown in Fig. 2. The apparatus consists of two adsorbent containers (adsorbers) with necessary interconnecting piping, valves, and auxiliaries consisting of filters, fans, activation air heater, controls, and, in some instances, a cooler for the dehumidified air. Before entering the adsorber the air to be dehumidified is drawn into a filter to remove dust and other impurities. In passing through the adsorbent bed, the moisture content of the air is reduced and the dehumidified air is then introduced into the space or process requiring it. While the first adsorber is dehumidifying the air, the second adsorber is being reactivated by means of outside air drawn through a filter and heater in which its temperature is raised to 300 F. The heat may be supplied by electric heating elements, steam coils, the direct products of combustion of gas, oil, waste heat, or any other convenient source. In passing through the adsorbent bed, the hot gases supply the necessary heat for releasing adsorbed water from the adsorbent and then carry it out of the adsorber to the activation gas outlet, where it is exhausted to the outside atmosphere. In some instances a thermostat placed in the activation outlet connection shuts off the activation fan and heater when the adsorbent is completely reactivated, as indicated by a rapid rise in the temperature of the outlet activation gas. The length of the adsorption period may be controlled by a timing device which changes the valves or dampers from the adsorbing to the activating position, or by a humidistat located in the dehumidified air connection or in the dehumidified space. The majority of commercial units are time controlled.

In applications where a continuous stream of dehumidified air is not required, a single adsorber type unit may be used, while in other cases where a continuous stream of dehumidified air is required, a multiple number of adsorbers or even a continuously rotating system may be used.

If the air to be dehumidified is very warm, and especially where exceedingly low dew-point dehumidified air is required, it is advantageous to install a pre-cooler to reduce the temperature of the inlet air. In this way the working temperature in the adsorber is lowered and the over-all performance appreciably increased. Some equipment manufacturers install cooling coils in the adsorbent beds for the same purpose, while others divide the adsorbent bed and install coolers between the sections.

Dehumidification equipment is employed to the best advantage where the air conditioning problem is primarily one of obtaining low relative humidity control rather than temperature control. This requirement is found in the case of the preservation of inactive naval vessels where the interior of the ship must be kept at a relative humidity below 30 per cent to avoid corrosion, mold, mildew, and other moisture damage that occurs at humidities substantially in excess of this figure. Other advantageous applications for dehumidification systems are found in industrial processes where low relative humidity atmospheres are required during the manufacture as well as for preservation of the finished products. Dehumidification with cooling may be used to advantage in work-rooms or other spaces occupied by humans, where the moisture load is high in comparison to the sensible heat load. In many instances where independent control of temperature and humidity is important dehumidification is used to advantage in conjunction with cooling.

ABSORBENTS

Any absorbent substance may be used as a dehumidifying agent if it has a vapor pressure with respect to water lower than the partial pressure of the water vapor in the mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are generally the solid forms of the liquid absorbents. Calcium chloride is frequently used because of low cost. At present they are used principally in small desiccating chambers and in small dryers of the cartridge type, through which air is forced under pressure.

Liquid Absorbents. These are primarily water solutions of materials in which the vapor pressure is reduced to a suitable level by controlling the concentration and temperature of the dehymidifying solution. Water solutions of the chlorides or bromides of various inorganic elements and certain organic compounds are the liquid absorbents used in air conditioning.

In addition to having suitable water vapor pressure characteristics, an absorbent to be satisfactory should also meet the following requirements:

1. Be widely available at low cost.
2. Be non-corrosive, odorless, non-toxic, and non-inflammable.
3. Be chemically inert against any impurities in the air stream.
4. Be stable over the range of use.
5. Must not precipitate at the lowest temperature to which the apparatus is exposed.
6. Have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by the moisture absorbed.

DEHUMIDIFICATION BY LIQUID ABSORBENTS

In liquid absorption systems the air-vapor stream is brought into intimate contact with the absorbent solution by passing the air stream through a tower into which the brine is introduced as a finely divided spray, or by passing the air through a tower or contactor which is continuously sprayed with the brine, thereby presenting a large surface of absorbent to the air to be dehumidified. The difference in the partial pressure of the water in the concentrated brine and the partial pressure of the water vapor in the air causes the water vapor to be given up by the air to the brine until equilibrium is approached. The water vapor is condensed during this operation, and its addition to the absorbent solution results in a decrease in the concentration of the solution. As the water vapor condenses the latent heat of condensation is released in the absorbent solution. An additional, frequently appreciable, quantity of heat known as the heat of solution or heat of mixing is also released. The heat released as the result of condensation and mixing is directly transferred to the brine, to the equipment, and to the air being dehumidified, thereby causing a rise in temperature.

A modified system includes means for removing heat from the absorbent solution, either within the contactor or externally. Thus the temperature of the solution may be higher than, equal to, or lower than that of the air, depending on the chemical employed and the ultimate use of the dehumidified air.

Fig. 3 shows the relationship between temperature, vapor pressure and moisture content of a typical absorbent of the *inorganic* type. These curves indicate the general performance of such absorbents, although the exact values vary for the different compounds.

When the given absorbent is in equilibrium with air having a dry-bulb temperature of 70 F and a relative humidity of 70 per cent, *i.e.* having a dew-point of 60 F, or a water vapor pressure of 13.2 mm Hg, the water

content of the solution is 4.8 lb water per pound of anhydrous absorbent. With air having the same dry-bulb temperature and a dew-point of 37 F, or a vapor pressure of 5.6 mm Hg, the water content is 2.1 lb per pound of absorbent. Therefore, when a solution of 2.1 lb water per pound of absorbent is exposed to an atmosphere of 70 F and 70 per cent relative humidity, it will absorb an additional 2.7 lb of water in reaching equilibrium.

The effect of temperature on the absorptive capacity may be observed by following a constant vapor pressure line in Fig. 3. At a temperature of 70 F and a vapor pressure of 13.2 mm Hg the moisture content is 4.8 lb

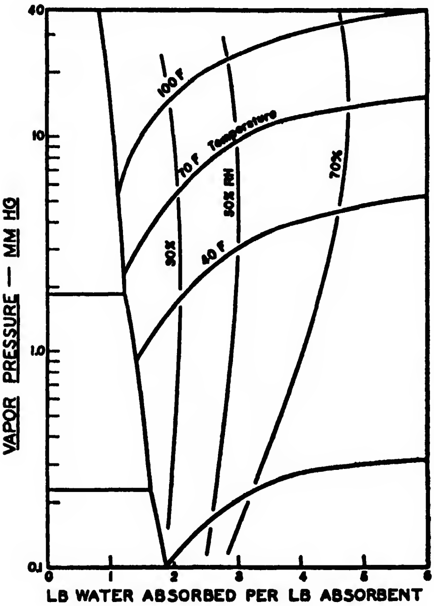


FIG. 3. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION CHARACTERISTICS FOR A TYPICAL INORGANIC ABSORBENT (HALOGEN SALT GROUP)

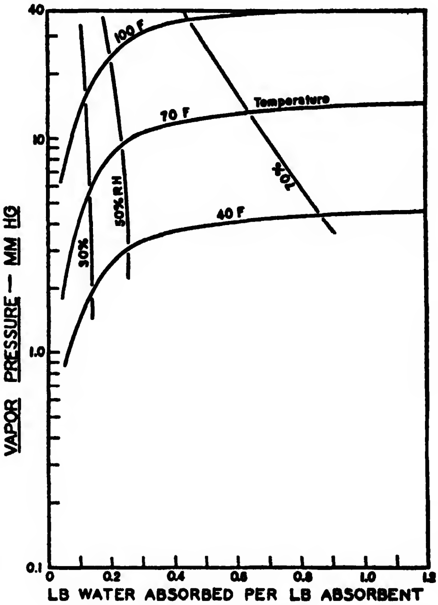


FIG. 4. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION CHARACTERISTICS FOR A TYPICAL ORGANIC LIQUID ABSORBENT

water per pound of absorbent. At 100 F the moisture content is 1.8 lb water per pound of absorbent.

Fig. 4 shows the relationship between temperature, vapor pressure, and moisture content of a typical liquid absorbent of the *organic* type.

DEHUMIDIFICATION EQUIPMENT USING LIQUID ABSORBENTS

One type of system utilizing liquid absorbents includes an external interchanger having essential parts consisting of a liquid contactor, a solution concentrator, a solution heater and a cooling coil all as shown in Fig. 5. The contactor and cooling coil are located in the wet air stream. The air to be conditioned is brought into contact with an aqueous brine solution having a vapor pressure below that of the entering air, resulting in a transfer of moisture from the air to the brine solution. This results in a conversion of latent heat to sensible heat which raises the solution temperature and consequently the air temperature. The temperature change of the air be-

ing processed is determined by the cooling water temperature and the amount of moisture removed in the equipment. Control of leaving air temperature may be obtained by precooling the absorbent solution in a suitable surface cooler, by tap, well, or chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor and returns it to the main brine reservoir for re-cycling. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine by the heating coils into a stream of regeneration air taken from and rejected to the outside atmosphere. Low pressure steam is normally used for heating the brine. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse

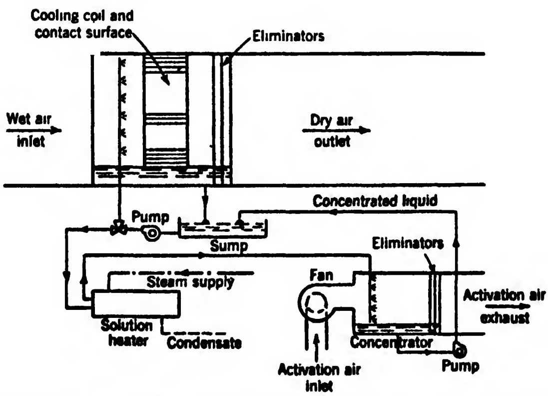


FIG. 5. LIQUID ABSORBENT EQUIPMENT IN WHICH SOLUTION COOLER AND CONTACTOR ARE COMBINED

of the absorption process. During concentration the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehumidification, the reverse is the case. Utilization of this principle permits winter humidification by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated into, instead of being condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

CALCULATION OF MOISTURE LOAD

Calculation of the dehumidification required to maintain lower than normal moisture content in a given room begins with determination of the rate of moisture gain in the room from all sources. It is common practice when maintaining a low humidity ratio to recirculate a large percentage of the air in the room through the dehumidifier, and to add only enough outside air to meet the needs of the problem. The humidity ratio of the mixture of outside and recirculated air and the dehumidifier performance data can be used to calculate the humidity ratio of the air leaving the dehumidifier. The difference between the humidity ratio of the air in the room and that of the dehumidified air entering the room represents the

effective dehumidification per pound of air. The rate of internal moisture gain in grains per minute divided by the effective dehumidification in grains per pound of air equals the air quantity required in pounds per minute. The following typical example using arbitrary values shows a general method of determining the dehumidifying requirements. Sensible heat determination considerations are discussed in other chapters and are purposely omitted here.

Example 1. A solid adsorbent dehumidifier having performance characteristics as shown in Fig. 6 is to be used to maintain inside conditions of 73 F and 20 per cent relative humidity, *i.e.* 24.1 grains per pound of dry air, 30 F dew-point, in a room, 20 ft x 30 ft x 10 ft high, having a total wall, ceiling, and floor surface area of 2200 sq ft. Outside design conditions are 72 F dew-point (118.4 grains per pound).

Internal sources of moisture are: 4 occupants; an open natural gas burner using 15 cu ft of natural gas per hour; an open top water tank, having an area of 2 sq ft exposed surface, in which water is maintained at 87 F, with air movement over the water surface being 100 fpm. Determine the quantity and condition of the dehumidified air to be supplied to the room.

Solution. The internal moisture gain consists of items 1 to 5.

		Grains per Minute
1. From occupants:	$4 \times 1800/60 =$	120
1800 grains per person per hour is obtained from Fig. 7, Chapter 12, by interpolation between curves C and D		
2. From burned gas:	$15 \times 650/60 =$	162
1 cu ft natural gas produces approximately 650 grains of moisture.		
3. From exposed water surface:	$2 \times 20 =$	40
Evaporation from water surface is assumed to be 20 grains per (minute) (square foot) at 87 F water with air movement of 100 fpm.		
4. From infiltration:	$\frac{6000}{60 \times 13.56} \times (118.4 - 24.1) =$	696
One air change, 6000 cu ft, assumed per hour (see Chapter 8).		
5. Moisture transmitted through room surface:	$\frac{2200}{60} \times 3 \times (0.783 - 0.176) =$	67
Permeability assumed to be 3 grains per (square foot) (hour) (inch Hg vapor pressure difference on two sides of wall).		
Total moisture gain from internal sources		1085

Let q be the air delivered to the room, pounds per minute. Let it be assumed for this problem that 85 per cent of the air is recirculated and 15 per cent is outside air. Enough air must be supplied to replace leakage from the system or to satisfy normal ventilating requirements for the occupants of the room as given in Chapter 12, whichever is greater. The amount is estimated from experience or obtained by test.

The humidity ratio of the mixture of recirculated and outside air entering the dehumidifier is then:

$$\frac{0.85q(24.1) + 0.15q(118.4)}{0.85q + 0.15q} = 38.3 \text{ grains per pound entering dehumidifier.}$$

For the dehumidifier shown in Fig. 6, for 38.3 grains per pound in entering air, the leaving humidity ratio will be 6.5 grains per pound.

Effective dehumidification in the room is 24.1 − 6.5 or 17.6 grains per pound of supply air.

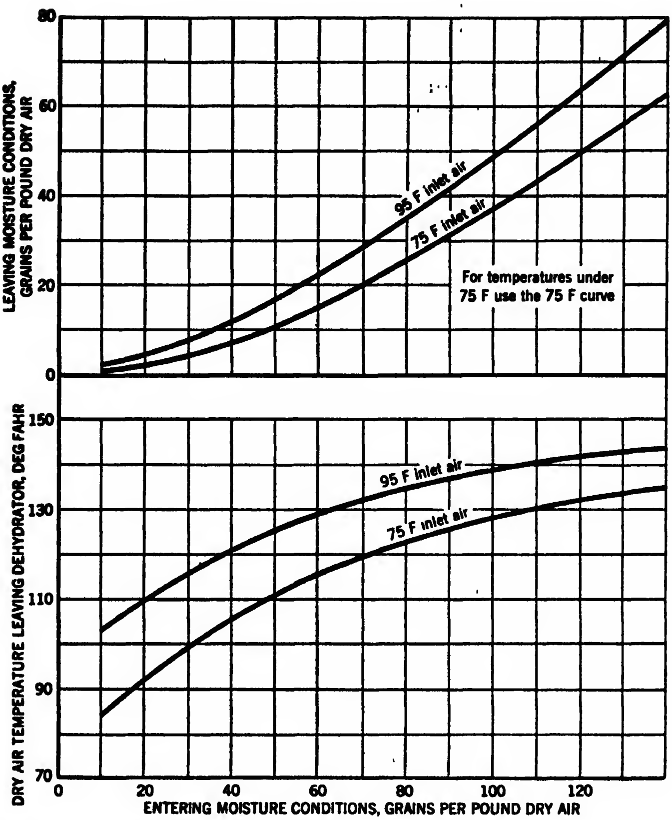


Fig. 6. Performance Data for Typical Commercial Solid Adsorbent Dehumidifier

Then q $\dfrac{\text{per minute}}{\text{17.6 grains per pound}}$ = 61.6 lb air per minute *minimum* that must be supplied to the room to maintain 30 F dew-point.

Note that this figure represents the minimum requirement for the arbitrary conditions set forth and that in practice safety margins should be added to the outside air percentage figure and to the calculated internal moisture gain.

VAPOR TRANSFER TO DEHUMIDIFIED SPACE

The walls enclosing a dehumidified space are subjected to a vapor pressure differential. The pressure of the vapor outside the walls tends to force moisture through the walls into the dehumidified zone of relatively low vapor pressure. As this process can be an unnecessary load on the dehumidifying equipment, provisions should be made for keeping the vapor transfer to a minimum. Also, if the space is cooled below the ambient dew-point there is a possibility that condensation may occur within the walls unless vapor transfer is controlled. For these reasons a vapor barrier should be located within the wall construction as near to the high vapor pressure side as feasible. To be effective a barrier must be continuous and should be so located within the structure that it will be protected from rupture. (See section on Vapor Transmission Through Materials, Chapter 6.)

CHAPTER 39

REFRIGERATION

Refrigeration Theory: Definitions and Basic Concepts, Refrigerants, Vapor Compression Refrigeration Cycles, Clearance and Volumetric Efficiency, Complex Refrigeration Cycles, Air Cycle, Steam Jet, Absorption and Ice Systems, Heat Pump; Basic Refrigeration Equipment: Compression Machines, Condensers, Evaporators and Coolers; Refrigeration Controls and Piping; Equipment Characteristics and Selection

WITH the increasing use of all-year comfort air conditioning installations the importance of refrigeration to the air conditioning engineer has been greatly magnified. The details of equipment operation, maintenance, and design remain problems for the refrigeration engineer, but the air conditioning engineer does retain a responsibility to the customer which requires some knowledge on his part of the different refrigeration cycles and the relative merits of each. In order to assist in meeting this need, the present chapter has been divided into four parts, the first covering the fundamental technical relationships which govern the selection and analysis of an operating cycle, the next two presenting brief discussions of basic refrigerating equipment and auxiliaries, and the last, information on selection criteria.

REFRIGERATION THEORY

Definitions and Basic Concepts

The *ton of refrigeration* is a quantity unit which originated in the days when harvested ice was the principal source of summer cooling. By definition the *ton* is the cooling effect realized when one ton of 32 F ice melts to water at 32 F; since the latent heat of fusion of ice is 144 Btu per pound, the *ton* represents a unit cooling effect of 144 × 2,000 = 288,000 Btu. In common practice the ton is usually considered a rate (rather than quantity) unit and is taken as 288,000 Btu per day (24 hours) or 12,000 Btu per hour or 200 Btu per minute. Thus for air conditioning calculations, the size of the requisite refrigeration machine, expressed in tons, can be obtained by dividing the heat gain of the structure, expressed in Btu per hour, by 12,000. In equation form:

$$H_t = (\text{Btu per hour heat gain}) \div 12{,}000 \tag{1}$$

where

H_t = load in tons.

The *working substance*, or *refrigerant*, is the fluid which carries heat through the refrigeration cycle from the evaporator, where heat enters the refrigerant, to the condenser where the heat is discharged to some cooling medium. The great majority of modern refrigeration systems use a liquefiable vapor as the working substance. By altering the pressure of the refrigerant its boiling temperature is changed, allowing the material to *boil* in the evaporator at a temperature sufficiently lower than that of the conditioned space to insure maintenance of an effective heat transfer rate from

the space (or in some cases from a secondary cooling fluid such as brine or cold water) to the refrigerant. The vapor formed in the evaporator is then raised in pressure (by a compressor, or by the absorber-generator combination of the absorption system) until its new boiling temperature exceeds the temperature of the available cooling medium; under these conditions heat transfer is established from the refrigerant vapor to the cooling medium with resultant condensation of the refrigerant. When condensed, the high-pressure liquid refrigerant is reduced in pressure and again allowed to *boil* in the evaporator.

In order to permit evaluation of the effectiveness with which any given cycle operates, some term is desirable which would be comparable to the *efficiency* that is used for heat engines. In refrigeration the desired effect is heat extraction and the cost of achieving this extraction is the amount of energy which must be supplied as shaft work. Thus the ratio of refrigerating effect to the heat equivalent of the compressor work is used as a measure of effectiveness and is defined as the coefficient of performance.

If the desired effect is the rejection of heat through the condenser instead of heat extraction through the evaporator, the refrigeration system is then termed a heat pump. In this case the coefficient of performance is the ratio of the heat rejected from the condenser to the heat equivalent of the compressor work. The coefficient of performance for the heat pump is greater than that for a system operating as a refrigerating machine because all mechanical shaft work required to operate the compressor is dissipated as useful heat through the condenser.

The Carnot cycle, an ideal, thermodynamically reversible cycle consisting of an adiabatic expansion and an isothermal expansion followed by an adiabatic compression and an isothermal compression to form a closed cycle, may be shown to be a measure of the maximum possible conversion of heat energy into mechanical energy. In its reversed form it is a measure of the maximum performance possible for any refrigeration cycle operating either as a refrigerator or as a heat pump. Although it cannot be applied in an actual machine because of the impossibility of obtaining complete reversibility, it is, nevertheless, extremely valuable as a criterion of inherent limitations. The coefficient of performance (CP) of a reversed Carnot cycle system operating as a refrigeration system is:

$$(CP) = \frac{T_s}{T_c - T_s} \qquad (2)$$

where

T_s = evaporator temperature, Fahrenheit degrees, absolute.
T_c = condenser temperature, Fahrenheit degrees, absolute.

With the ideal Carnot cycle operating as a heat pump, the coefficient of performance is:

$$(CP) = \frac{T_c}{T_c - T_s} \qquad (3)$$

The Carnot cycle coefficient of performance for both a refrigerating machine and a heat pump increases as the spread between the evaporator and the condenser temperatures decreases. In general, the same is true for an actual system operating as either a refrigerating machine or a heat pump.

Refrigerants

A desirable refrigerant should possess chemical, physical, and thermodynamic properties which permit its efficient application in refrigerating systems. In addition, when the volume of the charge is large, there should be little or no danger to health or to property in case of its escape.

Thermodynamically, a material for use as a refrigerant should have a large latent heat of vaporization since it is this heat quantity—subject to minor variations—which constitutes the working effectiveness of the refrigerant. Further, since the work required to compress a vapor increases rapidly with the pressure ratio, the thermodynamic characteristics of the fluid should be such that the required low-to-high temperature range can be achieved with only a moderate change in ratio. A further consideration, from the standpoint of practical operating effectiveness, is that the suction pressure should not be below atmospheric (to prevent leakage of air into the refrigerant lines) nor should the condenser pressure be excessively high (to prevent need for extra-heavy construction). The specific volume-specific enthalpy relationship is also important because some materials would have such low density, when in vapor form, that impractical compressor displacements would be needed to handle the suction vapor.

Properties of refrigerants are usually given either in tabular or graphic form. In contrast to the temperature-entropy plotting which is used almost exclusively in steam-power work, refrigeration problems are usually referred to a pressure-enthalpy chart. The advantage of pressure-enthalpy plotting is that linear distances on the chart correspond to energy gains or losses and the two types of processes, constant-pressure and constant-enthalpy, which occur most frequently in refrigeration cycles, can both be represented by straight vertical or horizontal lines. Figures 1 and 2 present pressure-enthalpy charts for ammonia and dichlorodifluoromethane (Freon-12). Although tabular arrangements of refrigerant properties require interpolation between values, they have the advantage of an accuracy greater than that obtainable from a chart. Tables 1, 2, 3, and 4 give the thermodynamic properties of four of the more common refrigerants: dichlorodifluoromethane (Freon-12), monochlorodifluoromethane (Freon-22), ammonia, and monofluorotrichloromethane (Freon-11); the first three of these materials are commonly used in reciprocating compressors; the last refrigerant is used in centrifugal machines.

Referring to Table 1, the first column gives the range of saturation temperatures likely to occur in practice. The second column gives the saturation pressure expressed in pounds per square inch absolute corresponding to a given temperature, while the next six columns give the three fundamental specific properties, volume, enthalpy, and entropy, of the saturated liquid and saturated vapor respectively. The last four columns give values of enthalpy and entropy for gases with 25 deg and with 50 deg of superheat; note particularly that the column heading *50 F superheat* means, not that the gas is at a temperature of 50 F, but that its temperature exceeds by 50 deg the saturation temperature corresponding to its actual pressure. Thus F-12 vapor at 38.0 psig and 91 F possesses 50 deg of superheat since its saturation temperature corresponding to 52.7 psia is 41 F.

The tabular arrangements of refrigerant properties are literally for saturated or superheated materials only. In many cases, however, the engineer must work with sub-cooled liquids. With an accuracy sufficient for all practical purposes the specific volume and the enthalpy of any sub-cooled refrigerant can be taken as equal to the values read from the tables for a saturated liquid at the same temperature. Thus if F-12 at 121 psia and

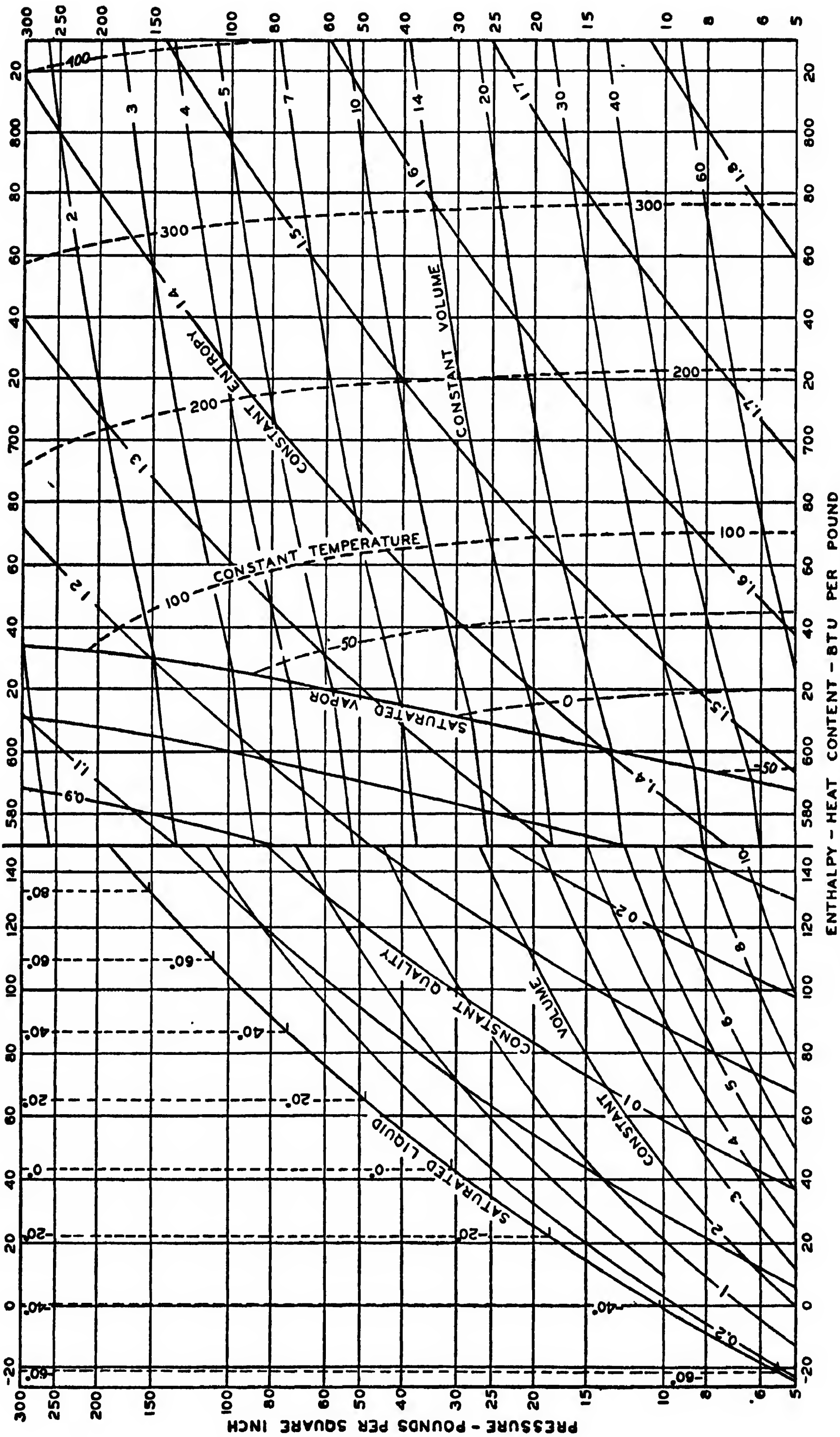


FIG. 1. PRESSURE-ENTHALPY DIAGRAM FOR AMMONIA

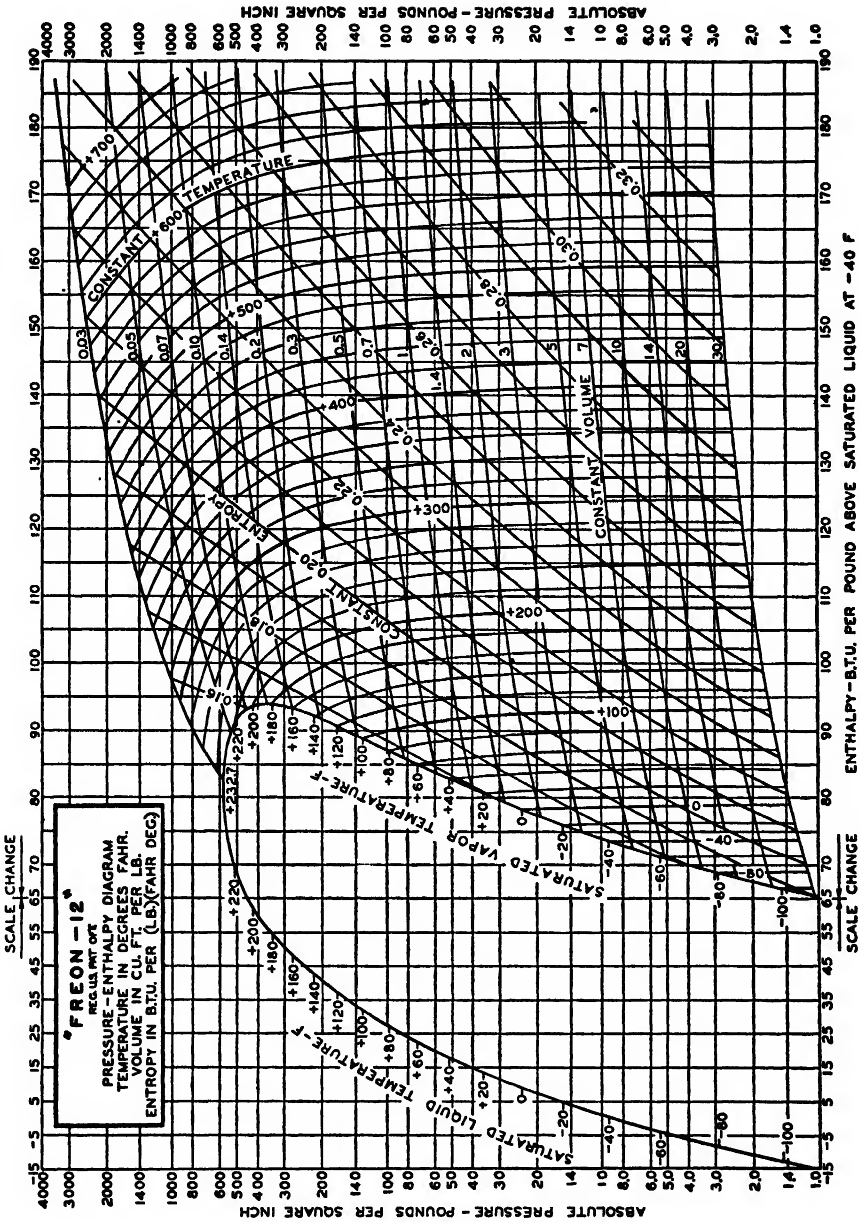


Fig. 2. Pressure-Enthalpy Diagram for Freon-12

40 F is passing through a pipe, its volume and enthalpy can be determined from Table 1 as 0.0116 cu ft per pound and 17.0 Btu per pound.

Frequently it is necessary to determine the properties of a *wet* vapor or of a mixture of liquid with some added vapor, such as is found at discharge from an expansion valve. This can be done from the tables by noting that the specific enthalpy of the mixture must be equal to that of the saturated liquid plus a fraction of the latent heat of vaporization equal to the fraction

Table 1. Properties of Dichlorodifluoromethane (F-12)

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Enthalpy and Entropy Taken From −40 F: Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
2	24.89	0.0110	1.574	8.67	78.44	0.01961	0.17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0.02052	0.17060	82.17	0.17795	85.76	0.18511
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86.26	0.18477
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10.82	79.59	0.02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33.08	0.0112	1.207	11.70	80.05	0.02601	0.16974	83.61	0.17706	87.26	0.18413
18	34.40	0.0113	1.163	12.12	80.27	0.02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0.18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0.0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0.0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0.18291
36	48.13	0.0116	0.848	16.10	82.27	0.03502	0.16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0.819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0.0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90.28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0.0117	0.742	17.91	83.15	0.03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0.03948	0.16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04215	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0.0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92.28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0.04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0.16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.17425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
72	87.50	0.0121	0.479	24.37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106
74	90.20	0.0122	0.464	24.84	86.22	0.05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0.18091
78	95.85	0.0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94.57	0.18083
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0.378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0.0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0.17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998
104	139.0	0.0128	0.302	32.15	88.95	0.06490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97.73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98.11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0.254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939
122	176.2	0.0132	0.233	36.66	90.28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0.227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0.214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	95.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204.8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

Table 2. Properties of Monochlorodifluoromethane (F-22)

Sat Temp F	Abs Press Lb per Sq In.	Volume		Enthalpy and Entropy Taken from −40 F							
				Enthalpy		Entropy		50 Deg Superheat		100 Deg Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	38.79	0.01192	1.373	10.63	105.02	0.0240	0.2293	112.35	0.2446	120.00	0.2590
2	40.43	0.01195	1.320	11.17	105.24	0.0251	0.2289	112.59	0.2442	120.26	0.2586
4	42.14	0.01198	1.270	11.70	105.45	0.0262	0.2285	112.83	0.2438	120.52	0.2581
5	43.02	0.01200	1.246	11.97	105.56	0.0268	0.2283	112.95	0.2436	120.65	0.2579
6	43.91	0.01201	1.221	12.23	105.66	0.0274	0.2280	113.07	0.2434	120.78	0.2577
8	45.74	0.01205	1.175	12.76	105.87	0.0285	0.2276	113.31	0.2430	121.04	0.2572
10	47.63	0.01208	1.130	13.29	106.08	0.0296	0.2272	113.55	0.2426	121.30	0.2568
12	49.58	0.01211	1.088	13.82	106.29	0.0307	0.2268	113.79	0.2422	121.56	0.2564
14	51.59	0.01215	1.048	14.36	106.50	0.0319	0.2264	114.02	0.2418	121.82	0.2560
16	53.66	0.01218	1.009	14.90	106.71	0.0330	0.2260	114.25	0.2414	122.08	0.2556
18	55.79	0.01222	0.9721	15.44	106.92	0.0341	0.2257	114.48	0.2410	122.33	0.2552
20	57.98	0.01225	0.9369	15.98	107.13	0.0352	0.2253	114.71	0.2406	122.59	0.2548
22	60.23	0.01229	0.9032	16.52	107.33	0.0364	0.2249	114.94	0.2402	122.84	0.2544
24	62.55	0.01232	0.8707	17.06	107.53	0.0375	0.2246	115.17	0.2398	123.10	0.2540
26	64.94	0.01236	0.8398	17.61	107.73	0.0379	0.2242	115.40	0.2395	123.35	0.2537
28	67.40	0.01239	0.8100	18.17	107.93	0.0398	0.2239	115.62	0.2391	123.60	0.2533
30	69.93	0.01243	0.7816	18.74	108.13	0.0409	0.2235	115.84	0.2387	123.85	0.2529
32	72.53	0.01247	0.7543	19.32	108.33	0.0421	0.2232	116.07	0.2383	124.10	0.2525
34	75.21	0.01250	0.7283	19.90	108.52	0.0433	0.2228	116.29	0.2380	124.35	0.2522
36	77.97	0.01254	0.7032	20.49	108.71	0.0445	0.2225	116.52	0.2376	124.59	0.2518
38	80.81	0.01258	0.6791	21.09	108.90	0.0457	0.2222	116.74	0.2373	124.84	0.2515
40	83.72	0.01262	0.6559	21.70	109.09	0.0469	0.2218	116.96	0.2369	125.08	0.2511
42	86.69	0.01266	0.6339	22.29	109.27	0.0481	0.2215	117.18	0.2366	125.32	0.2508
44	89.74	0.01270	0.6126	22.90	109.45	0.0493	0.2211	117.40	0.2363	125.56	0.2504
46	92.88	0.01274	0.5922	23.50	109.63	0.0505	0.2208	117.61	0.2359	125.80	0.2501
48	96.10	0.01278	0.5726	24.11	109.80	0.0516	0.2205	117.82	0.2356	126.04	0.2497
50	99.40	0.01282	0.5537	24.73	109.98	0.0528	0.2201	118.02	0.2353	126.27	0.2494
52	102.8	0.01286	0.5355	25.34	110.14	0.0540	0.2198	118.22	0.2350	126.50	0.2491
54	106.2	0.01290	0.5184	25.95	110.30	0.0552	0.2194	118.42	0.2347	126.73	0.2488
56	109.8	0.01294	0.5014	26.58	110.47	0.0564	0.2191	118.62	0.2343	126.96	0.2484
58	113.5	0.01299	0.4849	27.22	110.63	0.0576	0.2188	118.82	0.2340	127.19	0.2481
60	117.2	0.01303	0.4695	27.83	110.78	0.0588	0.2185	119.01	0.2337	127.42	0.2478
62	121.0	0.01307	0.4546	28.46	110.93	0.0600	0.2181	119.21	0.2334	127.65	0.2475
64	124.9	0.01312	0.4403	29.09	111.08	0.0612	0.2178	119.40	0.2331	127.87	0.2472
66	128.9	0.01316	0.4264	29.72	111.22	0.0624	0.2175	119.59	0.2327	128.10	0.2469
68	133.0	0.01320	0.4129	30.35	111.35	0.0636	0.2172	119.77	0.2324	128.32	0.2466
70	137.2	0.01325	0.4000	30.99	111.49	0.0648	0.2168	119.96	0.2321	128.54	0.2463
72	141.5	0.01330	0.3875	31.65	111.63	0.0661	0.2165	120.15	0.2318	128.76	0.2460
74	145.9	0.01334	0.3754	32.29	111.75	0.0673	0.2162	120.32	0.2315	128.97	0.2457
76	150.4	0.01339	0.3638	32.94	111.88	0.0684	0.2158	120.50	0.2312	129.19	0.2455
78	155.0	0.01344	0.3526	33.61	112.01	0.0696	0.2155	120.67	0.2309	129.40	0.2452
80	159.7	0.01349	0.3417	34.27	112.13	0.0708	0.2151	120.85	0.2306	129.61	0.2449
82	164.5	0.01353	0.3313	34.92	112.24	0.0720	0.2148	121.02	0.2303	129.82	0.2446
84	169.4	0.01358	0.3212	35.60	112.36	0.0732	0.2144	121.18	0.2300	130.02	0.2443
86	174.5	0.01363	0.3113	36.28	112.47	0.0744	0.2140	121.34	0.2297	130.23	0.2441
88	179.6	0.01368	0.3019	36.94	112.57	0.0756	0.2137	121.50	0.2294	130.43	0.2438
90	184.8	0.01374	0.2928	37.61	112.67	0.0768	0.2133	121.66	0.2291	130.63	0.2435
92	190.1	0.01379	0.2841	38.28	112.76	0.0780	0.2130	121.82	0.2288	130.83	0.2432
94	195.6	0.01384	0.2755	38.97	112.85	0.0792	0.2126	121.97	0.2285	131.03	0.2429
96	201.2	0.01390	0.2672	39.65	112.93	0.0803	0.2122	122.12	0.2282	131.23	0.2427
98	206.8	0.01396	0.2594	40.32	113.00	0.0815	0.2119	122.26	0.2279	131.42	0.2424
100	212.6	0.01402	0.2517	40.98	113.06	0.0827	0.2115	122.40	0.2276	131.61	0.2421
102	218.5	0.01408	0.2443	41.65	113.12	0.0839	0.2111	122.53	0.2273	131.80	0.2418
104	224.6	0.01414	0.2370	42.32	113.16	0.0851	0.2107	122.66	0.2270	131.99	0.2416
106	230.7	0.01420	0.2301	42.98	113.20	0.0862	0.2104	122.79	0.2267	132.17	0.2413
108	237.0	0.01426	0.2233	43.66	113.24	0.0874	0.2100	122.92	0.2264	132.35	0.2411
110	243.4	0.01433	0.2167	44.35	113.29	0.0886	0.2096	123.04	0.2261	132.53	0.2408
112	249.9	0.01440	0.2104	45.04	113.34	0.0898	0.2093	123.16	0.2258	132.71	0.2405
114	256.6	0.01447	0.2043	45.74	113.38	0.0909	0.2089	123.28	0.2255	132.88	0.2403
116	263.4	0.01454	0.1983	46.44	113.42	0.0921	0.2085	123.40	0.2253	133.05	0.2400
118	270.3	0.01461	0.1926	47.14	113.46	0.0933	0.2081	123.51	0.2250	133.22	0.2398
120	277.3	0.01469	0.1871	47.85	113.52	0.0945	0.2078	123.62	0.2247	133.39	0.2395

Data from Kinetic Chemicals, Inc., 1945

Table 3. Properties of Ammonia

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Enthalpy and Entropy Taken From −40 F							
				Enthalpy		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746.5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631.8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1.464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1.3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634.0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

of refrigerant which is present in vapor form. Consider, for example, F-12 with a quality (the per cent in vapor form) of 30 per cent; the enthalpy of this material would be equal to:

$$h_m = h_f + 0.30\,(h_v - h_f) \tag{4}$$

where

h_m = specific enthalpy of the mixture.
h_f = specific enthalpy of the liquid.
h_v = specific enthalpy of the saturated vapor.

Values of h_f and h_v are obtained from Table 1 for the actual pressure of the mixture.

By a reversal of this same procedure the tabular data can be used to determine the state of a mixture leaving an expansion valve. Consider a valve to which saturated liquid at pressure p_e is admitted and a mixture of saturated liquid and vapor at pressure p_d is discharged. The quality of the material at discharge is then determined by making use of the fact that the expansion process is completely irreversible, is a throttling process, and hence occurs without change in enthalpy. Thus the enthalpy of the mixture, h_m, is equal to the enthalpy of the saturated liquid at the entrance state, h_{fe}, and can therefore be read from the table.
Thus,

$$h_{fe} = h_m = h_{vd} - (1 - x)\;(h_{vd} - h_{fd}) \tag{5}$$

or,

$$x = (h_m - h_{fd}) + (h_{vd} - h_{fd}) \tag{6}$$

where

h_{fe} = enthalpy of saturated liquid at entrance to expansion valve.
h_m = enthalpy of mixture.
h_{vd} = enthalpy of saturated vapor at discharge.
h_{fd} = enthalpy of liquid at discharge.
x = proportion of liquid in the mixture, decimal.

Vapor Compression Refrigeration Cycle

Simple Cycle. The refrigerant *cycle* is the series of state changes which occur in the conditioning processes needed to restore the refrigerant to a condition in which it will possess the ability to extract heat from the space to be cooled. For all compression-type systems the cycle consists of four processes: heat gain in the evaporator; pressure rise in the compressor; heat loss in the condenser; pressure loss in the expansion valve. The compression process is accomplished at the expense of energy added to the compressor in the form of shaft work and the expansion process could be carried out, if the economics of the system would permit, in an expanding engine with consequent release of energy as shaft work. In ordinary systems, however, the additional first cost and maintenance costs of an expanding engine so greatly exceed the advantage resulting from the work realized that such engines are not used and the pressure reduction is allowed to occur irreversibly in an expansion valve. Basically, then, a refrigeration cycle consists of two heat transfer processes and two pressure change processes, no work entering into the heat transfer processes and—in the simple cycle—no heat transfer occurring during the pressure-change processes.

The most common and least complicated type of refrigeration cycle is called the simple saturation cycle and is shown diagrammatically in Fig. 3 and plotted upon pressure-enthalpy coordinates in Fig. 4. For this system saturated vapor flows without gain or loss of heat from the evaporator to the suction of the compressor. During passage through the compressor the energy added as shaft work goes entirely to increase the enthalpy of the refrigerant, and the compression process, which is assumed to occur irreversibly and without external heat transfer, is characterized by constant entropy. Thus the state of the superheated vapor leaving the compressor can be determined from the tables of thermodynamic properties by noting the discharge pressure and fixing also the entropy of the saturated vapor at entrance to the compressor.

TABLE 4. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F-11)

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Enthalpy and Entropy Taken From −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid*	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

Superheated vapor from the compressor flows to the condenser where de-superheating and condensation take place. From the condenser the refrigerant flows to the expansion valve, undergoes a constant-enthalpy pressure reduction and returns to the evaporator where it again removes a quantity of undesired heat. When the evaporator is arranged to permit direct cooling of room air by the refrigerant, the system is said to be of the *direct expansion* type, while a system in which the evaporating refrigerant cools water or brine, which in turn cools the air, is said to be *indirect*. Although many differences exist between most actual systems and that of the simple saturation cycle this latter is nonetheless of great value in that it provides an extremely simple method of rapidly achieving an approximate analysis of probable power requirements, compressor size, etc. Further, the equations used in analysis of a simple saturation cycle form the basis

of the more complex treatments required for compound refrigeration cycles. For these reasons a typical simple saturation problem will be worked in detail.

Example 1. A simple saturation cycle carries a 7 ton load when operating between suction and discharge pressures of 52.7 psia and 121 psia with F-12 as the refrigerant. Determine: (*a*) the cooling effect provided by each pound of refrigerant, (*b*) the refrigerant circulating rate, (*c*) the horsepower required, (*d*) the quantity of heat to be dissipated from the condenser, (*e*) the required condenser cooling water, in gallons per minute, if temperature rise of water passing through the condenser is 8 deg, (*f*) the bore and stroke of a double acting cylinder (neglecting the effect of the piston rod) if speed of compressor is 500 revolutions per minute, (g) coefficient of performance.

Solution. (*a*) Saturated liquid F-12 at 121 psia leaves the condenser and enters the expansion valve. The enthalpy of this material (from Table 1) is 29.68 Btu per pound and this must also be its enthalpy at entrance to the evaporator. Leaving the evaporator as a saturated vapor at 52.7 psia, its enthalpy is 82.82 so the refrigerating effect must be 82.82 − 29.68 = 53.14 Btu per pound.

(*b*) The refrigerant circulating rate is equal to the total heat to be picked up in unit time divided by the pick-up per pound of refrigerant or,

$$W_r = (7 \text{ ton} \times 200) \div 53.14 = 26.3 \text{ lb per minute.}$$

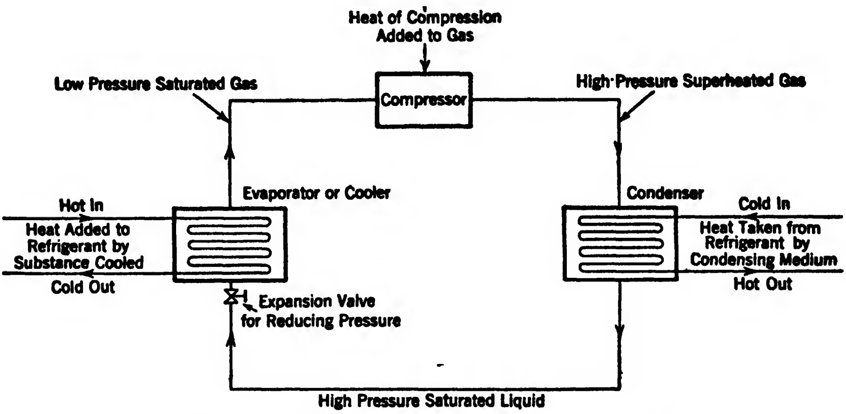


FIG. 3. MECHANICAL REFRIGERATION SYSTEM

(*c*) The horse power required is equal to the increase in energy of the refrigerant passing through the compressor (expressed in Btu per minute) divided by the conversion factor 42.42, which is the number of Btu per minute corresponding to 1 hp,

$$(\text{hp}) = W_r\,(h_d - h_{vs}) \div 42.42 \qquad (7)$$

where

hp = horsepower.
W_r = refrigerant circulating rate in pounds per minute.
h_d = enthalpy of vapor at condition of discharge from compressor.
h_{vs} = enthalpy of saturated vapor entering compressor.

W_r is known from (*b*) and h_{vs} is the enthalpy of refrigerant as it enters the compressor in a saturated vapor state at 52.7 psia; thus h_{vs} = 82.82.

In order to determine h_d, the state of the refrigerant must first be determined at the compressor discharge. At the known suction state the entropy (from Table 1 for saturated vapor at 52.7 psia) is 0.16828 and, since the compression is assumed to occur isentropically, it therefore follows that the discharge state must have the same entropy at 121 psia. From the table the entropy of vapor superheated 25 deg is

0.17330, so the superheat, t_{sd}, possessed by the actual gas discharged from this compressor can be obtained by interpolation as,

$$\frac{t_{sd}}{25} = \frac{0.16828 - 0.16608}{0.17330 - 0.16608}$$

from which $t_{sd} = 7.6$ deg.

As the saturation temperature at 121 psia is 94 F the actual temperature, t_d, of the vapor leaving the compressor is, $t_d = 94 + t_{sd} = 94 + 7.6 = 101.6$ F. By the same kind of interpolation the enthalpy of the discharged vapor can be determined from the enthalpies given for vapor superheated 25 F and for saturated vapor,

$$\frac{(h_d - 88.10)}{(92.16 - 88.10)} \qquad \frac{(0.16828 - 0.16608)}{(0.17330 - 0.16608)}$$

from which, $h_d = 89.34$ Btu per pound.

Then substituting in Equation 7,

$$(\text{hp}) = 26.3\ (89.34 - 82.82) \qquad 42.42 = 4.03$$

(*d*) The rate of heat loss from the condenser, Q_c, must be equal to the sum of the energies picked up by the refrigerant in the evaporator and the compressor,

$Q_c = 53.14 + (89.34 - 82.82) = 53.14 + 6.52 = 59.66$ Btu per pound or $26.3 \times 59.66 = 1569$ Btu per minute. This same figure can, of course, be determined more

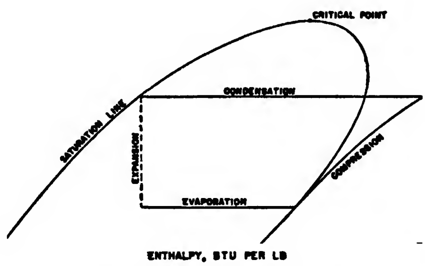


Fig. 4. Pressure-Enthalpy Diagram for Simple Saturation Cycle

directly by subtraction of the enthalpy of liquid leaving the condenser from the enthalpy of superheated vapor going into it, thus,

$$Q_c = 26.3\ (89.34 - 29.68) = 1569 \text{ Btu per minute.}$$

(*e*) The cooling water rate (based on a gallon as 8.34 lb) is $1569 \div (8 \times 8.34) = 23.5$ gpm.

(*f*) The compressor size is fixed by the volume of gas which must be drawn into the machine per unit time. Saturated vapor at 52.7 psia has a specific volume, from Table 1, of 0.779 cu ft per pound, hence $26.3 \times 0.779 = 20.49$ cfm of gas must be handled. Assuming a volumetric efficiency of 90 per cent the compressor must then displace $20.49 \div 0.9 = 22.8$ cfm. The speed is given as 500 rpm and as the unit is known to be double-acting the displacement is therefore $(22.8 \times 1728) \div (2 \times 500) = 39.4$ cu in. If the unit were designed so that bore, d, and stroke were the same,

$$(\pi d^3) \div 4 = 39.4 \qquad\qquad d = 3.69 \text{ in.}$$

$$(\text{CP}) = \quad {}_{s} - h_{te}) \div (h_d - h$$

$$= (82.82 - 29.68) \div (89.34 - 82.82) = 8.17$$

where h_{te} is the specific enthalpy of liquid at discharge from the condenser.

The coefficient of performance of *Example* 1 may be compared with that of an ideal system operating on the Carnot cycle between the same tempera-

ture limits. Then T_e = 501 F (which is 41 F + 460) and T_c = 554 F (which is 94 F + 460) and,

$$554 - 501 = 9.6$$

The actual cycle is therefore 8.17 ÷ 9.6 or 85 per cent as effective as a Carnot cycle between the same temperature limits.

Influence of Suction Pressure

Brief consideration of the analytical procedure used in discussion of the simple saturation cycle will bring out the need for maintaining the suction pressure on any refrigeration system as high as the load will permit. As the suction pressure increases, for fixed discharge pressure, the enthalpy of refrigerant entering the evaporator remains unchanged, but the leaving enthalpy increases and hence the refrigerating effect increases. Further, compressor energy input is reduced not merely because of the greater enthalpy of the gas at suction, but also because of a reduction in the enthalpy of the superheated gas at discharge. Since the refrigerating effect is greater and the work less, it is obvious that there will be a substantial gain in the coefficient of the performance.

The actual value of suction pressure on any system is obviously determined by the required temperature which must be maintained in the conditioned space. For a direct expansion system the evaporator can be held at a temperature not much less than that of the conditioned enclosure except in cases where lower temperatures may be needed in order to establish a desired ratio of dehumidifying to cooling load. When dehumidification requirements dictate the use of unusually low evaporator temperatures the increased operating cost should properly be charged against the dehumidification rather than the sensible cooling.

Influence of Discharge Pressure

In contrast to the suction pressure, the compressor discharge pressure should be kept as low as operating conditions will allow. This pressure must be high enough to provide a saturation temperature of refrigerant within the condenser which is greater than the exit temperature of the cooling water. The discharge pressure therefore is a direct function of the temperature of the cooling fluid and will automatically rise whenever the temperature of cooling water (or air) rises; it will also rise when the flow rate of the cooling medium is decreased.

Increase in discharge pressure (for fixed suction pressure) raises the enthalpy of the gas leaving the compressor, hence increases the work of compression. Further, the enthalpy of saturated liquid leaving the condenser increases with pressure so the refrigerating effect must decrease. Thus the effect of such a pressure rise is to require more work per pound of refrigerant handled and at the same time to necessitate an increase in the refrigerant flow rate.

Influence of Water Jacket

The preceding discussion has, in every case, assumed isentropic compression. Where exact performance data are not available this assumption is a desirable one since it leads to a conservatively large determination of the power required. In most actual systems the compression process departs from isentropic due to irreversible heat transfers which occur

between the vapor in the cylinder and the cylinder wall and also because of intentional heat dissipation from the outside of the cylinder walls to the surroundings, or to a cooling fluid passing through a water jacket around the cylinder. Compressor cooling is highly desirable as a method of reducing power consumption.

Influence of Superheating and Subcooling

The most common departure from conditions of the simple saturation cycle is that resulting from admission of superheated vapor to the compressor. Thermodynamically, superheat is undesirable since the enthalpy increase required to compress a vapor through a given pressure range increases with superheat. Further, superheated vapor leaving an evaporator is usually an indication that the suction pressure is lower than necessary. Under practical operating conditions, however, superheat is almost universally used as a means of assuring complete vaporization of the refrigerant going to the compressor. With modern compressors operating at high speed and with relatively small clearance space it is

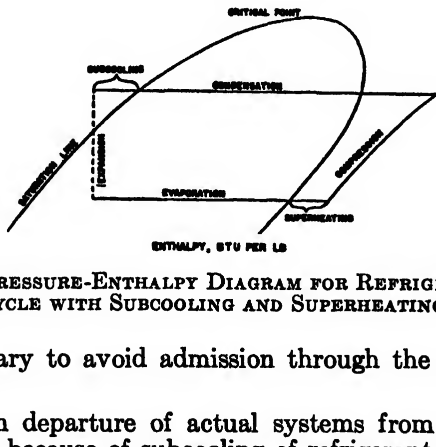


FIG. 5. PRESSURE-ENTHALPY DIAGRAM FOR REFRIGERATION CYCLE WITH SUBCOOLING AND SUPERHEATING

particularly necessary to avoid admission through the suction valves of liquid refrigerant.

Another common departure of actual systems from the simple saturation cycle occurs because of subcooling of refrigerant in the condenser. Thermodynamically such subcooling is advantageous since it increases the refrigerating effect without affecting the unit energy requirements of the compressor. Further, it can be shown that for a fixed ratio of condenser cooling water to refrigerant circulating rate the total compressor power requirements will be greater when operating simple saturation than when operating with maximum sub-cooling. What is even more surprising is that condenser pressure may be lower for the sub-cooling cycle than for the saturation cycle; this condition results from the fact that, for the same capacity on a heavily loaded condenser, the refrigerant flow rate is less when there is sub-cooling.

Because of the advantages attendant upon the use of sub-cooling, many methods are in use for obtaining some sub-cooling effect outside of the condenser. One common procedure is to use the cold vapor leaving the evaporator to cool the liquid flowing from condenser to expansion valve.

Another somewhat unusual subcooling cycle allows cold refrigerant from the downstream side of the expansion valve to cool liquid refrigerant from the condenser down to the evaporator temperature. Fig. 5 shows the pressure-enthalpy diagram for a typical refrigeration cycle operating with

both sub-cooling of the refrigerant from the condenser and superheating of the refrigerant leaving the evaporator.

Clearance and Volumetric Efficiency

Clearance, like displacement, is a characteristic—usually fixed—of a given compressor. In some cases clearance pockets are provided which place within the operator's control the ability to alter the clearance of the machine, but most moderate size compressors are built with fixed clearance. By definition the clearance is the percentage of the volume swept by the piston which is represented by spaces in the end of the cylinder (including valve spaces, etc.) when the piston is at the end of its stroke.

Because of the trapping of high pressure vapor in the clearance space, and its subsequent re-expansion, the suction valves of the compressor do not open until the piston has completed part of its stroke. Hence the volume of fresh vapor introduced into the compressor per stroke is less than the volume swept by the piston. The ratio of actual volume of fresh gas to swept volume is, by definition, the clearance volumetric efficiency, CVE. In equation form,

$$(\mathrm{CVE}) = 100 - V_c\left[\frac{v_s}{v_d} - 1\right] \tag{9}$$

where

CVE = clearance volumetric efficiency.
V_c = clearance, per cent of volume swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.).
v_s = specific volume of gas at compressor inlet.
v_d = specific volume of gas at compressor discharge.

Values of v_s and v_d can be obtained directly or by calculation from the tables of properties of refrigerants.

In addition to clearance, there are several other factors which tend to reduce the volumetric efficiency. The suction gases from the evaporator are heated and expanded upon contact with the hot cylinder walls during the suction stroke. This results in a reduction of the actual charge drawn into the cylinder. Wire-drawing through the suction and discharge valves reduces the suction pressure in the cylinder below that in the evaporator and increases the discharge pressure above that in the condenser. Leakage of gases around the pistons also decreases the volumetric efficiency. The total volumetric efficiency (TVE) includes all of these factors and is reliably obtained only by laboratory measurements. It is too difficult to predict the effects of these factors to any degree of accuracy comparable to actual tests.

Complex Refrigeration Cycles

The preceding sections have dealt only with refrigeration systems in which there is but one evaporator, compression is accomplished through but a single stage, and expansion proceeds through a single expansion valve. In large systems or in low temperature systems in which the compression ratio is high, the compression process can be carried out in stages with the refrigerant passing through several cylinders arranged for operation in series. The thermodynamic advantage of such compound compression arises from the fact that intercoolers can be placed between the stages of

compression to extract heat from the vapor and thereby cause the over-all compression process to approach more closely the ideal condition of isothermal compression. Essentially such intercoolers serve the same purpose as a cooling jacket but with greater effectiveness because of the more satisfactory heat transfer conditions.

In the simple saturation cycle the saturated liquid entering the expansion valve commences to vaporize as soon as its pressure starts to drop. The vapor produced during the expansion process has no further use, in terms of refrigerating effect, since it has already picked up its latent heat of vaporization as a result of heat which it has extracted from the unvaporized residue. Thus the instant such vapor forms, its usefulness is at an end, and to allow such material to undergo a further drop in pressure is uneconomical. With compound compression, there is at least one intermediate pressure at which flash vapor can be extracted. In such cases several expansion valves can be utilized with all of the refrigerant from the condenser passed through a first expansion valve to the higher suction pressure and the flash vapor then extracted and returned to the condenser through the high compression stage. The remaining refrigerant can then pass through a second expansion valve where the pressure is dropped to that corresponding to the low-pressure evaporator. The number of expansion valves is limited by the number of stages of compression.

Further cycle complications may arise if more than one evaporator is to be operated with a single compressor and particularly if the pressures in these evaporators are to differ. The most common solution is to operate the compressor at the suction pressure of the lowest pressure evaporator and to equip all other evaporators with back-pressure regulating valves or throttling devices between the evaporator and the compressor suction. This permits these evaporators to operate at higher pressures and, therefore, higher temperatures than those corresponding to the compressor suction conditions. However, this is accomplished only with a loss of power, since all of the refrigerant from all of the evaporators must be compressed through the maximum lift from the lowest pressure in the system.

The Air Cycle System

Air cycle refrigeration, one of the earliest forms of cooling, became obsolete for many years because of its low coefficient of performance and high operating costs. Recently, however, it has been applied with success to aircraft cooling systems where, with low equipment weight, it can utilize a portion of the cabin air supercharger capacity. It is unique among refrigeration systems in that the refrigerant remains in the gaseous phase throughout the cycle.

Fundamentally, the air cycle is essentially the same as the vapor cycle. Compression is accomplished by a reciprocating or centrifugal compressor, and, since there is no change of phase of the refrigerant upon expansion, an *air cooler* replaces the condenser and a *refrigerator* the evaporator. Although some cooling would result from the expansion of the gas through an ordinary expansion valve, a much greater drop in air temperature is accomplished if the expansion is controlled to approach the isentropic by replacing the valve with an expansion engine or turbine. Furthermore, the work recovered by such an expansion engine can be utilized to supply part of the work of compression or to drive other devices.

It is a common misconception that aircraft flown at high altitudes do not require comfort cooling. With pressurized cabins the work of com-

pression results in an air temperature increase which, when added to the heat supplied by solar radiation, electrical and mechanical equipment and the occupants of the plane, may make the conditions intolerable without comfort cooling. Several different refrigeration systems have been applied to aircraft conditioning, but by far the most successful system to date is the air cycle system. For example, in one commercial plane the weight requirements were reduced from approximately 60 pounds per ton for a vapor compression system or an initial weight of 130 pounds per ton for a dry ice system to approximately 25 pounds per ton for an air cycle system. For further details on the air cycle system, see bibliography references A and H.

The Steam Jet System

The *steam jet* system under certain circumstances is desirable for use in air conditioning[1]. Steam supplies directly the power used for com-

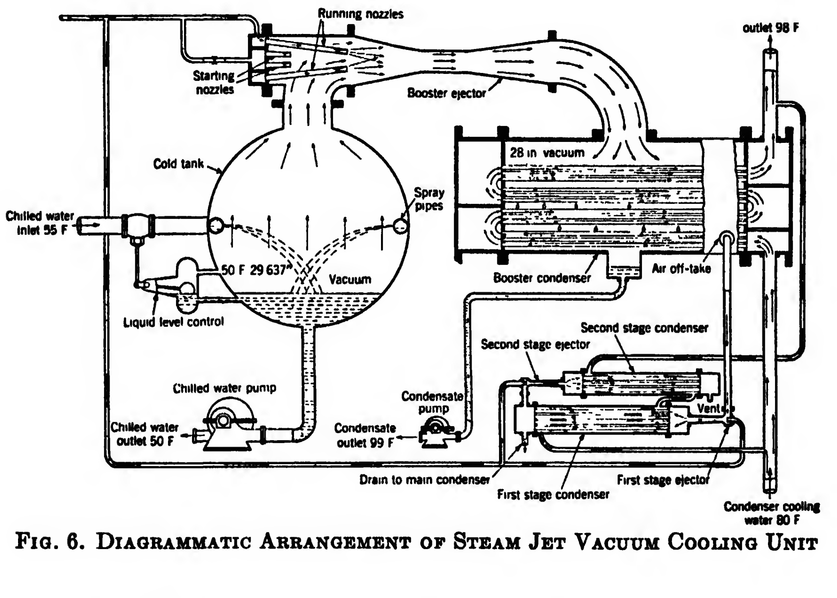


FIG. 6. DIAGRAMMATIC ARRANGEMENT OF STEAM JET VACUUM COOLING UNIT

pressing the refrigerant, thus eliminating the losses connected with other methods of supplying energy. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiency of the equipment is somewhat lower than that of the positive mechanical type compressor. The condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures. Steam jet boosters or compressors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 6. The figures correspond to an average repre-

sentative system. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam. As this requires heat, and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken from the circulated water, to a somewhat higher absolute pressure and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture then passes from the ejector into the condenser.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A secondary condenser is then necessary to condense the steam in the secondary jet.

While a single booster of smaller than 15 tons capacity is difficult to build, steam jet vacuum cooling units have been built for as small as 5 to 6 tons capacity. They can readily be built for steam pressures of from 5 to 200 lb per square inch and condenser water temperatures as high as 90 F. The steam consumption in pounds per hour per ton of refrigeration increases rapidly as the booster steam pressure is lowered. For example, the lowering of the booster steam pressure from 200 to 90 lb per square inch results in an increase in steam consumption of approximately 5 per cent, whereas a further decrease in booster steam pressure to 10 lb per square inch increases the steam consumption by approximately 72 per cent over that required at 200 lb per square inch.

The capacity of a steam jet system is usually controlled by controlling the number of boosters in use since the unit usually has several boosters operating on the same evaporator. Usually one booster is automatically controlled whereas the others are manually operated. The capacity is dependent, as for all compressors, upon the evaporator temperature, or in other words, the suction pressure. For example, the capacity is lowered approximately 17 per cent if the evaporator or chilled water temperature is lowered from 50 to 45 F. The capacity therefore can be controlled to some extent by regulating the evaporator temperature.

The Absorption System

The absorption and compression refrigeration cycles differ only with respect to the method of compression. Each cycle requires a condenser, expansion valve, and evaporator, but the absorption cycle utilizes three major equipments in place of the mechanical compressor; these equipments are the *absorber*, the *pump*, and the *generator*. Vapor from the evaporator is absorbed by a low temperature absorbent fluid which is then pumped to the generator where heat is supplied to boil off the refrigerant. The absorbent is now cooled and readmitted, through a pressure-reducing valve, to the absorber.

In addition to the three primary equipments of the absorption cycle it

is necessary to provide auxiliary equipment, usually an analyzer and a rectifier, to remove from the refrigerant leaving the generator, insofar as is possible, the absorbent which vaporizes and leaves the generator with the refrigerant. Removal of this material is of great importance to effective operation of the system, since even a small concentration of absorbent in the refrigerant will suffice to reduce greatly the evaporator pressure required for maintenance of a given evaporator temperature. Thermodynamic analysis of absorption cycles is relatively complex and requires the use either of tables or graphs showing the equilibrium relationships and thermodynamic properties of the refrigerant-absorbent combination. Data of this kind are available in bibliography item A and a discussion of various absorbents is given in bibliography item F. Thermodynamically the effectiveness of a refrigerant-absorbent combination increases directly with its negative deviation from Raoult's Law.

Fig. 7 shows a typical absorption cycle flow diagram. Cooling water first goes through the absorber (where it extracts the heat of absorption

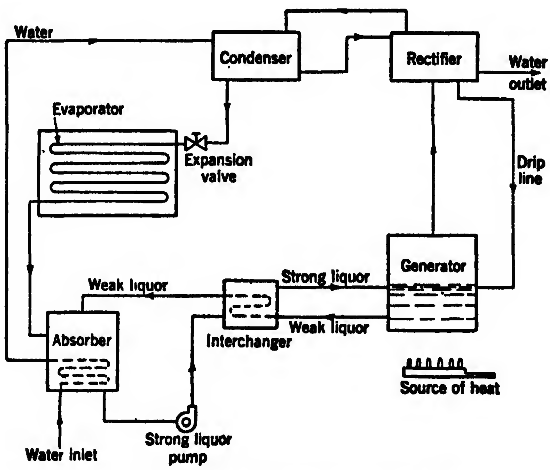


FIG. 7. CLOSED ABSORPTION SYSTEM

which is liberated by the refrigerant vapor as it goes into solution), then through the condenser, and finally through the rectifier. Refrigerant from the evaporator enters the absorber where it goes into solution in the absorbent; the high concentration solution is then pumped to the generator where heat is supplied; the refrigerant (with some absorbent vapor) leaves for the rectifier and the warm low concentration solution is returned to the absorber. In the rectifier selective condensation occurs, the concentration of the absorbent in the condensate being much greater than its concentration in the entering vapor mixture; rectifier condensate is dripped back to the generator.

The total energy requirements of an absorption cycle greatly exceed those of a compression system, but the energy required is of low availability (heat) in contrast with the high availability requirements (shaft work) of the mechanical compressor. Thus in localities where heat and cooling water are obtainable at low cost, it will be more economical to use a large quantity of inexpensive thermal energy in preference to a much smaller quantity of expensive shaft energy. For most absorption systems the

heat required will be from one and one-half to five times as much as the heat extracted in the evaporator; cooling water requirements are proportionally high.

Ice Systems

Cold water systems using ice as the cooling agent have been installed in many theaters, restaurants, funeral homes, churches and other places where short hours of operation and high peaks of cooling demand make this type of system desirable. A comparatively small quantity of ice in the water cooling tank of such a system can release refrigeration at a relatively rapid rate. For instance, neighborhood theaters having a peak demand of 1,200,000 Btu per hour (100 tons refrigeration) have found 8 ton capacity ice bunkers satisfactory.

In operation, the water in the air conditioning system is circulated over ice placed in an insulated box and is cooled to the 38 or 40 F range or higher

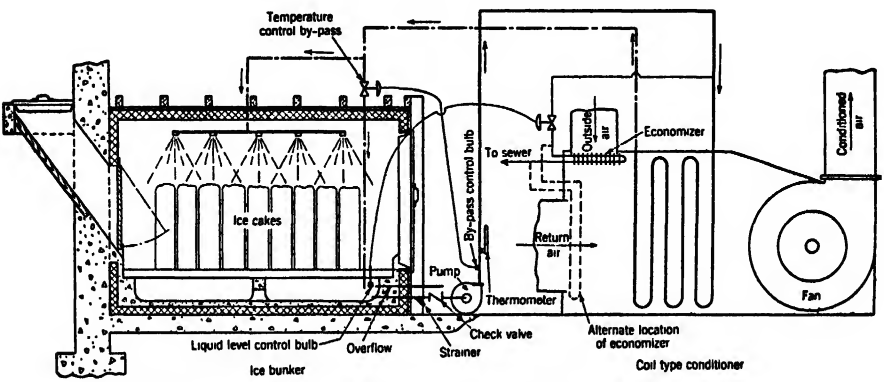


FIG. 8. TYPICAL ICE SYSTEM SHOWING BUNKER DETAILS

if desired. This cold water is pumped from the ice bunker to air cooling coils or spray type air washers. The blowers, coils, air washer or air handling sections are the same as those parts in any system employing cold water as a refrigerant.

The ice water cooler or ice bunker is usually built at the installation in a location where it can easily be iced. It can be constructed of any desired material such as concrete, steel, or wood with an adequate amount of insulation to save the ice from one period of use to the next. The basic requirement is that the tank be durable and water tight. A typical bunker with connections to a coil type air conditioning system is shown in Fig. 8. About 60 cu ft of gross bunker volume are allowed per ton of ice capacity.

The shape of the bunker usually conforms to the available space. The one illustrated has overhead sprays, but if head-room is lacking the ice is placed on the floor of the bunker with the water returned around the lower part of the blocks from a perforated distribution pipe run along one side of the bunker. To secure good circulation the supply water is extracted from a similar perforated pipe on the opposite side of the bunker.

The temperature of the water is controlled at a predetermined point by a thermostat in the supply line. If the temperature drops too low, a part of the return water is by-passed directly to the sump and is not cooled over the ice. In the larger systems it is customary to install an overflow

control which, as the ice melts, discards the excess water through an economizer coil, the surface of which is large in relation to the flow so that the water is warmed to 60 F or more as it is discharged from the system.

In an attempt to lower initial equipment cost and operating expense, or increase the refrigeration capacity of an existing air conditioning system, storage refrigeration has been utilized in a few applications. Some of the methods which have been adopted include the storage of refrigeration in the form of chilled water, chilled brine, ice on evaporator coils[2] and the accumulation of thin sheets of ice on copper plates in a steel tank[3]. If the peak load factor is low as compared with a long period of operation, such as in a restaurant, or if the hours of operation are short but the usage factor high as in a church, then it is possible to consider storage refrigeration. This method of accumulating refrigeration frequently makes it possible to use low cost off-peak electric power. Power costs may also be reduced by installing a smaller refrigeration plant, augmented by a storage system, and by operating it for longer periods.

The Heat Pump

Although frequently referred to incorrectly as the reverse cycle system, the heat pump cycle is identical with the ordinary refrigeration cycle and differs only in the sense that the desired effect is rejection of the heat from the condenser rather than absorption of heat in the evaporator. A discussion of the coefficient of performance for the heat pump is found earlier in this chapter.

From an analysis of the equation for the coefficient of performance, it is evident that the economical adaption of the heat pump as a practical means of heating requires that the temperature of the source from which the heat is extracted be as high as possible and that the temperature of the sink to which the heat is rejected for heating purposes be as low as possible. Thus, with a small temperature spread between the evaporator and the condenser, six or more times as much heat may be obtained theoretically and three to five times practically as the heat equivalent of the work necessary to operate the system. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

The source of heat supply for the evaporator may be air, ground water or the earth itself depending upon the climate and topography:

1. Air may be used, but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is the lowest thus resulting in the least favorable temperature combination. Practical considerations seem to limit the use of present air systems to climates such as those encountered in the southern United States where temperatures under 20 F are not experienced.

2. Water from wells, lakes or rivers may be used. Well water is the most desirable since its temperature is fairly constant throughout the entire year. As water temperature is relatively high, even in winter, a large amount of heat may be removed relative to the weight of water handled. Means of returning the water to the underground reservoir should be provided to prevent depletion. The disadvantages of using water include the problem of locating an adequate supply, the cost of pumping and the problem of water disposal.

3. The earth may be used as a source of heat with the refrigerant coils buried in the ground or with a heat exchanger supplied by a water-circulating coil buried in the ground. One disadvantage appears to be the large amount of heat transfer surface required.

Some of the other factors which act as limitations are: the large temperature spread when using air as a source of heat and when attempting

to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring extra equipment for a complete heating load, and the relatively high initial cost of equipment as compared to that at present available for heating by conventional means.

Both water and air are practical media to which the condenser heat may be rejected, but the generation of steam requires too high a temperature. Practical operation therefore dictates that the heat pump be used in conjunction with either an air or water heating system.

Since development of a heat pump to date includes the use of air, water, and earth as heat sources and air and water as heat sinks, there are six possible combinations of source and sink in application: air to air, air to water, water to air, water to water, earth to air and earth to water. A typical arrangement of a heat pump system with air as the source of heat is shown in Fig. 9. If the air seldom drops below freezing, heating is often required in the morning and cooling during the afternoon in order to maintain comfortable conditions in such a system. The arrangement as shown lends itself to changing over automatically as required.

BASIC REFRIGERATION EQUIPMENT

Compression Refrigeration Machines

Compression of the refrigerant gas drawn from the evaporator may be accomplished by one of several means. Positive displacement may be used as in the reciprocating, rotary or gear types of compressors; centrifugal force may be applied as in the centrifugal compressor; an ejector may be used as in the steam jet refrigeration cycle; or absorption of a low pressure refrigerant gas in a secondary fluid, followed by the absorbent's release upon application of heat, may be utilized. A detailed discussion of the equipment required for each of these types of systems is beyond the scope of this chapter. For a more comprehensive treatment, reference may be made to the bibliography. The present discussion is limited to positive displacement reciprocating compressors, rotary compressors and centrifugal compressors.

Reciprocating Compressors

Reciprocating compressors may be classified according to (a) cylinder design, (b) compressor drive, (c) valves, and (d) lubrication and cooling.

Cylinder Design. Cylinder design may vary as to number, arrangement, and action (i.e. single-acting or double-acting). Single-acting compressors usually have their cylinders arranged vertically, radially, or in a V or W shaped arrangement. Double-acting compressors with refrigerant gas drawn in and compressed on both the head and crank ends of the cylinder are usually arranged horizontally. Reciprocating units are available with from one to sixteen cylinders with the V, W, or radial arrangements best adapted to the greatest numbers. The present trend is toward higher operating speeds with a low displacement per cylinder together with an increase in the number of cylinders. Whereas the original reciprocating compressors were slow speed (50 to 55 rpm) steam driven devices, modern electric motor driven compressors range up to 3500 rpm. Cylinder heads are usually bolted tight to the cylinders, but in some large compressors where there is danger of wet compression or of foreign materials entering the compressor space, a secondary head known as a safety head may be seated at the end of the cylinder and held in position with heavy springs. Normally this head remains stationary, but excessive pressures in the clearance space are relieved by movement of the safety head and thus prevent damage to the cylinder.

Compressor Drives. Reciprocating compressors may be subdivided as to the source of motive power and as to whether they are open or hermetic. Practically all

modern compressors are electric motor driven although a few large, steam-driven compressors are still being installed where steam forms the most economical source of energy. In a few cases, as with truck transportation, the compressor may be internal combustion engine driven.

The division of compressors into open or closed types is dependent upon whether the motive power is received from an external source or whether the motor is direct drive and sealed within the housing. In the open type, power is received from an external source with one end of the compressor crankshaft extending through the crankcase and usually V-belt driven. The point of emergence of the shaft from the crankcase forms a weak point of refrigerant leakage and is most frequently sealed with a bellows type crankshaft seal. Horizontal double-acting compressors operate with a sliding piston rod, moving back and forth through a stuffing box. If the motor is direct drive and enclosed within the compressor housing, the compressor is classified as closed or hermetic. This eliminates the necessity of any shaft seal and not only prevents refrigerant leakage at this point but reduces operating noise. One disadvantage is the inaccessibility of moving parts for repairs, but lubrication is greatly simplified since both the motor and compressor operate in a sealed space with the lubricating oil.

Compressor Valves. All refrigeration compressor valves are dependent for their operation upon a difference in pressure between the inside of the cylinder and the

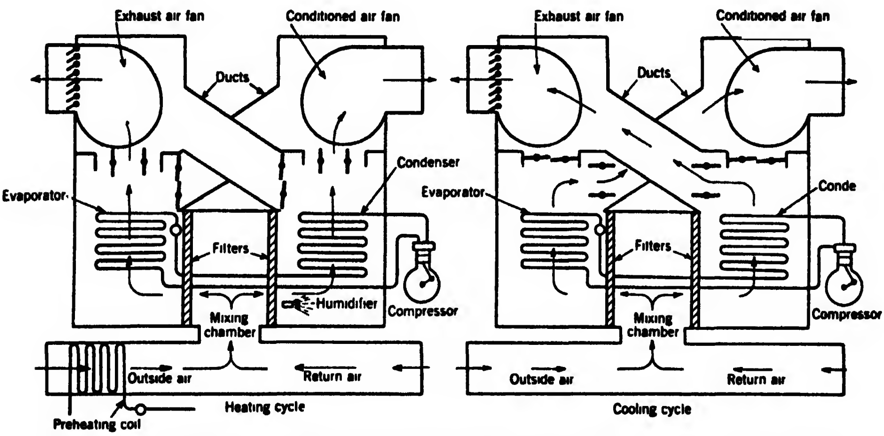


FIG. 9. SCHEMATIC OPERATION OF REVERSED CYCLE CONDITIONING SYSTEM

suction or discharge line. Although mechanically operated valves might have some advantage, they have proved unsatisfactory because each change in the evaporator or condenser operating pressures requires a change of valve setting. The pressure differentials required for operation of the valves depend upon the valve design and the compressor speed. The suction and discharge valves may be arranged with both located in the compressor head or with the suction valve on the top of the piston and the discharge valve in the compressor head (uniflow arrangement). The valves themselves are usually classified as either poppet, ring-plate or flexing.

Lubrication and Cooling. Lubrication of modern compressors is accomplished by either splash lubrication or forced lubrication. The latter is used on large compressors, while simple splash lubrication is used in the smaller units.

Large compressors are usually water cooled with the water jacket either cooling the cylinder walls or both the cylinder walls and the compressor head. Small compressors are either water cooled or air cooled with extended finned surfaces cast on the exterior of the cylinder. In a few cases small compressors may be found in which there is no attempt to add any purposive cooling other than through non-finned surfaces to the lower temperature air. Water cooling is more effective than air cooling, but even under the best conditions cylinder cooling removes only a portion of the superheat in the refrigerant gas. This removal of heat from the cylinder results in some decrease in the work of compression as well as reduction in condenser load.

Rotary Compressors

In recent years rotary compressors, usually hermetically sealed, have become quite popular for fractional tonnage applications and are being designed in increasingly larger sizes. Of the various designs attempted, the single blade rotary compressor shown diagrammatically in Fig. 10 is the most popular. An eccentric driven rotor revolves within a housing in which the suction and discharge passages are separated by means of a sealing blade. When the rotating eccentric first passes this blade and the suction opening, the compressor suction space is very small. As the eccentric rotates, this crescent-shaped space becomes increasingly larger thereby drawing in a charge of suction gas. When the eccentric again passes the blade, the gas charge is cut off from the suction inlet, compressed, and discharged from the compressor. Such rotary compressors are quiet in operation and reasonably free from vibration. In common with other types of hermetically sealed units, they have the advantages of comparatively low loss of refrigerant and sealed-in lubrication.

Centrifugal Compressors

Centrifugal compressors are used with very low pressure refrigerants;

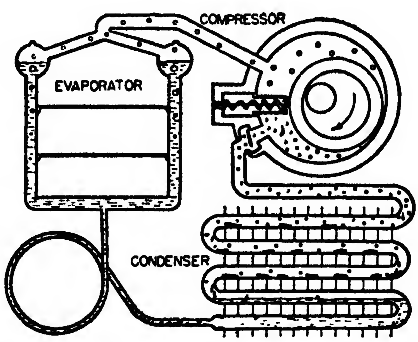


Fig. 10. Diagrammatic View of Rotary Compressor with Flooded Evaporator and Capillary Tube

usually both evaporator and condenser work below atmospheric pressure. Water and monofluorotrichloromethane (F-11) are the refrigerants commonly used in centrifugal machines.

Compression of the refrigerant is accomplished by means of centrifugal force; therefore, this type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are usually required and high speeds are necessary to obtain good efficiency.

The evaporator is usually constructed as an integral part of the centrifugal type condensing unit, to chill water which is then circulated to the air conditioning system. This is done because it would not be economical to pipe these large volumes of refrigerant any distance.

Centrifugal compressors like reciprocating compressors can be divided into two general types, open and enclosed. In general, the open type compressor is geared to the driving mechanism, and operates at higher speed than the driving motor or turbine. A modern, completely enclosed, direct-driven centrifugal compressor is illustrated in Fig. 11.

Centrifugal refrigeration compressors are particularly well suited to direct steam turbine drive because of their high operating speed. Water cooling equipment of one design is operated between 3500 and 4000 rpm for units developing 1000 to 2000 tons capacity and from 7000 to 8000 rpm for units developing 100 to 200 tons capacity. However, a great many applications, particularly in the smaller sizes, are electric motor driven and equipped with standard gear-type speed increasers. Centrifugal systems are particularly well adapted to large capacities (up to 3000 tons) although it is also possible to secure units as low as 50 tons in rating. Because centrifugal units operate best with refrigerants possessing a high specific volume and because of the simplification of lubrication difficulties, they are frequently used for extremely low temperature applications. They are adaptable to a wide range of temperatures from −130 F to 50 F. One important advantage is their flexibility under varying loads since units may be designed to operate with reasonable efficiency at capacities as low as 20 per cent of normal load.

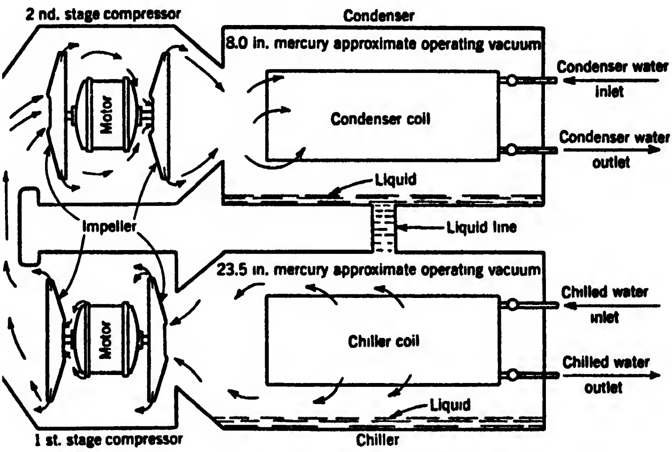


FIG. 11. ENCLOSED TYPE CENTRIFUGAL CONDENSING UNIT

Condensers

Condensers used for liquefying the refrigerant are of three general designs: (1) *air cooled*, (2) *water cooled*, and (3) *evaporative* (combination air and water).

1. *Air cooled condensers* are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to obtain, as, for instance, in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than would be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot discharge gas enters the coil at the top and, as it is condensed, flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooled air.

The principal disadvantages of air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on hot days requires that equipment of increased capacity

be selected to meet the peak load. Thus at normal loads the equipment is oversized. Their principal advantages are low installation costs and simplicity, and for these reasons they are frequently used in small self-contained units.

2. *Water cooled condensers* are commonly used with compressors of one horsepower or larger in size, and they are found almost exclusively on large installations. They usually prove to be the most economical choice if an adequate water supply and means for its disposal are available. Although water cooled condensers may be of many designs, the shell and coil and the shell and tube are most commonly found in present day practice.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is

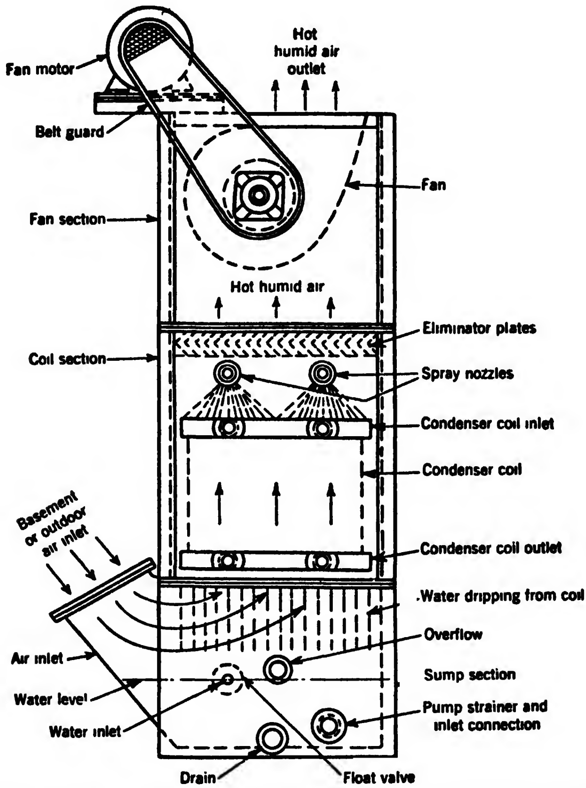


Fig. 12. Schematic View of an Evaporative Condenser

therefore necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs. For further information on cooling towers, reference may be made to Chapter 37.

3. *Evaporative condensers* were developed to alleviate the over-burdened water supply and drainage facilities of communities where many small air conditioning systems using water cooled condensers were applied. The adaptation of cooling

towers to small installations is not practicable. The evaporative condenser combines the functions of the two by using a minimum amount of water on a finned surface, cooling it to approximately the wet-bulb temperature of the surrounding atmosphere.

The end view of a typical evaporative condenser is shown in Fig. 12. The fan draws the air over a finned tube condenser which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil and the liquid refrigerant is drained from the bottom of the coil into a liquid receiver and then circulates through the remaining portion of the system in the usual way.

The water is circulated through the spray nozzles and the level is maintained in the sump by means of a float valve. The eliminator plates are placed in the path of the water-air mixture so as to remove the entrained water. The air leaving the unit is almost completely saturated, so that care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 100 tons or more. These units use only a small portion of the water required for a water cooled condenser. The water is vaporized by the heat of the refrigerant so that each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas under standard rating conditions where the water temperature rise is 20 F, each pound of water extracts only 20 Btu from the refrigerant. Including the water lost by entrainment in the discharge air, by overflow and stand-by evaporation, the water used is about 3 to 5 per cent of the amount that would be required for a water cooled condenser.

The evaporative condenser requires more maintenance, occupies greater space (must be located where air is available), and has a higher first cost than the water cooled condenser, but where the use of water is restricted or expensive, the evaporative condenser has become widely accepted. Compared with a water cooled condenser and cooling tower, which combination uses about the same quantity of water, the evaporative condenser has the advantage of lower cost and smaller space requirements.

Evaporators and Coolers

Refrigeration evaporators must be designed for efficient removal of heat from the medium being cooled as well as effective boiling of the refrigerant and a minimum drop of pressure through the coil. There are two general types of evaporators, dry and flooded. In the dry evaporator the refrigerant enters in the liquid state, and the design provides for complete evaporation with the vapors leaving slightly superheated. In flooded evaporators not all of the refrigerant is evaporated, the liquid-vapor mixture leaving the evaporator flows into a surge drum from which the vapors are drawn into the compressor suction line, and the liquid is recirculated through the evaporator.

The types of coolers used in connection with air conditioning work fall into three general groups: (1) direct water coolers, (2) direct air coolers, and (3) brine coolers for circulation of the brine in a closed system and thus cooling indirectly either water or air.

1. *Water coolers.* One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls

of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

2. *Air coolers*. When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

When refrigeration evaporators are used for cooling air or other gases by forced convection, they are usually termed blast coils or unit coolers. A blast coil may be placed in a duct or in an assembled unit and the air forced across the coil and discharged through distributing ducts or directly into the space to be conditioned. Unit coolers, designed much like unit heaters, consist of a finned coil, propeller fan, and controls suspended directly in the space to be cooled.

3. *Indirect brine coolers*. The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

REFRIGERATION CONTROL

In addition to the compressor, evaporator, and condenser, several auxiliaries are required for proper operation of a refrigeration system. Some device must be supplied for the controlled expansion of the refrigerant from the high condenser pressure to the low evaporator pressure; controls are required for the on-off operation of the compressor, the flow of the condensing medium, and for safety devices; proper piping is required for connecting the various portions of the systems.

Expansion Devices

Some form of expansion device must be provided to control the rate of flow of the liquid refrigerant between the high and low side pressures of the system. This device is usually an expansion valve and may be either manual or automatic; however, with few exceptions, manual valves are obsolete and no longer used.

Automatic Expansion Valves. An automatic or pressure controlled expansion valve operates to maintain a constant pressure in the evaporator. The liquid refrigerant passes through an orifice the opening size of which is controlled by means of a needle valve connected to a flexible bellows. This bellows expands or contracts with variations in the evaporator pressure transmitted to the expansion chamber through the refrigerant outlet from the evaporator. The position of this needle valve is controlled by the degree of compression in an adjustable spring, balanced against the bellows, and these two forces operate to maintain a constant pressure in the evaporator by increasing or decreasing the flow of liquid refrigerant. Such an expansion valve is usually applied to evaporators of the direct expansion type but is not satisfactory for fluctuating loads such as are encountered in air conditioning installations.

Thermostatic Expansion Valves. A thermostatic expansion valve controls the flow of liquid refrigerant to the evaporator so as to maintain the entire coil filled with evaporating refrigerant and to keep a constant superheat in the refrigerant gas leaving the coil. The construction of such a valve is shown in Fig. 13 and is similar to that for an automatic expansion valve but incorporates, in addition, a power element

responsive to changes in the degree of superheat of the refrigerant gas leaving the coil. This power element consists of a bellows connected by means of a capillary tube to a feeler bulb fastened to the suction line from the evaporator. The bulb, bellows, and tube are usually charged with the same liquid refrigerant used in the evaporator itself. A starved condition in the evaporator results in a greater superheat in the gas leaving the evaporator, and this in turn operates through the power element to increase the flow of liquid refrigerant. A flooded evaporator reduces the discharge superheat and thus tends to reduce the flow of liquid refrigerant. Such an expansion valve is satisfactory for operation with fluctuating loads since this type of control tends to keep the evaporator filled with refrigerant at all times.

Low-Side Float Valves. A liquid refrigerant control of the low-side float valve type consists of a ball float located in either a receiver or the evaporator itself on the low pressure side of the system. A needle valve operated through a simple lever mechanism attached to the float permits the passage of more or less refrigerant as the level in the receiver or the evaporator fluctuates. Such a control must be used in conjunction with a flooded evaporator. They have been applied extensively to household refrigerators and to some extent in commercial and industrial installations.

High-Side Float Valves. A high-side float valve differs from a low-side float valve in that the float is located in a receiver or container on the high pressure side of the system. Proper operation again depends upon a metering of the refrigerant through

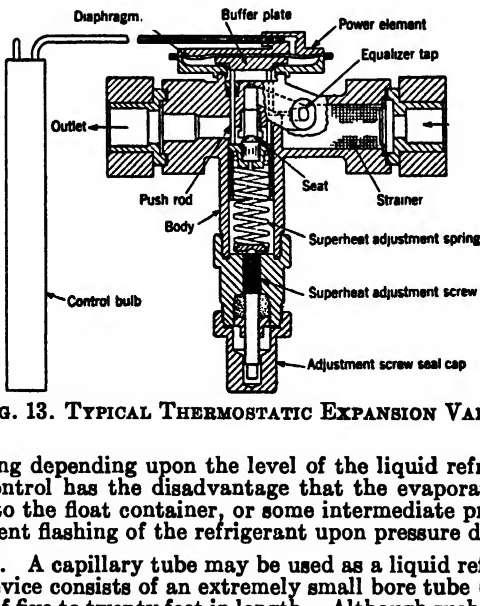


FIG. 13. TYPICAL THERMOSTATIC EXPANSION VALVE

a controlled opening depending upon the level of the liquid refrigerant in the container. Such a control has the disadvantage that the evaporator must be placed directly adjacent to the float container, or some intermediate pressure device must be applied to prevent flashing of the refrigerant upon pressure drop.

Capillary Tubes. A capillary tube may be used as a liquid refrigerant expanding device. Such a device consists of an extremely small bore tube (in the order of 0.04 inch in diameter) of five to twenty feet in length. Although such a restricting device operates as a very simple means of expanding the liquid refrigerant, it has the disadvantage that no modifications are possible to adjust the rate of expansion under various operating conditions. The bore and length of the tube as well as the proportions of the rest of the system are critical. It is for these reasons that its application has been limited to factory assembled domestic and commercial units.

Refrigeration Control Devices

In addition to automatic control of expansion of the liquid refrigerant, a completely automatic refrigeration system requires (1) some means for on-off operation of the compressor motor, (2) control of the flow of the condensing medium, and (3) safety devices for prevention of damage to the equipment. In addition, special controls designed for specific applications are frequently required. The various types of devices used to accomplish these purposes are so numerous that it would be impossible to describe all

of their modifications. Only the general purposes and operating characteristics of the more typical mechanisms are here discussed.

Compressor Motor Controls. Two types of controls are used for intermittently starting and stopping compressors. The first of these is a pressure motor control responsive to the evaporator pressure and the second a thermostatic motor control responsive to the temperature of the load surrounding the evaporators. In the first case the compressor operation is indirectly dependent upon the temperature of the load and is controlled by the refrigerant pressure at the point of control location. The second type is dependent upon the temperature of the load being cooled.

With the pressure actuated device, the control is frequently located directly on the condensing unit and the low pressure in the suction line or the crankcase of the compressor is used to control motor operation. Such a control usually consists of a low pressure bellows connected through tubing directly to the low pressure control source and an electrical switch operated through linkage by the movement of the bellows. The electrical circuit is closed on rising pressure and opened on falling pressure. The thermostatic type of motor control is usually similar in construction to the pressure control with the exception that a temperature bulb and capillary tube replace the pressure line, and the temperature bulb is located adjacent to the evaporator itself. In this case motor control is directly responsive to changes in the temperature of the load surrounding the evaporator. Frequently a high pressure safety cutout switch is combined with the motor control and operates to cut off the power from the motor in case the high side pressure exceeds a predetermined limit.

Solenoid Valves. Solenoid or magnetic stop valves are frequently used in refrigeration systems for control of gas and liquid flow. When applied as liquid stop valves, they are placed in the liquid line between the condenser or receiver and the evaporator, and the line is open to passage of the refrigerant only when the compressor is in operation. When the compressor is not in operation, leakage of liquid refrigerant in the evaporator is prevented. In some cases such a magnetic stop valve is operated directly by a thermostat located at the point of the load, and the compressor motor operation is controlled independently by a low pressure switch. Magnetic liquid stop valves are also widely used for the control of the refrigerant flow to individual evaporators in a multiple evaporator system operated by one compressor. In some installations magnetic liquid line and suction line valves are used to isolate completely an evaporator for defrosting purposes. A magnetic valve may be installed in a by-pass around one or more cylinders of a multiple cylinder compressor and thereby be used to unload a compressor during starting to reduce load. Additional applications are found in control of the circulation of brine in a secondary refrigeration system.

Suction Pressure Valves. Suction pressure control valves, frequently called back pressure regulators or two-temperature valves, are sometimes placed in the suction line to prevent the evaporator pressure and temperature from dropping below a predetermined level. Typical applications of such controls occur in water cooling or milk cooling systems where freezing and other damage would result if the evaporator pressure dropped too low or in multiple systems where several evaporators are supplied by one condensing unit. Thus, different evaporators may be kept at different temperatures by maintaining a pressure drop between the evaporator and the suction line.

Condensing Water Control. The majority of the refrigeration systems, other than fractional horsepower, use water cooled rather than air cooled condensers since the lower condensing temperatures result in more economical operation. Automatic control of the water flow to the condenser must be maintained if water wastage is to be eliminated. Such control may be provided through the use of either an electric solenoid water valve or by means of a pressure control valve. With a solenoid valve, the flow is two-position, either *off* or *on*, and its operation is simultaneous with starting and stopping of the compressor motor. With a pressure operated valve, the flow is modulated and is dependent entirely upon condenser pressure rather than condensing unit operation. Similar water valves controlled thermostatically by the temperature of the water discharged from the condenser are also available.

Safety Controls. Many controls are designed not to aid in proper operation of the system but to prevent damage in case of improper operation. One such safety control is the high-pressure cut-off frequently combined with the low-pressure motor control as previously described. Another safety control often used is a low voltage cut-off which shuts down the system automatically in case the line voltage drops below a minimum value. High pressure relief valves are used for safety purposes to prevent damage in case excessive condensing pressures are encountered. Oil separators are often installed between the compressor and the condenser to prevent excessive oil

TABLE 5. MAXIMUM TONS REFRIGERATION FOR AMMONIA MAINS

Pipe Size, In.	Suction Line: Suction pressure psig (F temp) 5 (−17.2 F)	Suction Line: Suction pressure psig (F temp) 20 (5.5 F)	Suction Line: Suction pressure psig (F temp) 45 (30 F)	Discharge Line	Liquid Line: Condenser to receiver	Liquid Line: Receiver to system
3/8	—	—	—	—	2.5	12.0
1/2	0.6	1.1	2.0	3.1	6.0	20.0
3/4	1.2	2.2	4.1	6.0	14.0	75.0
1	2.2	4.0	7.5	11.4	24.0	137.
1 1/4	4.4	8.0	15.0	22.4	50.0	245.
1 1/2	6.4	11.8	21.6	30.9	77.0	400.
2	12.1	22.2	42.0	62.0	140.	850.
2 1/2	19.1	35.5	65.0	97.5	220.	1475.
3	31.5	59.0	108.	160.	375.	2400.
3 1/2	46.6	87.5	156.	238.	540.	3500.
4	64.0	118.	240.	330.	740.	
5	117.	208.	385.	560.	1320.	
6	175.	306.	600.	905.	2030.	
8	362.	650.	1200.	1810.	4200.	
10	640.	1180.	2160.	3200.		
12	940.	1850.				

pumping from the crankcase into the condenser and the evaporator. Separation of the oil from the refrigerant gas is usually accomplished by slowing down the gas velocity sufficiently to allow the oil to separate out by gravity.

REFRIGERATION PIPING

The pressure drop which occurs during passage of the refrigerant through connecting piping is similar in effect to that which occurs through suction and discharge valves of the compressor. Thus, the effect of the pressure drop in the suction line between evaporator and compressor requires that a lower pressure be maintained inside the compressor during suction than is maintained in the evaporator. The pressure drop through the connecting piping between the compressor and the condenser requires that a higher

TABLE 6. FREON-12 LIQUID LINES, TONS CAPACITY PER 100 FT EQUIVALENT LENGTH

Line Size, Inches	Pressure Drop per 100 Ft Equivalent Length, Psi: 3	5	10	20
3/8 OD	0.88	1.14	1.80	2.58
1/2 OD	2.89	3.64	5.56	8.50
1/2 IPS	4.86	6.81	10.2	15.8
5/8 OD	4.86	6.81	10.2	15.8
3/4 IPS	9.73	12.6	18.5	27.0
7/8 OD	10.5	14.1	21.8	33.0
1 IPS	21.4	28.2	41.3	60.8
1 1/8 OD	21.4	28.2	41.3	60.8
1 1/4 IPS	36.9	48.1	70.5	101.
1 3/8 OD	36.9	48.1	70.5	101.
1 1/2 IPS	62.0	80.2	114.	160.
1 5/8 OD	62.0	80.2	114.	160.
2 IPS	124.	161.	231.	328.
2 1/2 IPS	230.	297.	426.	607.
3 IPS	364.	469.	676.	972.
3 1/2 IPS	539.	704.	1005.	1430.
4 IPS	753.	972.	1385.	1945.

Note: Tonnage values above those underlined give velocities of 300 fpm or less.

pressure be maintained inside the compressor during discharge than in the condenser. These losses result in a greater compression ratio and therefore greater power requirements as well as a lower volumetric efficiency and higher displacement requirements. Pressure losses in the liquid line between condenser or receiver and the expansion valve may result in some flashing of the liquid refrigerant unless the liquid is subcooled. In all cases frictional losses should be kept to a minimum, and piping should be

TABLE 7. MAXIMUM TONS OF COMPRESSOR CAPACITY FOR FREON-12 LINES
(Only for temperatures indicated)

Line Size, Inches	Suction Lines Based on 105 F Condensing Temperature						Discharge Lines	
	Psi Pressure Drop per 100 Ft Equivalent Length at 40 F Saturation						Condensing Temperature	
	½	1	2	3	4	5	115 F	90 F
½ OD	0.14	0.20	0.28	0.35	0.41	0.45		
⅜ IPS	0.17	0.24	0.34	0.42	0.49	0.54		
⅝ OD	0.25	0.35	0.51	0.62	0.73	0.81	1.43	1.15
½ IPS	0.35	0.45	0.65	0.79	0.93	1.03	1.87	1.50
⅞ OD	0.55	0.76	1.10	1.34	1.58	1.75	2.97	2.38
¾ IPS	0.68	0.94	1.35	1.65	1.92	2.12	3.26	2.62
1⅛ OD	1.26	1.80	2.57	3.17	3.76	4.15	5.05	4.05
1 IPS	1.43	2.01	2.89	3.54	4.17	4.60	5.29	4.25
1⅜ OD	2.21	3.12	4.45	5.50	6.38	7.05	7.72	6.19
1¼ IPS	2.70	3.82	5.37	6.72	7.68	8.48	9.16	7.35
1⅝ OD	3.40	4.78	6.79	8.42	9.77	10.8	10.92	8.75
1½ IPS	4.05	5.75	8.10	10.12	11.6	12.8	12.5	10.0
2⅛ OD	6.12	8.60	12.1	15.1	17.4	19.2	19.2	15.3
2 IPS	7.66	10.9	15.3	19.2	32.2	24.5	20.6	16.5
2⅝ OD	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
2½ IPS	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
3⅛ OD	19.1	27.2	38.2	47.8	55.0	60.7	51.5	39.8
3 IPS	20.9	29.4	42.3	51.8	60.0	66.2	54.5	43.8
3⅝ OD	27.8	39.7	55.7	69.8	80.3	88.7	72.0	57.6
3½ IPS	30.2	43.2	61.0	76.1	87.0	96.0	78.8	63.3
4⅛ OD	38.6	55.2	78.0	97.3	111	123	95.8	77.1
4 IPS	40.7	58.6	83.0	103	118	130	101.6	81.6
5 IPS	71.3	100	141	176	203	224	171.5	137.8
6 IPS	126	183	257	322	366	403	266	214
8 IPS	211	297	422	523	602	664	461	370
10 IPS	352	503	712	887	1024	1130	725	582
12 IPS	550	780	1106	1373	1582	1748	1041	836

selected which will give the smallest loss consistent with overall economy in the system.

Refrigerant liquid lines from the receiver to the expansion valve should be preferably designed with a pressure drop of less than 5 psi and with 10 psi as the maximum. A velocity of 100 to 250 fpm is recommended to prevent a pressure drop great enough to cause vaporization of the refrigerant ahead of the expansion valve. If the evaporator is to be located at a higher elevation than the condenser or receiver, account should be taken of the pressure drop for each foot of static liquid lift. Approximate values are 0.26 psi per foot for ammonia, 0.57 psi per foot for Freon-12, 0.51 psi

per foot for Freon-22, and 0.64 psi per foot for Freon-11. Where there is a possibility of vaporization of some of the liquid before reaching the expansion valves, means for subcooling should be provided.

Since a reduction of suction pressure at the compressor results in an appreciable reduction in capacity and more power input per ton of refrigeration, great care should be given to the proper sizing of suction lines between the evaporator and the compressor. Although comparatively high velocities, 500 to 5000 fpm, may be used, the optimum value will depend upon the refrigerant and the operating pressure range. Since return of the oil to the compressor must be considered in the case of Freon and methyl

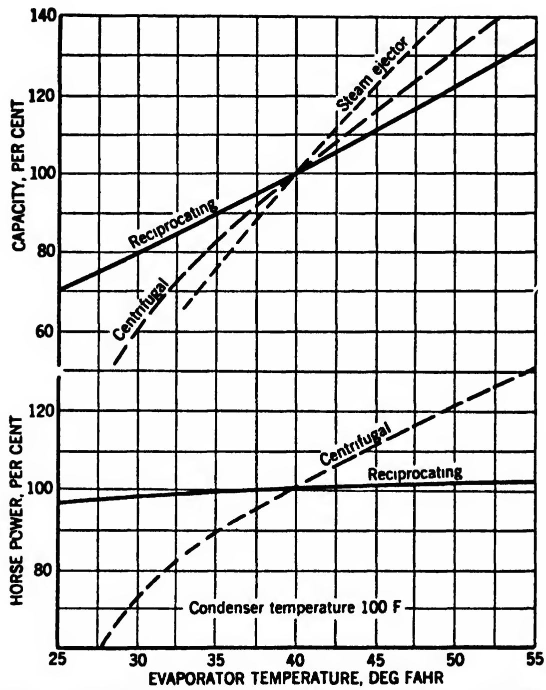


FIG. 14. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

chloride, for these refrigerants the minimum velocity should be 500 fpm for horizontal runs and 1000 fpm for vertical runs. For the Freons, the usual design velocities range between 1000 and 2000 fpm. Too high velocities create noise problems and excessive pressure drops. The total pressure drop in the suction line should be between one and two psi if the velocity can be kept to within the specified limits.

Compressor discharge or hot gas lines may be designed with velocities from 1000 to 5000 fpm except for dense gases such as carbon dioxide where noise considerations will reduce the upper limit. A pressure drop of 2 to 4 psi is recommended for the discharge lines. Extensive tables are available in the literature for the determination of pressure drops through refrigerant lines with various refrigerants. The capacities listed in Tables 5, 6 and 7 are published in ACRMA Equipment Standards-1946, of the *Air Conditioning and Refrigerating Machinery Association* and are used by permis-

sion. Table 5 shows the maximum tonnage of refrigeration normally allowed for ammonia lines, assuming a maximum of 100 feet of equipment length of pipe. Table 6 shows the tonnage capacity normally allowed for Freon-12 liquid lines per foot equivalent length of pipe and Table 7 the maximum tonnage for suction and discharge Freon-12 lines.

EQUIPMENT CHARACTERISTICS AND SELECTION

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 14 and 15.

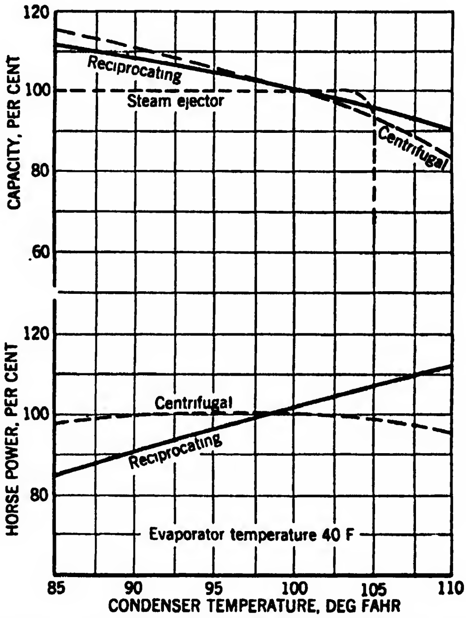


FIG. 15. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

From Fig. 14 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evaporator temperature. Thus, both the steam jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 14 that the steam jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 15. It may be noted that the power required by the reciprocating compressor increases rapidly with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the ca-

TABLE 8. BASIS OF EQUIPMENT SELECTION

CAPACITY TONS	MAJORITY USED	SOME USED	FEW USED
0 to 5	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built up central systems.
5 to 25	Built up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Unit systems in conditioned space. Built up systems using absorption and adsorption systems.
25 to 50	Built up central systems using reciprocating compressors.	Built up central systems using centrifugal compressors.	Central systems using adsorption systems.
50 to 400	Built up central systems using reciprocating compressors.	Built up central systems using steam jet and centrifugal compressors.	
400 and Over	Built up central systems using centrifugal compressors.	Built up central systems using steam jet.	

pacity of the stem jet compressor is independent of condenser temperature until a certain point is reached where it drops to zero. As previously stated, steam jet equipment requires more condensing water than other types of compression systems. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

The selection of proper refrigeration equipment for any air conditioning job is of utmost importance for satisfactory results. The most important factors in the selection of the equipment are:

1. Loads (as determined by the conditions of the space to be cooled).
2. Economics (both initial and operating costs).
3. Codes (local safety codes must be adhered to and influence the type of system to be used).

A broad division of equipment to be used for a particular installation or application may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 8.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator, and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete, ready to install and are the most economical (see Chapter 36).

The reciprocating compressor in the built-up central system (see Chapter 43) covers the widest range of application since it is applicable to either the direct expansion or indirect systems and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system with the exception of the centrifugal compressor, which uses the same amount.

Centrifugal compressors are used for large installations, and usually

TABLE 9. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

Type of Enclosure	Load, Btu per Hour			Ratio Sensible to Total	Air Entering Coil		Operating Balance Point		
	Sensible	Latent	Total		F Deg	Per Cent R.H.	Evaporator Temp F Deg	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant.....	103,000	45,000	148,000	0.695	82	45	34.4	123	69.9
Office...........	121,000	27,000	148,000	0.820	82	45	42.2	100	82.1

where the indirect system is required. The driving mechanism can be a steam turbine or electric motor. The steam jet system is used where steam is available and cooling water can be had in large quantities.

It will be noted by referring to Fig. 14 that all systems using compressors have a common characteristic and that is that the capacity varies with the evaporating temperature. Not only can the equipment be selected to produce a given result, but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 16. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

Data given in Table 9 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different ratios of sensible to total heat. In the case of the office with a ratio of

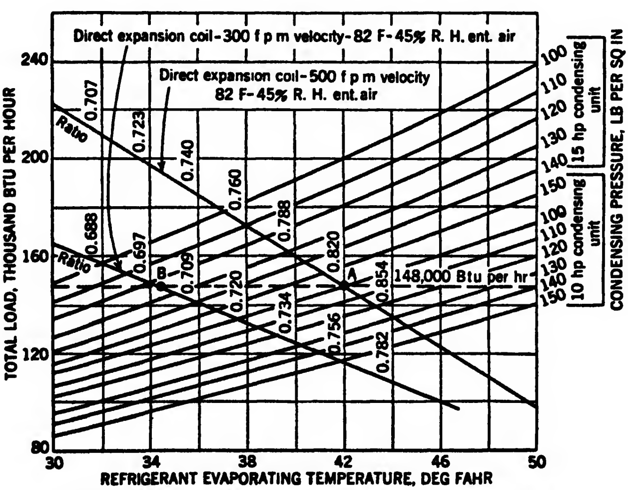


FIG. 16. COMPRESSOR AND COIL PERFORMANCE

82 per cent sensible to total heat, the operating point *A* in Fig. 16 is found to be 42.2 F evaporating temperature with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 per cent sensible to total heat, the air velocity is lowered to 300 fpm and the evaporating temperature is lowered to 34.4 F as shown in point *B* of Fig. 16. In order to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in Fahrenheit degrees.

ABBREVIATIONS AND SYMBOLS IN CHAPTER

(CP) = coefficient of performance, ratio of refrigerating effect to the heat equivalent of the compressor work.
(CVE) = clearance volumetric efficiency.
d = internal diameter in inches.
H_t = cooling load in tons.
h_d = enthalpy of vapor at condition of discharge from compressor.
h_{fc} = enthalpy of liquid at discharge from compressor.
h_{fd} = enthalpy of liquid at discharge of expansion valve.
h_{fs} = enthalpy of liquid at entrance to expansion valve.
h_m = enthalpy of mixture.
h_v = enthalpy of saturated vapor.
h_{vd} = enthalpy of saturated vapor at discharge of valve or compressor.
h_{vs} = enthalpy of saturated vapor at state s entering compressor.
hp = horsepower.
p_d = pressure of saturated liquid and vapor at discharge of compressor.
p_s = pressure of saturated liquid.
psig = pressure pounds per square inch, gage.
psia = pressure pounds per square inch, absolute.
Q = quantity of heat, Btu.
Q_c = heat loss from condenser, Btu per pound refrigerant.
Q_j = heat dissipated in cooling water, Btu per hour.
s = entropy.
Δs = entropy change between suction and discharge.
T = absolute temperature, Fahrenheit degrees.
T_{avg} = average temperature, Fahrenheit degrees, absolute, of gas passing through compressor.
T_c = condenser temperature, Fahrenheit degrees, absolute.
T_s = evaporator temperature, Fahrenheit degrees, absolute.
(TVE) = total volumetric efficiency.
t_{sd} = degrees superheat at discharge condition of vapor leaving compressor.
t_d = discharge temperature, Fahrenheit degrees.
V_c = clearance, percentage of volume, swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.).
v_s = specific volume of gas at suction, cubic feet per pound.
v_d = specific volume of gas at discharge, cubic feet per pound.
W_r = refrigerant rate, pounds per minute.
x = proportion of liquid in mixture of vapor and liquid.

REFERENCES

[1] Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 33).

[2] The Application of Storage Refrigeration to Air Conditioning, by C. F. Boester (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 675).

[8] Use of Cold Accumulators in the Air Conditioning Field, by R. W. Evans and C. J. Otterholm (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 123).

BIBLIOGRAPHY

A. Refrigerating Data Book, Vol. 1 (*American Society of Refrigerating Engineers*).

B. Refrigeration Engineering, by H. J. Macintire (John Wiley & Sons).

C. Theory of Mechanical Refrigeration, by N. R. Sparks (McGraw-Hill Book Co., Inc.).

D. Refrigeration and Air Conditioning Engineering, by B. F. Raber and F. W. Hutchinson (John Wiley & Sons).

E. Refrigeration, by J. A. Moyer and R. U. Fittz (McGraw-Hill Book Co., Inc.).

F. Refrigerants and Absorbents, by W. R. Hainsworth (*Refrigerating Engineering*, August and September, 1944).

G. Air Conditioning and Refrigeration, by B. H. Jennings and S. R. Lewis, 1944 (International Textbook Company).

H. Refrigeration and Air Conditioning, by Jordan and Priester (Prentice-Hall, Inc., 1948).

CHAPTER 40

AIR DISTRIBUTION

Standards for Satisfactory Conditions, Definitions, Mechanics of Air Distribution, Outlet Performance, Types of Air Outlets, Outlet Location and Selection, Directional and Volume Control, Return and Exhaust Intakes, Specific Applications

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. An air conditioning system may deliver the required quantity of conditioned air and still fail to give satisfactory room conditions because of poor air distribution. The scope of the chapter is limited to the air distribution within the conditioned space. Reference is made to the distributing duct system only insofar as it affects the performance of the air distribution outlet. See Chapter 41 for information on air duct design.

STANDARDS FOR SATISFACTORY CONDITIONS

The object of air distribution is to create within the space the proper combination of room temperature, air motion and humidity, whether by cooling, heating or ventilating. The purpose to be accomplished determines the factors to be controlled. For instance, in many industrial applications it is necessary to maintain proper standards throughout a large portion of the space; sometimes almost throughout the entire enclosure. In these cases design room temperature, room air motion and humidity will depend entirely upon the requirements of the product and its manufacturing processes.

If, however, comfort of the occupants is the principal objective, consideration of the occupied zone (floor to 6 ft above floor level) is primarily required. In order to obtain comfort conditions within this zone, standard limits have been set up as acceptable *effective temperatures*. This term comprises air temperature, motion, humidity and their physiological effect on the surface of the human body. Any variation from accepted standards of one of these elements may result in discomfort to the occupants. The same effect may be caused by lack of uniformity of conditions within the space or by excessive fluctuation of conditions in the same part of the space. Such discomfort may arise due to excessive room air temperature variations (horizontally, vertically, or both), excessive air motion (draft), failure to deliver or distribute the air according to the load requirements at the different locations, or too rapid fluctuation of room temperature or air motion (gusts).

In addition the noise level created by the introduction of supply air should be kept within acceptable limits, and streaking or smudging of walls or ceilings should be prevented.

With reference to permissible room air motion it is not possible to establish a specific standard covering the entire complex problem of air distribution. Velocities less than 15 fpm generally cause a feeling of air stagnation, whereas velocities higher than 65 fpm will disturb loose paper sheets

on desks and may result in a sensation of draft. Air velocities of 25 to 35 fpm in the occupied zone are most satisfactory, but air motion of 20 to 50 fpm will usually be acceptable, particularly when the lower part of this range of velocity is used in cooling applications and the higher values on heating jobs. In any case, it is certain that the effect of room air motion on comfort or discomfort depends on air temperature and direction as well as on velocity.

Reference should be made to Chapter 12, Physiological Principles, for information on effective temperature and comfort zones. Material in Chapter 42, Sound Control, covers acceptable room noise levels and noise generated by air outlets.

DEFINITIONS

The following definitions referring to air distribution equipment have gained general acceptance.

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

2. *Exhaust Opening or Return Intake:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

5. *Grille:* A covering for any opening and through which air passes. A supply grille discharges air axially with a limited spread.

6. *Register:* A grille equipped with a damper.

7. *Free Area:* The total minimum area of the openings in the air outlet or inlet through which air can pass.

8. *Core Area:* The total plane area of the portion of a grille, bounded by a line tangent to the outer edges of the outer openings through which air can pass.

9. *Mean Area:* The total of the core and free areas divided by two.

10. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

11. *Aspect Ratio:* The ratio of length of the core of a grille to the width.

12. *Vane Ratio:* The ratio of depth of vane to shortest opening width between two adjacent vanes.

13. *Plaque:* A ceiling outlet in which the supply air impinges against a plate or series of parallel plates and is discharged horizontally in all directions.

14. *Diffuser:* An outlet discharging supply air in various directions and planes, thereby effecting its mixture with the room air.

15. *Primary Air:* The air delivered to the outlet by the supply duct.

16. *Induction:* The entrainment of room air by an air stream.

17. *Internal Induction:* The induction of room air drawn into an outlet by the primary air stream. (Commonly called *aspiration*).

18. *External Induction:* The induction of room air by the air stream discharged from the outlet (commonly called *secondary air motion*).

19. *Induced Air:* The room air entrained by the primary air through internal induction or by the discharged air through external induction or both.

20. *Total Air:* The mixture of primary air and induced air.

21. *Induction Ratio:* The total air divided by the primary air.

22. *Throw (Blow):* The horizontal distance an air stream travels on leaving the outlet (grille) to a position at which air motion reduces to a maximum velocity of 50 fpm.

23. *Drop:* The vertical distance, the lower edge of the air stream drops between the outlet and the end of its throw.

24. *Rise:* The converse of drop.

25. *Envelope:* The outer boundary of an air stream moving at a perceptible velocity.

26. *Spread:* The divergence of the air stream in a horizontal or vertical plane after it leaves the outlet.

27. *Diffusion:* Distribution and mixing of air within a space, accomplished by an outlet discharging supply air in various directions and planes in order to effect the desired air conditions in the occupied zone of that space.

28. *Radius of Diffusion:* The horizontal distance from the diffuser outlet to the perimeter of the space, within which effective diffusion is accomplished and air motion in the occupied zone is reduced to 50 fpm maximum.

29. *Outlet Velocity:* The average velocity of air emerging from the outlet measured in the plane of the opening.

30. *Terminal Velocity:* The average air stream velocity at the end of the throw.

31. *Temperature Differential:* Temperature difference between primary and room air.

32. *Temperature Variation:* Temperature difference between points of the same space.

MECHANICS OF AIR DISTRIBUTION

In the mechanics of air distribution, two major problems are involved: (1) complete mixing of the primary air and air outside of the zone of occupancy in order to reduce the temperature difference and air motion to acceptable limits before the air enters the occupied zone; and (2) counteraction of the natural convection and radiation effects within the room.

The theory concerning the distribution of conditioned air within an enclosure is still incomplete and no general law governing outlet performance has been formulated. The characteristics and performances of the various existing types of outlets must therefore be evaluated largely by experimental work. Some progress has been made concerning the theoretical analysis of the characteristics of a primary air stream discharged in an unconfined space, i.e., a space large enough so that the primary air stream is not disturbed by contact with surfaces, or by adjacent streams. The approach to this problem is usually made by means of the momentum theory. Development of this theory has so far been confined to side wall distribution of air, because this is its most elementary application. Fundamentally, the same laws apply also to ceiling distribution, but a great amount of additional research is still required to adapt them to the more complicated conditions of deflection of air up to 90 degrees, spread up to 360 degrees and the resulting rapid induction. There is also an element of downward diffusion which usually is not obtainable with side wall grilles.

Momentum Theory

When air is discharged from an outlet into a free open space, the primary air stream entrains room air as it traverses the space. This entraining effect increases the cross-sectional area and reduces the velocity of the resulting air stream. Induction takes place with the conservation of linear momentum; this has been confirmed by tests which indicate that the momentum remains almost constant throughout the entire measurable length of the air stream. This relationship may be expressed by Equation 1:

$$M_1V_1 + M_2V_2 = (M_1 + M_2)V_3 \tag{1}$$

where

M_1 = mass of primary air.

M_2 = mass of induced air.

V_1 = velocity of primary air.

V_2 = velocity of induced air (for practical use, $V_2 = 0$).

V_3 = velocity of the mixture.

If the velocity of induced air is zero, Equation 1 changes to:

$$M_1V_1 = (M_1 + M_2)V_3$$

or,

$$\frac{V_1}{V_3} = \frac{M_1 + M_2}{M_1} = r \qquad (2)$$

Since in many applications the densities of primary and room air are about equal, air volumes may be substituted for mass and Equation 2 becomes:

$$\frac{V_1}{V_3} = \frac{Q_1 + Q_2}{Q_1} = \frac{Q_3}{Q_1} = r \qquad (3)$$

where

Q_1 = volume of primary air, cubic feet per minute.

Q_2 = volume of secondary air, cubic feet per minute.

Q_3 = volume of mixture of primary air and induced air, cubic feet per minute.

r = induction ratio.

Jet Pattern From Round or Rectangular Openings in a Large Room

The relation between the shape of the discharge of a jet and the shape of the conventional outlet has long been the subject of research. It has been proved to be incorrect to assume that the jet retains the outlet shape when it discharges into a free open space[1]. Air streams from rectangular outlets having low aspect ratios develop a symmetrical or cone shape within a few diameters from the outlet face. From there on, the jet continues to expand at a fairly constant rate. Beyond 20 diameters there is very little difference between round and rectangular jets. The assumption can be made that the apex of the cone is in the same position for any jet having a small aspect ratio. For the more usual problems of the conventional room with outlets near the ceiling, there are insufficient experimental data to justify a definite statement on the effect of aspect ratio.

If the round or rectangular opening is divided into a number of orifices having straight sides, the performance of the air stream will be similar to that of a plain opening.

Velocity Across Jets

Results of many tests[1] indicate that the ratio of centerline velocity to average velocity is about 3, irrespective of outlet size, shape or initial velocity. This statement is true for stream cross-sections located beyond 10 diameters from the outlet, and is fairly accurate for distances up to 50 diameters. Experimental data are lacking for distances beyond 50 diameters.

Effect of Aspect Ratio on Entrainment

In slotted outlets, the air entrainment of the primary jet is a function of aspect ratio[1]. This effect is most pronounced when large changes in the ratio are made. A comparison between a slot of aspect ratio 24 and a square opening of the same area is given in curves *A* and *B* of Fig. 1. At

a distance of 8 ft from the outlet, the entrainment of the slot is 8.1 as compared with 6.9 for the square, or an increase of about 17 per cent.

Curve *C* shows the further increase in entrainment obtained by using an aspect ratio of 48. An increase of 40 per cent is obtained over the 24 in. x

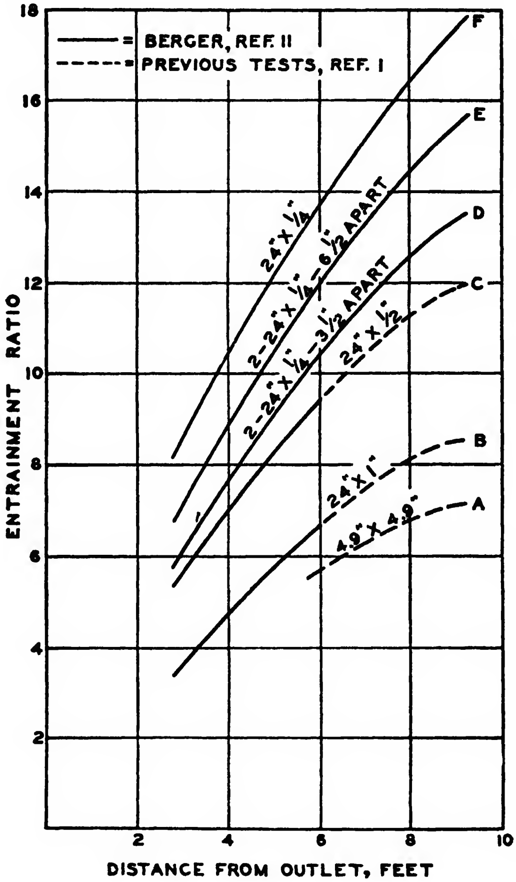


Fig. 1. Typical Relation of Entrainment Ratio to Distance from Outlet for Slotted Outlets. (Based on 800 fpm Outlet Velocity.)

1 in. slot. This indicates that long narrow slots produce air streams that give high induction of secondary air.

Parallel Slots

The use of several slots in parallel to vary the rate of air entrainment depends mainly on the distance between the slots. If close together, the air pattern is about the same as for a single opening of equal area. Spacing the openings farther apart gives an increase in entrainment as shown on curves D and E of Fig. 1. It will be noted that 2 openings 24 in. x ¼ in. located very close together will obtain an entrainment which is about the same as obtained with one 24 in. x ½ in. opening. However, if the slots are spaced 6½ in. apart there is a marked increase in entrainment.

Throw

Equations for the throw of straight flow side wall outlets have been developed on the basis of the momentum theory. Equation 4 states the throw in terms of the area of the outlet and the primary air volume[2]:

$$L = 0.82 \frac{Q_1}{\sqrt{A_1}} \tag{4}$$

where

L = throw, ft.

A_1 = effective outlet area, in square inches = (gross measured area) × (percentage of free area/100) × (discharge coefficient).

The discharge coefficient is approximately 0.8.

Equation 4 has been developed under the assumption that the temperature of the supply air is the same as the temperature of the room air. It applies only to straight flow outlets with aspect ratios less than 16.

Equation 5 for the performance of straight flow outlets evolved from research[1] allows the calculation of the maximum residual velocity at any distance perpendicular to the outlet face. It applies for aspect ratios up to 50.

$$V_r = K \frac{V_1\sqrt{A_1}}{X} = K \frac{Q_1}{X\sqrt{A_1}} \tag{5}$$

where

V_r = maximum residual velocity in air stream, i.e., the highest maintained velocity at the given cross section in the room, feet per minute.

V_1 = average initial velocity across outlet, feet per minute.

K = constant of proportionality.

A_1 = effective outlet area in square feet = (gross measured area) × (percentage of free area/100) × (discharge coefficient).

X = normal distance from outlet face, feet.

Equation 5 together with Equation 6 (which reduces to Equation 7 if the jet angle is 20 deg) for the entrainment ratio,

$$\text{Entrainment Ratio} = \frac{0.785\,K}{RX\sqrt{A_1}}\left(\sqrt{\frac{A_1}{0.785}} + 2\,X \tan\frac{\theta}{2}\right)^2 - 1 \tag{6}$$

where

R = ratio of maximum residual velocity to average residual velocity.

θ = jet angle or spread angle in degrees.

$$\underset{\text{(20 deg jet angle)}}{\text{Entrainment Ratio}} = \frac{0.785\,K}{RX\sqrt{A_1}}\left(\sqrt{\frac{A_1}{0.785}} + 0.35\,X\right)^2 - 1 \tag{7}$$

has been used to develop charts[3] which provide the graphical solution of problems involving the determination of the throw of air from slots and jets, the residual velocity, and the size of openings. (See Figs. 2 and 3). The charts apply only to *air discharging into room air of same temperature as the stream*. They can be used to determine the throw of air and

ratios up to 40:1 with initial velocities of 1000 to 6000 fpm and with residual velocities of 100 to 1000 fpm. The charts furthermore are for use with sharp-edged orifices or slots, and *include the coefficient of discharge.* If air is discharged from an orifice with a well-rounded entrance or from a length of straight duct, the coefficient of discharge is unity and the actual area of the opening is the effective area. For such rectangular openings the effective diameter is the diameter of a circle with an area equal to the actual area of the rectangle. The following examples will illustrate the use of the charts:

Example 1: Air is delivered to a cooler through independent slots each 24 in. x 2 in.

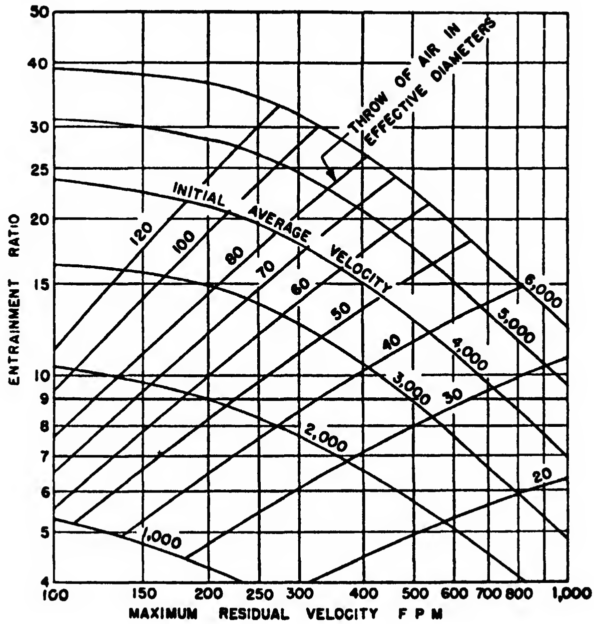


FIG. 2. RELATION BETWEEN INITIAL VELOCITY, RESIDUAL VELOCITY, ENTRAINMENT RATIO AND THROW OF AIR FROM JETS AND SLOTS

with an initial velocity of 2000 fpm. Determine the maximum residual velocity and the entrainment ratio at a distance of 15 ft from the slot.

From Fig. 3 the effective diameter = 6.2 in. = 0.52 ft. The number of effective diameters in 15 ft = 15/0.52 = 28.8.

From Fig. 2 at 2000 ft initial velocity read entrainment ratio = 6.6 and maximum residual velocity = 390 fpm. From tests it has been shown that the *average* residual velocity may be taken as $\frac{1}{3}$ of the *maximum* or 130 fpm in this case.

Example 2: Using the data from *Example 1* determine the distance at which the maximum residual velocity will be 150 fpm.

From Fig. 2 at V_1 = 2000 and V_r = 150, the number of effective diameters is read directly as 73 and the throw of the air is therefore 73 x 0.52 = 38 ft.

Example 3: Air issues from a round orifice plate with an initial average velocity of 4000 fpm. It is to have a maximum residual velocity of 400 fpm at a distance of 30 ft from the opening. Calculate the size of the opening required and the entrainment ratio.

On Fig. 2 at the intersection of the curve of 4000 fpm, the entrainment ratio is read directly as 15 and the effective diameters of throw = 55.

Since 55 effective diameters are equal to 30 ft as required, 1 effective diameter = $\frac{30}{55}$ = 0.545 ft or 6.56 in.

On Fig. 3 vertically below intersection of 6.56 in. effective diameter line and equivalent round opening line read 8.5 in. in lower margin.

Example 4: A jet of air issues from a pipe or from air orifice having a well-rounded

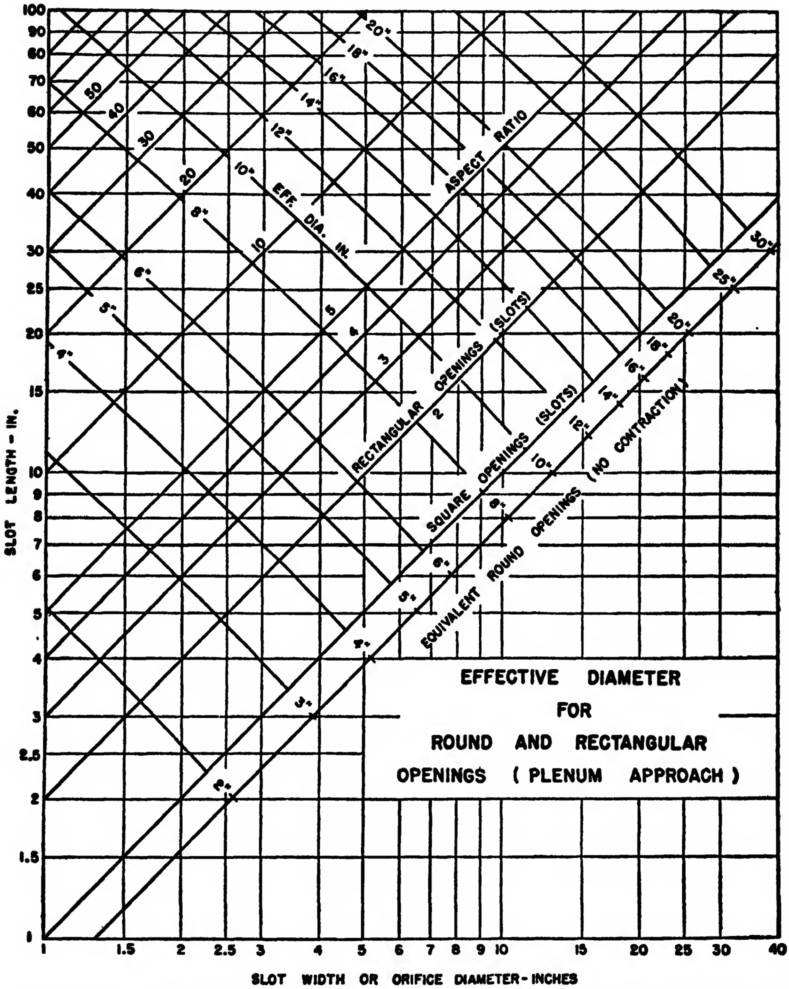


FIG. 3. EFFECTIVE DIAMETERS FOR ROUND AND RECTANGULAR OPENINGS (PLENUM APPROACH)

entrance (coefficient of discharge = 1.0) and delivers air with the same velocities and with the same throw as in *Example 3*. What is the required diameter?

In this case since the coefficient of discharge is unity, the effective diameter of the jet is the actual diameter of the pipe or orifice, or 6.56 in., as obtained in *Example 3*.

Spread

The induction effect results in the spreading of the air stream. The total angle included by the air stream from straight flow outlets has been meas-

ured and found to be between 14 and 25 deg. The angle will depend on the type of approach, type of outlet and velocity.

The effect of vertical bars placed in the face of the outlet to increase the spread, may also be deduced from the momentum theory. Assuming that there are no horizontal deflecting bars and that the air spreads vertically through a total angle of 14 deg; that a uniform velocity exists at any section of the air stream; and that the conservation of momentum principle applies down to a velocity of 60 fpm; the following approximate equations for throw are to be substituted for equation (4)[2]:

$$\text{For a spread of 15 deg on each horizontal side } L = 0.55 \frac{Q_1}{\sqrt{A_1}} \tag{8}$$

$$\text{For a spread of 30 deg on each horizontal side } L = 0.37 \frac{Q_1}{\sqrt{A_1}} \tag{9}$$

$$\text{For a spread of 45 deg on each horizontal side } L = 0.28 \frac{Q_1}{\sqrt{A_1}} \tag{10}$$

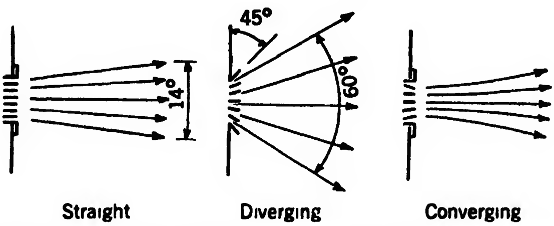


Fig. 4. Spread of Air Stream with Various Vanes

Guide Vanes

Vanes should have a depth of one to two times the spacing between the vanes. If the ratio of vane depth to spacing is less than one, effective control by means of the vanes cannot be obtained. Little improvement is obtained by increasing the ratio beyond two. The effect of various types of vanes is given in following paragraphs.

Straight Vanes. As mentioned previously, the included angle between both planes will be in the neighborhood of 14 deg, for a straight setting of the vanes as shown in Fig. 4.

Diverging Vanes. Such vanes set for an angular spread will have a marked effect on the direction and distance of travel of an air stream. An outlet having vertical vanes set straight forward in the center, with uniformly increasing angular deflection to a maximum at each end of 45 deg, will produce an air stream with a horizontal included angle of approximately 60 deg as shown in Fig. 4. The throw will be reduced one-half for such a vane setting. Increasing the divergence of the vanes reduces the air quantity handled by an outlet for a given duct static pressure. The primary function of the vanes is to spread the air horizontally. Spreading the air vertically entails the risk of hitting beams or other obstructions or of blowing primary air at excessive velocities into the occupied zone.

Converging Vanes. The blow of an outlet may be somewhat increased by converging the vanes of an outlet as illustrated in Fig. 4. Even with converging vanes, the resultant angle of spread of an air stream will not

be less than 14 deg. The air converges for a few feet in front of the outlet, and then diverges more than if the vanes had been set straight.

Both the horizontal and vertical vanes of an outlet are important. After an installation has been made, many conditions of draftiness or stuffiness can be alleviated by some vane adjustment, provided an independent means for regulation of static pressure behind the vanes is included.

Vertical Drop and Rise

The distance that the lower edge of the air stream drops below the bottom of the outlet is important, since the air stream should not reach the occupied zone until the velocity has fallen to about 50 fpm. The drop (H, ft) is influenced by two forces; the natural vertical spread of the stream and the gravitational force due to the difference in density between supply air and

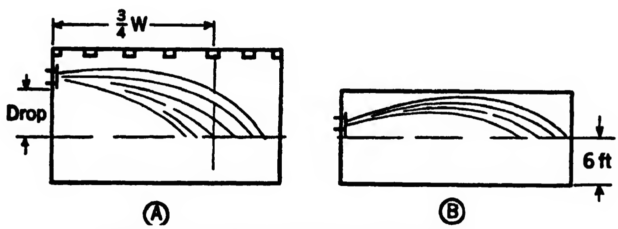


FIG. 5. THROW OF WALL OUTLETS

room air. For air emerging at room temperature, the drop will be a function of the spread only and will be equal to:

$$H_1 = L \times \tan\left(\frac{\text{Spread Angle}}{2}\right) \qquad (11)$$

where

H_1 = drop due to spread (when emerging air and room temperature are the same), feet.

L = throw, feet.

When there is a temperature difference between the air stream and the room, there is an *additional* drop which is approximately[4]:

$$H_2 = \frac{n_1(t_r - t_{as})L^{n_2}}{V_1} \qquad (12)$$

where

H_2 = additional drop due to temperature difference, feet.

n_1 and n_2 = constants (tentative suggested values $n_1 = 5$, $n_2 = 1.2$).

t_r = room temperature, degrees Fahrenheit.

t_{as} = supply air temperature, degrees Fahrenheit.

V_1 = jet velocity, feet per minute.

It should be remembered, that the total drop $H = H_1 + H_2$. H_1 is positive for either heating or cooling; H_2 is positive for cooling, negative for heating. In consequence, there will always be vertical drop in cooling, and a vertical rise in heating only if $H_2 > H_1$.

Another empirical equation for the *total drop* is[2]:

$$H = \frac{m(t_r - t_{sa})L}{V_1} \tag{13}$$

where

m = constant (tentatively suggested value of m = 16).

In other words, for a given throw L the drop or rise increases as the temperature difference increases and the outlet velocity decreases. This equation is only valid, if a temperature difference exists between room air and supply air.

Room Air Motion (Wall Outlet)

One of the most important problems in air distribution is to achieve air motion in the occupied zone within acceptable velocity limits. Therefore, outlet performance and characteristics of the space have to be related to this air motion.

The air moving in the occupied zone is (for a side wall outlet) equal in quantity to the total air contained in the outlet stream at the end of the throw and it is generally moving in a direction opposite to the stream. Assuming that the maximum volume of air is in circulation when the air stream velocity V_3 drops to 200 fpm, that the free area for return flow is 0.6 of the area of the wall in which the outlets are located, then, according to the momentum theory[2,4]:

$$V = \frac{Q_3}{0.6 \times A_w} \tag{14}$$

where

V = average room velocity, fpm.

Q_3 = volume of room air in motion, cfm.

A_w = area of wall in which outlet is located, square feet.

Since $Q_3 = Q_1 \times r$, (by definition); and $r = \frac{V_1}{V_3}$ according to Equation 3; the average room velocity is:

$$V = \frac{Q_1 r}{0.6A_w} = \frac{Q_1}{0.6A_w}\left(\frac{V_1}{V_3}\right)$$

or, with $V_3 = 200$ fpm

$$V = \frac{Q_1 V_1}{120A_w} \tag{15}$$

When Q_1, the volume of primary air, V_1, the velocity of primary air and A_w, the wall area are known, the average room velocity may be calculated from Equation 15 in order to determine the acceptability of the air distribution system.

OUTLET PERFORMANCE

The factors of outlet performance, throw, drop, room air motion, capacity, temperature differential, dirt and noise place considerable limitations on the design of a satisfactory distribution system.

1. *Throw.* The throw of a wall outlet must be sufficient to produce satisfactory conditions over the area to be conditioned. Underblowing may cause heated air to rise too rapidly above the occupied zone and thus create excessive vertical temperature variation (stratification); in cooling operation it may cause cold air to drop into the occupied zone before a satisfactory mixing of supply and room air has been accomplished by induction and thereby create a condition of acute discomfort (draft). On the other hand, overblowing will result in objectionable downdrafts from any surface the primary air stream may strike.

On the average it is considered most practicable to select throw as ¾ of the distance toward an exposed wall or window, as shown in A of Fig. 5. However, structural characteristics, mounting height, temperature differential and resultant drop or rise, or location of greatest heating or cooling loads strongly affect the selection of the optimum throw. In spaces with beamed ceilings, the outlets should be located below the bottom of the lowest beam level, and preferably low enough so that an upward or arched blow may be employed. The blow should be arched sufficiently to miss the beams and, at the same time, in such a manner as to prevent the primary or induced air stream from striking furniture and obstacles and producing objectionable drafts.

In the case of ceiling diffusers air is distributed with a spread of 360 deg horizontal. In addition there is a downward component of air motion. Therefore, both throw (radius of diffusion) and mounting height are important and interdependent factors. Due to the 360 deg spread of air diffusion the rate of induction will be higher and the throw shorter than that of a wall grille opening handling the same air quantity at the same outlet velocity. Therefore, ceiling diffusers will frequently permit the use of higher air velocities than wall outlets and consequently may be sized smaller to handle the same air volumes. If such ceiling outlets are installed flush with the ceiling, impingement of the air stream along the ceiling surface restricts induction of secondary air and the throw is increased approximately 20 per cent above that of an unrestricted air stream.

In the use of perforated ceiling plates as air distributing devices the term throw could hardly be applied in its proper meaning. Although this type of outlet can handle the greatest amount of air in proportion to room size, jet velocities must be kept low.

In all types of ceiling air distribution the following should be noted:

If cold air is used it must be brought to the proper temperature by mixing with room air before entering the zones of occupancy.

Air slightly above room temperature will usually be properly distributed by outlets selected for cooling.

When delivering warm air the same may be projected downward and the amount of dispersal of the jet varied to obtain proper mixing and control.

2. *Drop.* The outlets should be located so that the air stream at the termination of the blow is not less than 5 or 6 ft above the floor level. As illustrated in *B* of Fig. 5 the maximum permissible blow for a given ceiling height may be obtained by locating the outlet low on the wall, arching the blow, and sweeping the air across the flat ceiling. The air, as it traverses the room, will adhere to the ceiling. The objection to this method is the possible streaking of the ceiling with dirt.

3. *Room Air Motion.* Various features may cause room air motion to exceed acceptable standards. Some of these are: excessive air discharge

velocities; high air volume per cu ft of space (often referred to as number of air changes per hour); premature drop of cold air into the occupied zone; overblow causing spilling of high velocity air into the occupied zone; heating in severe climates by means of downward projection of hot air. It should be realized that these factors will not equally affect all types or designs of outlets at different temperature differentials, mounting heights, etc. For instance, certain outlets may safely handle more air per cubic foot of space at higher discharge velocities than others, and downward projection of supply air will sometimes not be considered excessive if the supply air temperature is substantially higher than the room temperature.

4. *Capacity.* The quantity of air to be handled is determined by the heating, cooling, or ventilating requirements. Manufacturers' rating sheets are usually consulted for selection of the proper number, size and type of outlets for a given air quantity. The basis of rating used should be carefully noted to make certain that resulting velocities are suitable for the application.

5. *Temperature Differential.* This is one of the most important factors affecting outlet performance. The quality of the temperature control, or the extent of the control problem, is directly a function of temperature difference. Obviously a system which carries under design conditions only a 5 deg difference between supply air stream and room temperature would require no control at all, for even a 50 per cent change in load could only effect a $2\frac{1}{2}$ deg change in room temperature under the worst conditions. Because of the self-equalizing nature of most load factors, even this extreme is never realized. It is obvious that the greater the temperature differential between supply air and room temperature, the greater will be the change in room temperature for a given change in load. The use of outlets that give rapid mixing permits greater temperature differentials. These principles apply in both heating and cooling practice.

6. *Dirt.* Although the primary air may be carefully filtered, small particles of dirt and dust will not be captured by mechanical filters and may finally be deposited on the walls or ceiling. With ceiling outlets, dirt streaking may be minimized by carefully controlling the discharge of the outlets. With wall outlets, dirt streaking may be minimized by preventing direct impingement of the air on any ceiling or room surface. Floor outlets may offer objection as dirt collectors.

7. *Noise.* The increase of noise level caused by an outlet is primarily a function of its air discharge velicity and its size. The maximum acceptable noise level in a space may dictate completely the selection of the permissible outlet velocity. In addition, however, noise may be caused by excessive restriction of free outlet area due to outlet design; by unnecessary turbulence due to one sided air flow through the outlet; or by the impingement of high velocity air on sharp edges. Such high frequency noises due to excessive turbulence are especially annoying (see Chapter 42 for discussion of permissible room noise levels and noise generation by outlets).

TYPES OF AIR OUTLETS

Two types of air supply outlets are commonly used; side wall and ceiling. A variety of designs has been developed for both types and the final selection depends to a large degree upon the specific problems arising in the air distribution system to be used.

In addition to the comments on use and application of outlets which fol-

low, reference should also be made to sections of this chapter on Outlet Location and Selection as well as on Specific Applications.

Wall Outlets

Wall type openings in general use are: perforated grilles, vaned outlets, registers, slotted outlets, ejector nozzles, and wall diffusers.

1. *Perforated Grilles.* Due to the non-adjustibility and small vane ratio these outlets, although inexpensive, have not met with favor as wall type supply openings. They are useful primarily where directional air control is unnecessary, and for return air intakes.

2. *Vaned Outlets.* Outlets equipped with either vertical and horizontal adjustable vanes or both are particularly suited to sidewall distribution. For proper control over the air flow, the vane ratio should be from 1 to 2. Outlets with non-adjustable vanes may be employed but they should only be used where the performance is not critical or can be adequately predicted. Vanes should be properly designed to prevent an increase of noise above permissible level.

3. *Registers.* Perforated grilles or vaned outlets equipped with a vane damper are termed registers. They are used primarily for residential heating systems, where the outlet distribution is not critical and low cost is of importance.

4. *Slotted Outlets.* Slotted outlets essentially consist of either flat steel plates containing a number of long narrow slots or a single long narrow slot. In order to give a good conversion from static pressure to velocity pressure, the sides of the slots are rounded to give a venturi effect. Due to their high aspect ratio, the slotted outlets have a greater induction effect than the comparable vaned outlets of equal area and consequently the throw is reduced. They are primarily useful where an unobtrusive means of distribution is desired, and where it is desirable to submerge the outlets into the room decoration and to minimize the effect of obstructions in the line of discharge. They are adaptable to narrow rooms having low ceilings. In this case the slots should extend the full length of the room. In all applications air quantity and distribution must be carefully planned as correction after installation is difficult.

5. *Ejector Nozzles.* These are outlets operating at high static pressure. They give a high conversion from static in the duct to velocity pressure in the outlet, and have a high induction effect due to their high outlet velocity. They are chiefly used for long throw and industrial process installations, such as drying, freezing, cooking, etc. Another type of ejector is sometimes referred to as a *louver nozzle* having a 45 to 90 deg elbow, which can be rotated similarly to a universal joint about an axis perpendicular to the surface to which it is fastened. These outlets give a considerable degree of adjustability and are, therefore, desirable for use in confined spaces where spot cooling is employed. The use of very high velocities is gradually disappearing due to noise difficulties.

6. *Wall Diffusers.* These outlets incorporate design features originally developed for ceiling outlets and use therefore semi-conical or semi-pyramidal guide vanes instead of the straight vanes of the conventional side wall outlet.

Ceiling Outlets

Generally used ceiling outlets are: plaques, ceiling diffusers, and perforated ceilings and panels; a discussion of each follows.

1. *Plaques.* Plaques are of simple design. The air from the supply opening impinges on a plate, which permits the air to be discharged horizontally in all directions. Plaques, although inexpensive, are difficult to control and are not generally satisfactory. In certain applications a properly designed plaque yields satisfactory results.

2. *Ceiling Diffusers.* Ceiling diffusers are round or rectangular outlets installed on or parallel to the ceiling, discharging supply air in a variety of directions and planes. Performance of the different designs varies according to principle employed. Some have no internal induction, but hasten external induction by supplying air in multiple layers. Others have internal induction and distribute air over an entire half sphere. The induction effect is greatest in the direction of the axis of the outlet, and least in the plane perpendicular to the axis and located at the ceiling level. Thus the induction is greatest in the vertical direction where the least throw can be tolerated and least in the horizontal plane at the ceiling where the greatest blow is both desired and permissible.

3. *Perforated Ceilings and Perforated Panels.* This method obtains air diffusion by discharging air through perforations in the ceiling, or part of the ceiling or walls. Some perforated panels feature a control plate frame which is inserted in the conventional ceiling duct. Supply air enters the plenum above the distribution plates through an adjustable air valve which can be set for varying air quantities and velocities. The advantages are unobtrusive appearance and the ready application of sound absorbing material to the design. Also, if designed properly, this system provides a low rate of room air motion and consequently lends itself to applications having high load or high ventilating requirements. The perforations should be kept free of accumulations of dirt, as clogging will cause uneven distribution and result in smudging of the ceiling. Best results are obtained in systems having efficient cleaning devices.

In present practice relatively low velocities are used because the perforated material offers only small resistance to the air flow. Therefore, great care must be taken to distribute the primary air at uniform velocities over the perforated panels to avoid uneven air distribution and primary air streams of undesirable velocities and direction.

OUTLET LOCATION AND SELECTION

In selecting the location of outlets, consideration must be given to the factors of physical construction, physical appearance, location of heating or cooling loads, and outlet performance:

1. The *physical construction* of a building, particularly of old buildings, immediately places limitations on the type of distribution system which can be employed. The first factor in the selection of outlet locations, therefore, is a consideration of the possible location of the supply duct, that is, whether it is above the ceiling, within the walls, through furred spaces above corridors, or in the conditioned space, etc. A particular method of distribution may be highly desirable but its execution, due to the location of beams and masonry walls, may be impossible.

2. The *physical appearance* of the outlets should conform to the esthetic appearance of the room. In factories, warehouses, etc., the esthetic demand may not be high; however, in department stores, clubs, theaters, etc., the location of the grilles may be dictated largely by such demands.

3. The *location of heating or cooling loads* in a room dictates to a great extent the general location of the outlets. The outlets should be located

to neutralize any undesirable cold drafts or radiation effects set up by a concentration of the heating or cooling load. The problem can be divided into *natural loads* due to outside weather and *internal heat loads*.

In winter the natural or primary heating load is caused by exposed walls, windows and skylights. Heat is lost primarily through convection to these exposed surfaces. The convection currents or cold drafts drop down the exposed surfaces and seriously impair the comfort conditions in the room, particularly at the floor level near the exposed surfaces. The outlets should be located to counteract these down drafts. Methods which may be employed are:

a. Direct counteraction of convection currents from cold surfaces can be obtained by locating the outlets to blow upward from beneath windows or exposed walls or to blow across the exposed wall. This method is desirable in small offices or bedrooms, or any location where people are seated or working near exposed surfaces. In northern climates, where the outside temperature may be constantly below 40 F, and the construction consists of uninsulated walls and single glass, this method of distribution is particularly useful for the maintenance of comfort requirements.

b. High induction by ceiling or wall outlets may be employed to nullify the convection currents from exposed surfaces. If outside temperatures are consistently below 40 F, and the exposed surfaces are not well insulated, the induction effort required for neutralization of the downdrafts is so great that the air motion in the room may exceed comfort limits unless care is taken in selection and location of the outlet. Where comfort conditions are not critical as in factories for heavy manufacturing, warehouses, etc., satisfactory results can be obtained even in cold climates. For uninsulated walls and glass areas some supplementary heating is often valuable. Wall diffusers, direct radiation or warm panels will satisfy these requirements for supplementary heating.

c. The location of exhaust or recirculated air openings at the base of large areas of glass is sometimes effective in reducing cold downdraft into the occupied space

If a *concentrated source of heat* creating an *internal heat load* is located at the occupancy level of the room, the heating effect may be counteracted by blowing the supply air toward the heat source or by locating an exhaust or return grille adjacent to the heat source. The latter method will prove more economical, as heat will be withdrawn at its source rather than be dissipated into the conditioned space. Where a lighting load is particularly heavy (five watts per square foot) and located high in a conditioned space, it may be economically desirable to locate the outlets below the lighting load. Warm air from the lights will stratify near the ceiling and can be removed by an exhaust or return fan, the former being advisable if the wet-bulb temperature of the air is above the outside temperature, and the latter being preferable if the wet-bulb temperature is below that of the outside air. Either method reduces the requirements for supply air. If the lamps are exposed, less saving can be realized than if enclosed, as a considerable portion of the total energy is radiant.

4. *Outlet Performance.* The laws of air distribution, previously discussed, will be found to exercise an important influence upon the design of an acceptable distribution system. This applies particularly to such features as throw, drop, capacity and room air motion.

Procedure for Outlet Location and Selection

In determining outlet location and selecting the type of outlets it is customary to proceed as follows:

1. Study the plan of the building and note the amount of air to be supplied to each enclosure.

2. Select number of outlets for each enclosure considering air quantity required and distance available for throw or as radius of diffusion. The same factors, as well as distance from floor level available as mounting height, structural characteristics of the space and frequently consideration of appearance will determine the type of outlet used.

3. Arrange location of outlets in space. Usually the outlets will be evenly spaced to distribute air uniformly throughout the enclosure. Sometimes, however, more air should be supplied and directed towards zones of exceptional heating or cooling loads. An important point to consider is the combination of proper outlet location and efficient duct design (see Chapter 41). Consult manufacturers' tables for recommended location and spacing of outlets.

4. Select size of outlets according to air quantity handled, permissible throat or discharge velocities or effective throw, taking into consideration other factors such

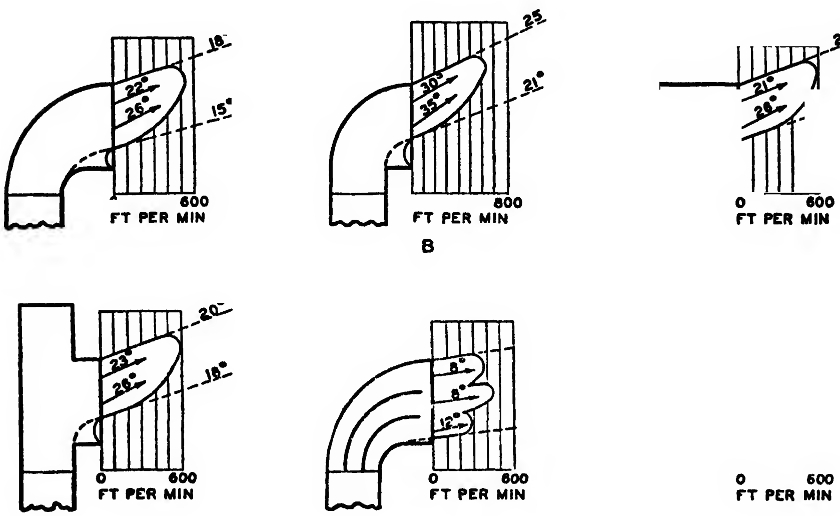


Fig. 6. Outlet Velocity and Air Direction Diagrams for Stack Heads with Expanding Outlets

Stack 14 in. x 6 in. Outlets 14 in. x 9 in. Stack Velocity 500 fpm

A. Rounded Throat and Round Back.
B. Square Throat and Round Back.
C. Square Throat and Back.
D. Square Throat and Cushion Chamber.
E. Rounded Throat and Back and 2 Splitters.
F. Square Throat and Back and 6 Guide Vanes.

as noise level, static pressure resistance, etc. It will be generally found that most selection tables for grille type outlets are based on capacity and throw, whereas data for ceiling or wall diffusers are usually based upon capacity and permissible outlet velocity. Choice and arrangement of either type of outlet should however satisfy the requirements of all aspects of air distribution. Therefore, type, location and size of any outlet should be checked against manufacturers' ratings to determine whether the selection made would satisfy the requirements of the job. The most important questions to be considered are:

a. Can drafts occur because of divergence between rated throw (radius of diffusion) and distance between outlet and nearest obstacle of air stream (wall, beam, pillar, ledge, etc.)?
b. Can drafts occur because of excessive cooling temperature differential and too low mounting height of the outlet?
c. Can drafts occur because of too low velocity causing a drop in cooling installations?
d. Will the outlet operate at too high a velocity and thereby cause an excessive increase in noise level?
e. Will the outlet operate against an excessive static pressure resistance?

Balancing the System

In designing an air conditioning system it should be the aim of the engineer to size ducts and outlets in such a manner that proper distribution of supply air takes place. In practice, however, this is almost impossible and therefore additional means for regulating air distribution are required to *balance* the system. Some of these means are:

1. Reducing the effective area of some supply openings by blank-offs.
2. Placing dampers in the supply and return (exhaust) openings.
3. Placing dampers in the supply and return (exhaust) ducts.
4. Using combinations of dampers in both supply and return (exhaust) ducts.

In selecting the desired type of damper or balancing method the following points should be kept in mind:

1. Unfavorable effect on air stream and noise level should be avoided. This will often eliminate blank-offs and dampers installed in the supply and return (exhaust) openings, unless such dampers are of special design.
2. It should be possible to alter the volume control setting and measure the amount

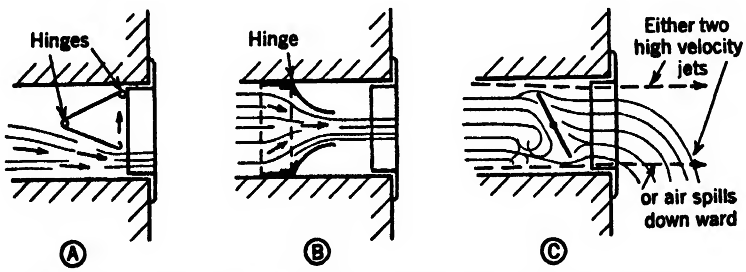


Fig. 7. Effect of Various Damper Arrangements Designed for Straight Blow

of air handled without difficulty. This will be particularly difficult to achieve in the case of blank-offs.

Generally speaking, it is most satisfactory to install dampers in the supply duct at some distance back of the outlets, so as to avoid disturbing the air flow. Dampers in both supply and return air ducts form the most flexible means of controlling supply of air to the room and static pressure within the room. Means of volume and directional control are discussed in detail in a following section of this chapter. Many types of air distribution control devices are now commercially available.

DIRECTIONAL AND VOLUME CONTROL

Duct Approaches to Outlets

In order to obtain proper direction of flow and distribution of air from outlets it is necessary that the air stream approaching the outlet be of uniform velocity over the entire connection to duct and perpendicular to the face.

Grilles and directional outlets cannot compensate for improper approach. Any attempt to secure a low face velocity and a high duct velocity by constructing an expanding chamber directly behind the grille is likely to be unsuccessful because the enlargement angle in even a straight duct cannot be

TABLE 1. RECOMMENDED RETURN INTAKE FACE VELOCITIES

INTAKE LOCATION	VELOCITY OVER GROSS AREA FPM
Above occupied zone	800 up
Within occupied zone, not near seats	600–800
Within occupied zone, near seats	400–600
Door or wall louvers	500–700
Undercutting of doors (through undercut area)	600

greater than 7 deg at each side if the stream is to fill the outlet without turbulence.

In elbow outlets or stack heads at the top of vertical stacks it is necessary to provide splitters or guide vanes in the elbows regardless of the shape of the elbows whether of rounded, square or expanding types. Cushion chambers at the top of the stack heads have no beneficial effect. The direction of flow, distribution and velocity (measured 12 in. from outlet) of the air, based on tests[5], are shown in Fig. 6 for various types of stack heads expanding from a 14 in. x 6 in. stack to 14 in. x 9 in. outlets, without grilles. The air velocity for each was 500 fpm in the stack below the elbow, but the direction of flow and the distribution patterns are generally indicative of performance obtainable with non-expanding elbows of similar shapes for a range of velocities 200 to 1400 fpm. Some of the conclusions drawn from the tests were:

1. Experiments with various elbow outlets on the 14 in. x 6 in. vertical stack[5] with stack air velocities of 200 to 1400 fpm indicated that enlargement of the outlet area, whether used in connection with square or rounded elbows, would not reduce either the angle of discharge (which was 20 to 30 deg above the horizontal) or the outlet velocity. The effect of the enlargement of the outlet was mainly to increase the reverse flow area in the lower part of the outlet, but in each case enlargement of the outlet reduced the static pressure in the duct below the elbow.

2. Splitters in the elbows had the effect of dividing the air stream into a number of streams flowing through rounded elbows and therefore lowered the angle of discharge, reduced or eliminated the reverse flow area, and made the outlet velocity quite uniform.

3. Turning vanes having 2 in. inner and 1 in. outer radii located in the center of the elbow were found most effective in improving performance in regard to angle of discharge, outlet velocity, and elimination of reverse flow area.

4. Pressure loss through stack heads may be reduced by use of splitters or turning vanes or by increasing the inner radius of an elbow. Considering the sum of the velocity and static pressure as a measure of the energy required to change the direction of the air stream and to deliver the air into the atmosphere, and considering the energy required for a plain fitting as 100 per cent, it was found that *turning vanes* dropped the energy requirement of square type stack heads to 45 per cent. Splitters

TABLE 2. APPROXIMATE PRESSURE DROPS FOR LATTICE RETURN INTAKES
Inches Water Gage—Standard Air

PER CENT FREE AREA	FACE VELOCITY, FPM 400	500	600	700	800	900	1000
50	0.06	0.09	0.13	0.17	0.22	0.28	0.35
60	0.04	0.06	0.09	0.12	0.16	0.20	0.24
70	0.03	0.05	0.07	0.09	0.12	0.15	0.18
80	0.02	0.03	0.05	0.07	0.09	0.11	0.14

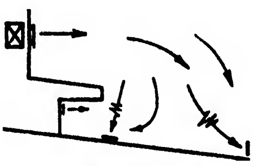
reduced the energy requirement to 90 per cent in long radius elbows and to 74 per cent in short radius turns. In expanding heads splitters reduced the energy requirement to 58 per cent.

Side Outlets in Horizontal Air Ducts

When air is supplied to a room from side outlets in horizontal ducts it is necessary to use directive devices within the duct at each outlet in order to obtain a uniform velocity of delivered air and to obtain a direction of flow perpendicular to the face of the outlet. In tests[6] conducted with 3 in. x 10 in., 4 in. x 9 in., and 6 in. x 6 in. outlets in a 6 in. x 20 in. horizontal duct at duct velocities of 200 to 1400 fpm (in the 6 in. x 20 in. section) it was found that multiple curved deflectors produced the best flow characteristics. Vertical guide strips in the outlet were not so effective as curved deflectors. A single scoop type deflector at the outlet did not improve the flow pattern obtained from a plain outlet and was therefore not found to be desirable.

Ceiling Outlets on Horizontal Ducts

Ceiling outlets are usually installed below horizontal supply ducts so that the supply air has to make a 90 deg turn before entering the outlet it-


Rear wall distribution

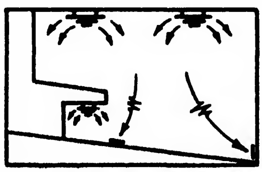

Ceiling distribution

Fig. 8. Air Distribution Methods for Theaters, Churches, and Auditoriums

self. The shorter the connection between bottom of duct and outlet, the greater is the need for directive devices to obtain uniformity of flow. Generally speaking, conditions and remedy in such cases strongly resemble those for side outlets in horizontal air ducts. Ceiling ducts often have a rectangular cross section while the connections to the ceiling outlets are circular. It will then be quite difficult to install turning vanes successfully, particularly if the ducts are shallow and the connection areas are comparatively large. This will be the case, when more than one outlet is installed on one duct run and restrictions of duct area must be avoided. In such cases good results have been obtained by using a series of vertical guide strips, installed at right angles to the direction of air approach in the outlet connection where it leaves the horizontal air duct.

Volume Control

Various methods are used to regulate volume of supply and return (exhaust) air. Some of these accomplish only minor changes in volume; most of them however permit a range of adjustment from maximum air supply to complete shut-off.

When selecting type and location of such dampers, the following points must be considered, especially when the volume control feature is to be ocated near the air outlet itself: (1) deflection of air stream by the damper,

need and feasibility of directional control; (2) increase of noise level due to irregular and localized high air velocities caused by damper operation.

The following types of volume control are most frequently encountered:

1. *Slide Damper.* A single plate which can be pushed across the duct. Since its operation changes the free area of air passage in a one sided manner it should not be located near any air outlet and is practicable only where no intermediate setting between full open and closed is required.

2. *Hit-and-miss Damper.* Two slotted plates or discs, closely adjacent; by moving one of the two plates the respective slots may be opened or closed. This type of volume control may be installed close to an air outlet and it is easy to operate, but its main disadvantage is that even in the open position the air passage area is blocked by at least 50 per cent. This requires oversizing of the air outlet in order to avoid excessive increase of noise level.

3. *Splitter Damper.* A single blade sheet metal plate hinged at one edge, usually located at the branch connection of a duct or outlet. It is easy to operate but often causes irregular air flow in the duct. When used in connection with, and near an outlet additional directional control is required.

4. *Butterfly Damper.* A single blade sheet metal plate hinged in the middle, usually located in a straight duct run. It is easier to handle than a splitter damper, since only half the motion is necessary to change its setting. However, if located too close to an air outlet, it is objectionable because its operation frequently results in a condition whereby two high velocity jets are created along the sides of the duct or the air spills immediately downward into the occupied zone (see C Fig. 7).

5. *Louver Dampers.* Numerous designs have been developed incorporating a series of splitter or butterfly dampers across the duct or air outlet. Their main advantage consists in retaining greater uniformity of air flow and in requiring less depth for installation. Some designs provide for louver blades moving in opposite directions, and while decreasing free air passage area, retain a constant air flow direction along the axis of the duct air outlet connection (see A and B in Fig. 7).

RETURN AND EXHAUST INTAKES

The *selection of return and exhaust intakes* depends on: (1) velocity in occupied zone near intake, (2) permissible pressure drop through intake, and (3) noise.

1. *Velocity.* The effect of air flow through return intakes upon air movement in the room is slight. Air handled by the intake is drawn from all directions, the velocity dropping off rapidly as distance from intake increases. The only locality where drafts may prove objectionable is adjacent to the intake. To prevent excessive air motion in this area due to the return intake, it is advisable to compute the total air motion toward the exhaust opening as outlined in Equation 14 where A is the exhaust wall area in square feet. Recommended return intake face velocities are given in Table 1.

2. *Permissible Pressure Drop.* The permissible pressure drop will depend on the choice of the designer. Table 2 gives pressure drop through plain lattice intakes as a function of free area and face velocity.

Proper pressure drop allowance should be made for control or directive devices.

3. *Noise.* The problem of *noise* generated by return intakes is the same as that for supply outlets. In computing resultant room noise levels from the operation of an air conditioning system, the return intake must be included as a part of the total grille area. The major difference between the supply outlets and return intakes is the frequent installation of the latter at ear level. When so located, it is recommended that the return intake

velocity be not in excess of 75 per cent of the maximum permissible outlet velocity.

Outlet Location

The control of the room air motion for the maintenance of comfort conditions depends on the proper selection of the *supply* outlets. The location of the return or exhaust intakes does not critically affect air motion unless room air velocities in the occupied zone adjacent to the intake exceed comfort limits. The locations of return or exhaust intakes are however important for obtaining the desired room temperature equalization.

Ceiling locations for outlets are recommended for bars, kitchens, lavatories, dining rooms, club rooms, etc., where warm air will rise to the ceiling level. In heating installations, location of the return grilles in the ceiling or high on the wall will result in stratification of the conditioned air and a high percentage of the heated air will be drawn into the return duct before it has served its purpose. (Refer also to considerations outlined previously in section *Outlet Location and Selection* in this Chapter.)

Some circular ceiling outlets combine the supply and return openings in a single unit. The return duct is in the center with the supply pattern on the

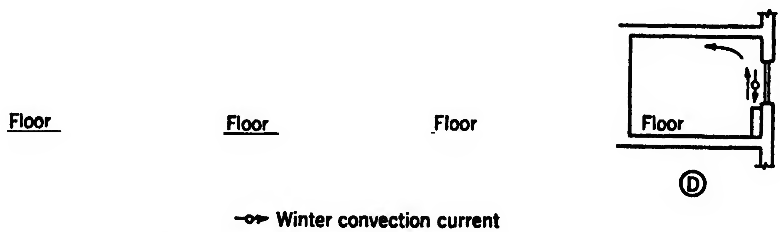


Fig. 9. Distribution Methods for Small Rooms

A. Satisfactory for cooling. Unsatisfactory for heating in severe climates where the outside temperature is consistently below 40 F. and single glass and uninsulated walls are prevalent.

B. Performance approximately that of A when small diffusers are used in bottom of the duct.

C. Satisfactory for cooling. Satisfactory for heating if direct radiation is properly controlled.

D. Satisfactory for both cooling and heating. The air should be discharged slightly away from the wall and for low velocities should be fanned out parallel to the wall.

outside. This method gives best results for cooling applications. The application for heating is more critical and requires consideration of ceiling height, amount of outside wall area, and number of air changes required. In some cases, stratification of warm air may cause short circuiting. Where the wall losses are a small part of the total, little difficulty is encountered with stratification.

Floor locations of outlets are used in heating installations for ceiling or side wall supply. When located so that air is drawn across exposed walls, the performance of the system may be somewhat improved. In general, floor locations tend to collect dirt and refuse.

Wall and door locations of outlets depending on their elevation, have the characteristics of either floor or ceiling returns. In large buildings with many small rooms, the return air may be brought through door grilles or door undercuts into the corridors and then to a common return or exhaust. The pressure drop through door returns should not be excessive; otherwise

the air distribution to the room may be seriously unbalanced with the opening or closing of the doors. Outward leakage through doors or windows cannot be counted upon for dependable results.

SPECIFIC APPLICATIONS

For *theaters and auditoriums* the air distribution methods used are the downward distribution system with ceiling diffusers and the horizontal distribution system with ejector nozzles or wall diffusers. Fig. 8 shows both methods. Ceiling distribution is accomplished by ceiling outlets under main ceiling and balcony It is indicated when main ceiling or balcony are cut up by architectural treatment or beams. The only critical points are under the balcony, and (occasionally) above the very rear of the balcony, where ceiling heights are low and where direct impingement of air is sometimes a hazard.

Wall or ejector distribution is particularly applicable for relatively long and narrow theaters. It is essential with this type of distribution that there be no interference with the movement of air throughout its entire path from the high velocity nozzles to the front of the theater. The ceiling should be

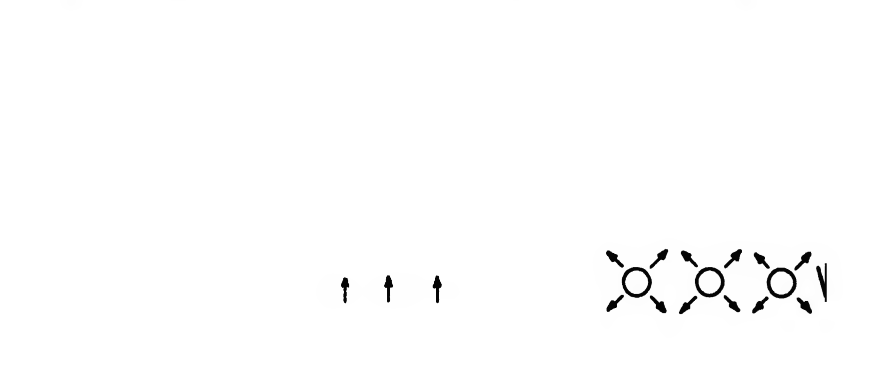


Fig. 10. Small Store Cooling Distribution

A. *Rear Wall.* High outlet velocity, satisfactory if properly designed; possibility of excessive air motion and drafts if used for wrong application.

B. *Front Wall.* High outlet velocity, results same as A.

C. *Front and Rear Walls.* Moderate room air motion, outlet blows should not impinge giving rise to down drafts in center.

D. *Center.* Moderate air motion, no impingement of air streams. Good results.

E. *One Side.* Moderate room air motion; should blow toward exposed wall. Good results.

F. *Ceiling.* Low room air motion. Good results. Outlets should be selected of sufficient size to allow for blocking when not located in perfect squares.

smooth without projecting beams or obstructing ornamentation. For large theaters, relatively high velocities can be used. These will work satisfactorily if adjustable outlets are used to avoid areas of local turbulence.

In small or medium size theaters, it is sometimes practicable to use side wall or front wall distribution. For the satisfactory operation of such a system during the winter heating period, the returns should preferably be located at the floor level and near the front of the theater to prevent cold spots which may result from exposed wall convection or infiltration from exists.

For *multi-room buildings* diagrams shown in Fig. 9 illustrate distribution

methods for small rooms with exposed wall, such as offices, hotel (guest) rooms, hospital (patients) rooms, apartments, etc.

For *a small store* the cooling performance of various distribution methods is illustrated in Fig. 10. *Marine applications* of air distribution systems are given in Chapter 49.

REFERENCES

[1] Control of Air Streams in Large Spaces, by G. L. Tuve and G. B. Priester (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 153).

[2] Air Conditioning Principles, by C. O. Mackey, (International Textbook Company, Scranton, Pa., 1941) p. 187.

[3] Throw of Air from Slots and Jets, by R. D. Madison and W. R. Elliot (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Nov. 1946, p. 108).

[4] Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne, and W. A. Grant (Pitman Publishing Corp., New York, N. Y., 1940).

[5] A.S.H.V.E. RESEARCH REPORT No. 1155—The Performance of Stack Heads, by D. W. Nelson, D. H. Krans and A. F. Tuthill (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 205).

[6] A.S.H.V.E. RESEARCH REPORT No. 1226—Performance of Side Outlets on Horizontal Ducts, by D. W. Nelson and G. E. Smedberg (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 58).

BIBLIOGRAPHY

A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645).

A.S.H.V.E. RESEARCH REPORT No. 1162—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 313).

A.S.H.V.E. RESEARCH REPORT No. 1165—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 351).

A.S.H.V.E. RESEARCH REPORT No. 1204—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 241).

Air Flow at Discharge of Fan Pipe Lines in Mines, by G. E. McElroy, (*Bureau of Mines*, Report of Investigations—No. 3730, November, 1943).

The Rationale of Air Distribution and Grille Performance, by C. O. Mackey (*Refrigerating Engineering*, Vol. 35, No. 6, June 1938, p. 417).

CHAPTER 41

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Circular Equivalents of Rectangular Ducts, Dynamic or Shock Losses, Pressure Loss in Elbows, Losses Due to Area Changes, Pressure Changes, Duct Design, Duct Construction Details, Heat Losses from Ducts, Maintenance

AIR ducts for the transmission of the air in forced air heating, ventilating, cooling or air conditioning systems must be carefully designed for functional as well as economical reasons. The design should be based upon the fundamental laws of fluid flow in pipes and should take into account recent analytical and experimental studies which complement and substantiate the fundamental laws. The basic equations of the flow of fluids will be found in Chapter 4, Fluid Flow.

PRESSURE LOSSES

Air ducts impose resistances to air flow which must be overcome by pressure differences resulting from the expenditure of energy in maintaining the flow. In ventilating and air conditioning work the flow of air takes place under very small pressure differences, and the assumption that the gas density remains constant throughout the flow will cause only a negligible error. It is therefore possible to use the equation for incompressible fluids (liquids) for the flow of air in a duct, instead of the complicated thermodynamic formulas for air discharge under conditions of adiabatic flow which would be necessary for large pressure differences.

A reasonably accurate estimate of the flow resistances offered by the system is essential for satisfactory duct design. The theoretical resistance of an air handling system can be computed from the methods and data given in this chapter. The actual resistance for any given installation, however, may vary considerably from the calculated resistance because of variation in the smoothness of materials, the type of joints used and the ability of the workmen to manufacture the system in accordance with the design. It is best to select fans and motors of sufficient size to allow a factor of safety. Dampers should be installed in each branch outlet to balance the system, and the necessary allowance for this balancing should be made in calculating the pressure loss in the system.

The drop in pressure in air transmission systems is due to *friction* losses and *dynamic* (shock) losses. Pressure increases and decreases may also be effected by changes in duct areas resulting in the conversion of velocity pressure to static pressure, and vice versa. The friction losses for turbulent flow (which occur in all practical air flow problems) are due to the friction of air against the sides of the duct and to internal friction between the molecules. The dynamic losses are caused by changes in the direction or in the velocity of air flow, such as caused by changes in size and shape of the cross-section of the duct, bends (elbows) and to obstructions to flow offered by dampers.

FRICTION LOSSES

Pressure drop in a *straight* duct is caused by surface friction and this friction loss is most readily calculated by means of the Air Friction Chart

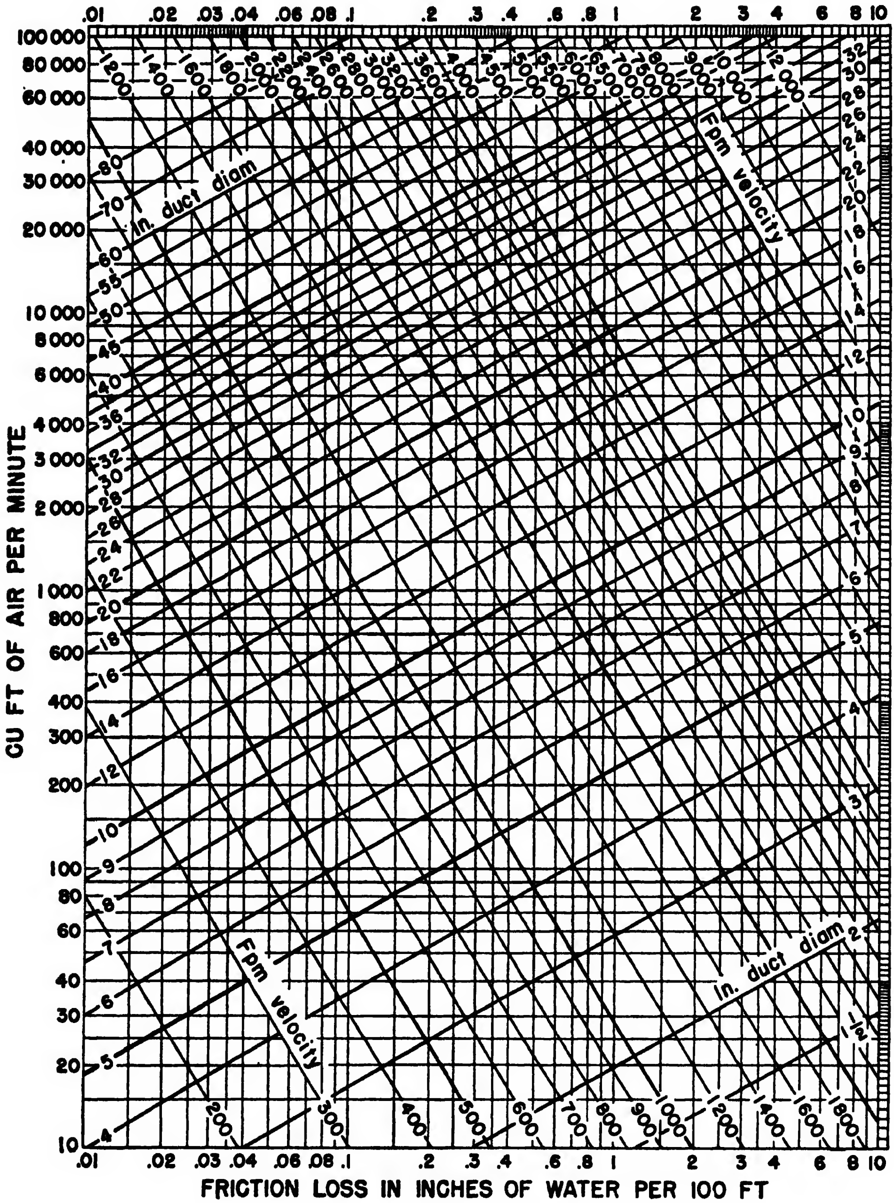


Fig. 1. Friction of Air in Straight Ducts

(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.) No safety factor included. Caution: Do not extrapolate below chart.

which was developed by the A.S.H.V.E. Research Laboratory[1]. The range of the chart has recently been modified to include smaller air capacities and duct sizes, and the upper limit of air capacity has been somewhat reduced.

The chart, Fig. 1, is constructed from the basic flow equation for the pressure loss in circular ducts (see Chapter 4):

$$. \quad .\, l\ V^2$$

where

h_f = head loss due to friction, in feet of fluid flowing.
l = length of conduit, feet.
D = inside diameter of conduit, feet.
V = fluid velocity, feet per second.
g = acceleration due to gravity, 32.17 fps per second.
f = a non-dimensional friction coefficient, which for ventilation work depends upon Reynolds Number and the relative roughness of the conduit. Appropriate values of f were taken from the work of Moody.[2] See Chapter 4, Fig. 4, Relation Between Friction Factor and Reynolds Number.

The air friction chart is based on standard air[3] with a density of 0.075 lb per cubic foot, flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft. Fig. 1 should not be used to obtain values below the chart by extrapolation because critical flow would occur in this region and values so obtained would be unreliable. For the average application, it should prove sufficiently accurate, without corrections, for any air temperature from 50 F to 90 F, for any relative humidity, and for any normal variation in barometric pressure. For widely varying air pressures or temperatures, or for unusual duct conditions, the friction values obtained from the chart should be corrected.[4]

For ordinary ventilating work, friction may be assumed to vary directly as the density without serious error, and therefore

where

h_o = friction loss under actual operating conditions, any consistent units
h_s = friction loss under standard conditions, any consistent units
ρ_o = density of air under actual operating conditions, any consistent units
ρ_s = density of air under standard conditions, any consistent units.

For ducts of other than standard sheet metal construction, correction factors may be obtained from Fig. 2[4]. The correction factors shown in Fig. 2 were computed for the values of ϵ, the roughness in feet, shown in Table 1[4]. The correct friction loss for such ducts may then be determined by multiplying the losses obtained from Fig. 1 by these factors.

Examples 1 and 2 illustrate the use of Fig. 1 to determine friction loss and the use of Fig. 2 to apply a correction for roughness.

Example 1. Assume that it is desired to circulate 10,000 cfm of air through 75 ft of 24 in. diameter galvanized duct. Find 10,000 cfm on the left scale of Fig. 1 and move horizontally right to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.50 in.; then for 75 ft the friction will be 0.75 ×

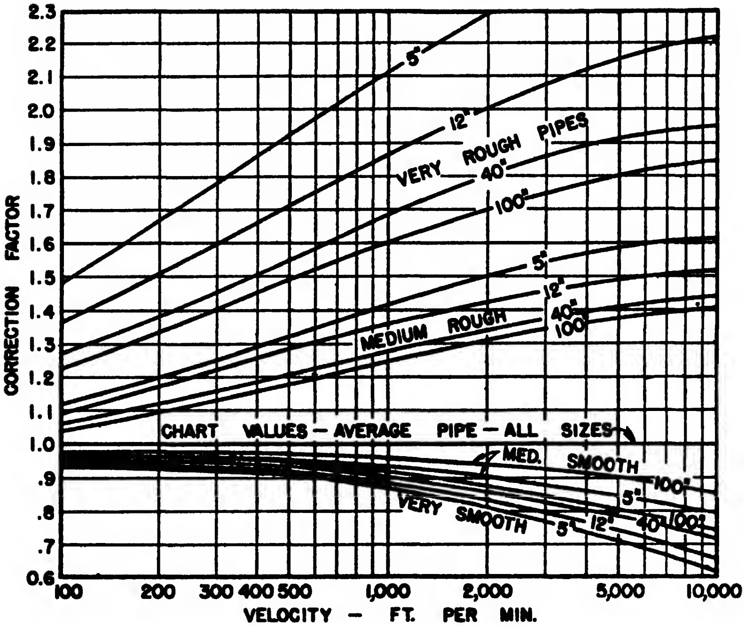


FIG. 2. CORRECTION FACTORS FOR PIPE ROUGHNESS

To correct for pipe roughness multiply friction loss obtained from Fig. 1 by correction factor obtained from Fig. 2.

0.50 = 0.38 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Example 2. If the duct in Example 1 is very rough, instead of galvanized, with 40 joints per 100 ft, find the total friction.

On Fig. 2 find (by interpolation between 12 in. and 40 in. pipe) the intersection of the 24 in. *very rough pipe* line and the 3200 fpm velocity ordinate and at the left margin read a correction factor of 2. The friction loss in the rough duct is therefore $2 \times 0.38 = 0.76$ in.

TABLE 1. VALUES OF ROUGHNESS ϵ FOR DIFFERENT PIPES[a]

PIPE	DEGREE OF ROUGHNESS	ϵ ROUGHNESS IN FEET
Drawn Tubing	Very smooth	0.0000015
New Steel or Wrought-Iron Pipe	Medium smooth	0.00015
Galvanized Iron	Average	0.0005
Average Concrete	Medium rough	0.003
Average Riveted Steel	Very rough	0.01

[a] Used in computing values for Fig. 2.

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS

An air handling system is usually sized first for round ducts and, if rectangular ducts are desired, their sizes are selected to provide air carrying capacities equivalent to those of the round ducts originally selected.

A recent comprehensive study of the A.S.H.V.E. Research Laboratory proved that for most practical purposes rectangular ducts of aspect ratios not exceeding 8:1 will have the same static friction pressure loss for equal lengths and mean velocities of flow as a circular duct of the same hydraulic diameter. When duct sizes are expressed in terms of hydraulic diameter and when equations for friction loss in round and rectangular ducts are equated for equal capacity and equal length, an equation giving the circular equivalent of a rectangular duct is obtained[5] (Equation 3).

$$D_e = 1.30 \, , \tag{3}$$

where

a = length of one side of rectangular duct, feet or inches. (Other side is b.)
b = length of one side of rectangular duct, feet or inches. (Other side is a.)
D_e = circular equivalent of a rectangular duct for equal friction and capacity, inches.

Table 2 gives the circular equivalents of rectangular ducts for equal friction and capacity for aspect ratios not greater than 11.7:1 based on Equation 3[5].

Multiplying or dividing the length of each side of a round duct by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24 in. duct is required, it will be twice that of 40 x 12 in. duct, or 2 x 23.0 = 46.0 in.

TABLE 2. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY
Dimensions in Inches

Side Rectangular Duct	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
3.0	3.8	4.0	4.2	4.4	4.6	4.8	4.9	5.1	5.2	5.4	5.5	5.6	5.7
3.5	4.1	4.3	4.6	4.8	5.0	5.2	5.3	5.5	5.7	5.8	6.0	6.1	6.3
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.3	6.4	6.6	6.8
4.5	4.6	4.9	5.2	5.4	5.6	5.9	6.1	6.3	6.5	6.7	6.9	7.0	7.2
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.1	7.3	7.4	7.6
5.5	5.1	5.4	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.4	7.6	7.8	8.0

Side Rectangular Duct	10.0	10.5	11.0	11.5	12.0	12.5	13.0	13.5	14.0	14.5	15.0	15.5	16.0
3.0	5.7	5.9	6.0	6.1	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0
3.5	6.3	6.4	6.5	6.7	6.8	6.9	7.0	7.1	7.2	7.3	7.4	7.5	7.6
4.0	6.8	6.9	7.1	7.2	7.3	7.5	7.6	7.7	7.8	7.9	8.1	8.2	8.3
4.5	7.2	7.4	7.5	7.7	7.8	8.0	8.1	8.2	8.4	8.5	8.6	8.7	8.9
5.0	7.6	7.8	8.0	8.1	8.3	8.4	8.6	8.7	8.9	9.0	9.1	9.3	9.4
5.5	8.0	8.2	8.4	8.6	8.7	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.8

Table 2. Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity (Continued)

Dimensions in Inches

Side Rectangular Duct	6	7	8	9	10	11	12	13	14	15	16	17	18	19
6	6.6													
7	7.1	7.7												
8	7.5	8.2	8.8											
9	8.0	8.6	9.3	9.9										
10	8.4	9.1	9.8	10.4	10.9									
11	8.8	9.5	10.2	10.8	11.4	12.0								
12	9.1	9.9	10.7	11.3	11.9	12.5	13.1							
13	9.5	10.3	11.1	11.8	12.4	13.0	13.6	14.2						
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3					
15	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4				
16	10.4	11.4	12.2	13.0	13.7	14.4	15.1	15.7	16.3	16.9	17.5			
17	10.7	11.7	12.5	13.4	14.1	14.9	15.5	16.1	16.8	17.4	18.0	18.6		
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.6	17.3	17.9	18.5	19.1	19.7	
19	11.2	12.2	13.2	14.1	14.9	15.6	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8
20	11.5	12.5	13.5	14.4	15.2	15.9	16.8	17.5	18.2	18.8	19.5	20.1	20.7	21.3
22	12.0	13.1	14.1	15.0	15.9	16.7	17.6	18.3	19.1	19.7	20.4	21.0	21.7	22.3
24	12.4	13.6	14.6	15.6	16.6	17.5	18.3	19.1	19.8	20.6	21.3	21.9	22.6	23.2
26	12.8	14.1	15.2	16.2	17.2	18.1	19.0	19.8	20.6	21.4	22.1	22.8	23.5	24.1
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.6	24.4	25.0
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.6	24.4	25.2	26.0	26.7
34	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	23.0	23.9	24.8	25.8	26.6	27.4	28.3
38	15.0	16.4	17.8	19.0	20.3	21.4	22.5	23.5	24.5	25.4	26.4	27.3	28.1	29.0
40	15.3	16.8	18.2	19.4	20.7	21.9	23.0	24.0	25.1	26.0	27.0	27.9	28.8	29.7
42	15.6	17.1	18.5	19.8	21.1	22.3	23.4	24.5	25.6	26.6	27.6	28.5	29.4	30.4
44	15.9	17.5	18.9	20.2	21.5	22.7	23.9	25.0	26.1	27.2	28.2	29.1	30.0	31.0
46	16.2	17.8	19.2	20.6	21.9	23.2	24.3	25.5	26.7	27.7	28.7	29.7	30.6	31.6
48	16.5	18.1	19.6	20.9	22.3	23.6	24.8	26.0	27.2	28.2	29.2	30.2	31.2	32.2
50	16.8	18.4	19.9	21.3	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8
52	17.0	18.7	20.2	21.6	23.1	24.4	25.6	26.8	28.1	29.2	30.3	31.4	32.4	33.4
54	17.3	19.0	20.5	22.0	23.4	24.8	26.1	27.3	28.5	29.7	30.8	31.9	32.9	33.9
56	17.6	19.3	20.9	22.4	23.8	25.2	26.5	27.7	28.9	30.1	31.2	32.4	33.4	34.5
58	17.8	19.5	21.1	22.7	24.2	25.5	26.9	28.2	29.3	30.5	31.7	32.9	33.9	35.0
60	18.1	19.8	21.4	23.0	24.5	25.8	27.3	28.7	29.8	31.0	32.2	33.4	34.5	35.5
62	18.3	20.1	21.7	23.3	24.8	26.2	27.6	29.0	30.2	31.4	32.6	33.8	35.0	36.0
64	18.6	20.3	22.0	23.6	25.2	26.5	27.9	29.3	30.6	31.8	33.1	34.2	35.5	36.5
66	18.8	20.6	22.3	23.9	25.5	26.9	28.3	29.7	31.0	32.2	33.5	34.7	35.9	37.0
68	19.0	20.8	22.5	23.2	25.8	27.3	28.7	30.1	31.4	32.6	33.9	35.1	36.3	37.5
70	19.2	21.	22.8	23.5	26.1	27.6	29.1	30.4	31.8	33.1	34.3	35.6	36.8	37.9

DYNAMIC (SHOCK) LOSSES

Any sudden change in the direction or magnitude of the velocity of the air flow causes a greater loss in pressure than would occur in steady flow through a straight duct of uniform cross-section and the same length. Although all shock losses may be considered as caused by changes in the

TABLE 2. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (CONCLUDED)

Dimensions in Inches

Side Rectangular Duct	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	Side Rectangular Duct
20	21.9																										20
22	22.9	24.1																									22
24	23.9	25.1	26.2																								24
26	24.8	26.1	27.2	28.4																							26
28	25.7	27.1	28.2	29.5	30.6																						28
30	26.7	28.0	29.3	30.5	31.6	32.8																					30
32	27.5	28.9	30.1	31.4	32.6	33.8	35.0																				32
34	28.3	29.7	31.0	32.3	33.6	34.8	36.0	37.2																			34
36	29.0	30.5	32.0	33.0	34.6	35.8	37.0	38.2	39.4																		36
38	29.8	31.4	32.8	34.2	35.5	36.7	38.0	39.2	40.4	41.6																	38
40	30.5	32.1	33.6	35.1	36.4	37.6	39.0	40.2	41.4	42.6	43.8																40
42	31.2	32.8	34.4	35.9	37.3	38.6	39.9	41.1	42.4	43.6	44.8	45.9															42
44	31.9	33.5	35.2	36.7	38.1	39.5	40.8	42.0	43.4	44.6	45.8	46.9	48.1														44
46	32.5	34.2	35.9	37.4	38.9	40.3	41.7	43.0	44.3	45.6	46.8	47.9	49.1	50.3													46
48	33.1	34.9	36.6	38.2	39.7	41.2	42.6	43.9	45.2	46.5	47.8	48.9	50.2	51.3	52.6												48
50	33.7	35.5	37.3	38.9	40.4	42.0	43.5	44.8	46.1	47.4	48.8	49.8	51.2	52.3	53.6	54.7											50
52	34.3	36.2	38.0	39.6	41.2	42.8	44.3	45.7	47.1	48.3	49.7	50.8	52.2	53.3	54.6	55.8	56.9										52
54	34.9	36.8	38.7	40.3	42.0	43.6	45.0	46.5	48.0	49.2	50.6	51.8	53.2	54.3	55.6	56.8	57.9										54
56	35.5	37.4	39.3	41.0	42.7	44.3	45.8	47.3	48.8	50.1	51.5	52.7	54.1	55.3	56.5	57.8	58.9	61.3									56
58	36.0	38.0	39.8	41.7	43.4	45.0	46.6	48.1	49.6	51.0	52.4	53.7	55.0	56.2	57.5	58.8	60.0	62.3									58
60	36.5	38.6	40.4	42.3	44.0	45.8	47.3	48.9	50.4	51.8	53.3	54.6	55.9	57.1	58.5	59.8	61.0	63.3	65.7								60
62	37.1	39.2	41.0	42.9	44.7	46.5	48.0	49.7	51.2	52.6	54.2	55.5	56.8	58.0	59.4	60.7	62.0	64.3	66.7								62
64	37.6	39.7	41.6	43.5	45.4	47.2	48.7	50.4	52.0	53.4	55.0	56.4	57.7	59.0	60.3	61.6	62.9	65.3	67.7	70.0							64
66	38.1	40.2	42.2	44.1	46.0	47.8	49.5	51.1	52.8	54.2	55.8	57.2	58.6	59.9	61.2	62.5	63.9	66.3	68.7	71.1							66
68	38.6	40.7	42.8	44.7	46.6	48.4	50.2	51.8	53.5	55.0	56.6	58.0	59.5	60.8	62.1	63.4	64.8	67.3	69.7	72.1	74.4						68
70	39.1	41.3	43.3	45.3	47.2	49.0	50.9	52.5	54.2	55.8	57.3	58.8	60.3	61.7	63.0	64.3	65.7	68.3	70.7	73.1	75.4						70
72	39.6	41.8	43.8	45.9	47.8	49.7	51.5	53.2	54.9	56.5	58.0	59.6	61.1	62.6	63.9	65.2	66.6	69.2	71.7	74.1	76.4	78.8					72
74	40.0	42.3	44.4	46.4	48.4	50.3	52.1	53.9	55.6	57.2	58.8	60.4	61.9	63.3	64.8	66.1	67.5	70.1	72.7	75.1	77.4	79.9					74
76	40.5	42.8	44.9	47.0	49.0	50.8	52.7	54.6	56.3	57.9	59.5	61.2	62.7	64.1	65.6	67.0	68.4	71.0	73.6	76.1	78.4	80.9	83.2				76
78	40.9	43.3	45.5	47.5	49.5	51.5	53.3	55.2	57.0	58.6	60.3	62.0	63.4	64.9	66.4	67.9	69.3	71.8	74.5	77.1	79.4	81.8	84.2				78
80	41.3	43.8	46.0	48.0	50.1	52.0	53.9	55.8	57.6	59.3	61.0	62.7	64.1	65.7	67.2	68.7	70.1	72.7	75.4	78.1	80.4	82.8	85.2	87.5			80
82	41.8	44.2	46.4	48.6	50.6	52.6	54.5	56.4	58.2	60.0	61.7	63.4	64.9	66.5	68.0	69.5	71.0	73.6	76.3	79.0	81.4	83.8	86.2	88.6			82
84	42.2	44.6	46.9	49.2	51.1	53.2	55.1	57.0	58.9	60.7	62.4	64.1	65.7	67.3	68.8	70.3	71.8	74.5	77.2	79.9	82.4	84.8	87.2	89.6	91.9		84
86	42.6	45.0	47.4	49.6	51.6	53.7	55.7	57.6	59.5	61.3	63.0	64.8	66.4	68.0	69.5	71.1	72.6	75.4	78.1	80.8	83.3	85.8	88.2	90.6	92.9		86
88	43.0	45.4	47.9	50.1	52.2	54.3	56.3	58.2	60.1	62.0	63.7	65.4	67.0	68.7	70.3	71.8	73.4	76.3	79.0	81.6	84.2	86.8	89.2	91.6	93.9	96.3	88
90	43.4	45.9	48.3	50.6	52.8	54.8	56.9	58.8	60.7	62.6	64.4	66.0	67.8	69.4	71.1	72.6	74.2	77.1	79.9	82.5	85.1	87.8	90.2	92.6	94.9	97.3	90
92	43.8	46.3	48.7	51.1	53.4	55.4	57.4	59.4	61.3	63.2	56.0	66.8	68.5	70.1	71.8	73.3	74.9	77.8	80.8	83.4	86.0	88.7	91.2	93.6	95.9	98.3	92
94	44.2	46.7	49.1	51.6	53.9	55.9	57.9	60.0	61.9	63.8	65.6	67.5	69.2	70.8	72.5	74.1	75.6	78.6	81.7	84.3	86.9	89.6	92.1	94.6	96.9	99.3	94
96	44.6	47.2	49.5	52.0	54.4	56.3	58.4	60.5	62.4	64.4	66.2	68.2	69.8	71.5	73.2	74.8	76.3	79.4	82.6	85.2	87.8	90.5	93.0	95.6	97.9	100.3	96

area actually occupied by the air flow, for convenience they are divided into two general classes—those caused by *changes in direction* of the duct and those caused by *changes in cross-sectional area* of the duct. Conduit transitions are representative of changes in cross-sectional area and bends (elbows) are representative of changes in direction of the duct.

Shock losses vary substantially as the square of the velocity of the air flow and are therefore conveniently expressed as a fraction of the velocity head.

$$h_v = c\,\frac{v^2}{2g} \tag{4}$$

where

h_v = the total shock pressure loss, inches of water.

v = velocity of air, feet per second.

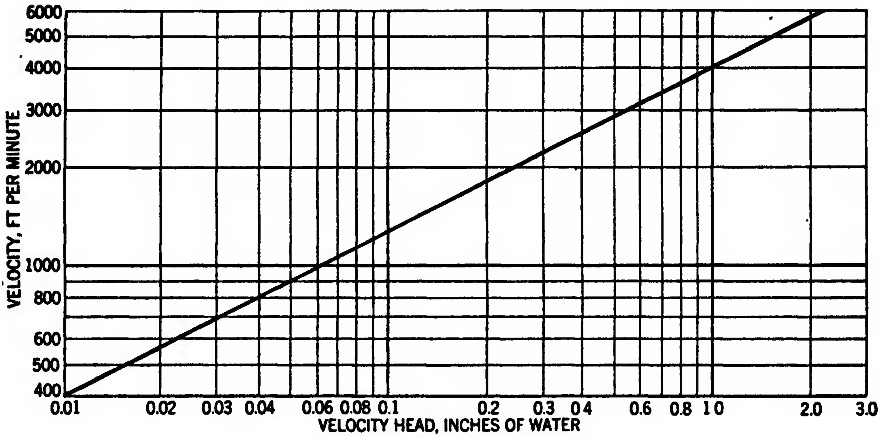


FIG. 3. RELATION BETWEEN VELOCITY AND VELOCITY HEAD FOR STANDARD AIR

$\frac{v^2}{2g}$ = the velocity pressure corresponding to the mean velocity of flow, inches of water.

c = an empirical factor based on experiments and termed shock factor or loss coefficient.

It can be seen from Equation 4 that the shock factor is independent of both density and the unit used and that it represents the number of velocity heads lost at the conduit transition or bend. Values of the shock factor for various duct elements are sometimes tabulated[6,7]. It should be kept in mind, however, that absolutely reliable shock factors have not yet been fully established for all duct elements and that difficulties have been encountered in correlating experimental data of different investigators. A comprehensive study of the loss factors of duct elements is being conducted by the A.S.H.V.E. Research Laboratory for the purpose of obtaining exact data.

For standard air Equation 4 changes to:

$$H_v = C\left(\frac{V_m}{4005}\right)^2 \tag{5}$$

where

H_v = total shock pressure loss, inches of water.
V_m = velocity of air stream, feet per minute.
C = an experimentally determined constant (shock loss coefficient)

Fig. 3 which shows diagrammatically the relation of velocity pressure to velocity for standard air ($V_m = 4005\sqrt{H_v}$) can be conveniently used to find the total shock pressure loss for any duct element with known shock loss coefficient C.

Shock losses are independent of the roughness of the duct walls and therefore cannot be computed correctly as friction losses. It is neverthe-

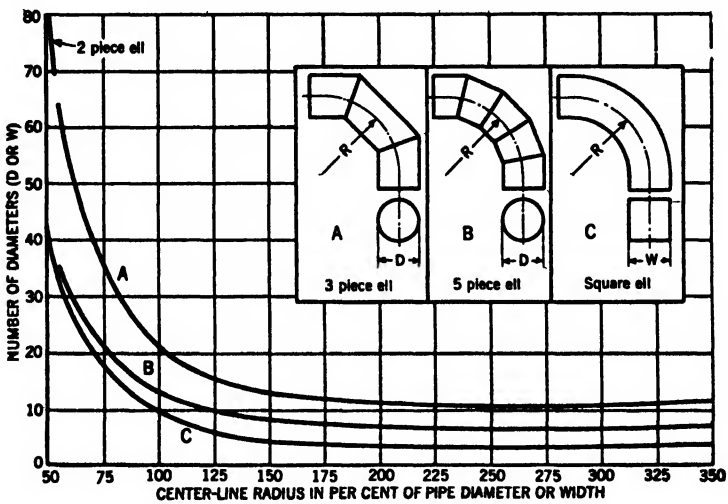


Fig. 4. Loss of Pressure in Elbows

less customary to express the dynamic and friction losses in duct sections or elements in equivalent length of duct in feet or in diameters to facilitate their computation as friction losses. Formulas have been developed for expressing these relations[8].

PRESSURE LOSSES IN ELBOWS

It is customary to express the dynamic and friction losses in elbows as equal to a number of diameters of round pipe, or a number of widths of rectangular pipe, or equivalent length of duct[9]. The curves in Fig. 4 give the number of diameters or widths of pipe which have a frictional resistance equivalent to the pressure drop in the elbows. Curves *B* and *C* are based on tests of round and square elbows[10] of ordinary good sheet metal construction.

Values obtained from Curve *A* should be used when there is any doubt as to quality of duct construction. It is suggested that this curve be used for rectangular elbows and five-piece elbows as it will thus allow an additional factor of safety without seriously affecting the design.

As indicated in Fig. 4, long radius elbows will offer much less resistance to the flow of air than short radius elbows. Experience has shown that

good results may be expected when the radius to the center of the elbow is 1.5 times the pipe diameter or duct width parallel to the radius. Examination of Fig. 4 will indicate that little advantage is to be gained by selecting elbows having a centerline radius of more than two diameters[11]. Elbows having a radius of more than three diameters show a slightly increased resistance due to the increased length of pipe but, when used, they reduce the over-all resistance of the system and therefore should not be avoided.

Table 3. Effect of Vanes on Pressure Loss of 7-inch Square Ventilating Duct[a]

Expressed in feet of total equivalent length of duct (ELD)

SQUARE MITER ELBOW							
Radius Ratio	R1/W	0	.2	.4	.6	.8	1.0
ELD, ft		41.1	30.5	27.5	30.1	37.7	38.5
Radius Ratio	R1/W	0	.2	.3			
	R2/W	0	.4	.5			
ELD, ft		41.1	23.5	23.3			
Radius Ratio	R1/W	0	.2	.3			
	R2/W	0	.4	.5			
	R3/W	0	.6	.7			
ELD, ft		41.1	20.7	22.2			
VANE	A	B	C	D			
ELD, ft	21.8	17.0	17.9	17.8			

STANDARD ELBOWS WITH VARIOUS RADII								
	Radius Ratio	R1/W	0	.2	.4	.6	.8	1.0
	ELD, ft		39.7	23.3	22.0	25.7	28.9	39.7
	Radius Ratio	R1/W	0	.2	.3	.4	.5	.6
		R2/W	0	.4	.5	.6	.7	.8
	ELD, ft		39.7	20.0	22.0	23.0	23.8	25.7
1½" I.R.	Radius Ratio	R1/W	0	.4	.6	.8	1.0	1.2
	ELD, ft		25.3	17.7	16.5	18.7	23.5	25.6
4" I.R.	Radius Ratio	R1/W	0	.7	.8	.9	1.0	1.2
	ELD, ft		14.2	13.3	13.0	12.7	12.5	12.7

[a] For more complete data see A.S.H.V.E. Research Report No. 1216—Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 409).

Note A: Vane *A* made up of a large number of small splitters; *B* made up of a small number of large splitters bent on a large radius, *C* hollow vanes having different outside and inside curvature; and *D* four splitters with $R/W = 0.4$. Elbow same as *D* except 2 in. trailing edge on the end of each splitter, ELD in feet = 17.0.

Where space conditions necessitate the use of short radius or miter elbows in square or rectangular duct work, turning vanes should be used to reduce the pressure losses. Rough or raw edges on the vanes should be avoided to prevent objectionable noise. Table 3 shows typical types of vanes and gives the resistance, expressed in equivalent length of straight pipe, for a 7-in. x 7-in. elbow of each type.

The pressure loss through elbows of less than 90 deg may be assumed to be directly proportional to the ratio of the angle through which the turn is made. The resistance will vary widely for the large degree turns depending upon the aspect ratio and the length of straight pipe between the elbows but, for practical purposes, it may be assumed that the ratio remains proportional to the angle through which the turn is made. Reverse 90 deg elbow turns should be avoided wherever possible but, where used, the friction indicated in Fig. 4 should be doubled for the second elbow. Addi-

tional tests are needed on the loss of pressure in elbows and other types of duct fittings in order to reconcile the difference between values shown in Fig. 4 and results of more recent tests shown in Table 3. A study of this subject is in progress at the A.S.H.V.E. Research Laboratory for the purpose of obtaining such data.

LOSSES DUE TO AREA CHANGES

Area changes in ducts are generally unavoidable, necessitated frequently by the building construction or due to changes in the volume of air carried. Experimental investigations[12,13] of pressure changes and pressure losses at changes of the area of cross-section of duct indicate that the excess pressure loss over the normal friction loss is a shock loss due to a faster stream expanding into a slower stream as determined by the actual areas occupied by the flow rather than the areas of the duct. No perceptible shock loss is due to the converging of the air stream itself where

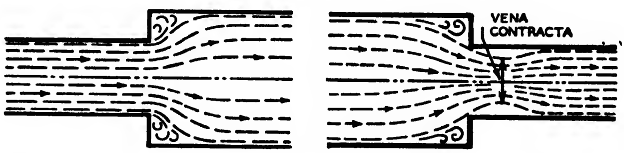


FIG. 5. AIR FLOW AT ABRUPT ENLARGEMENT OR CONTRACTION OF AIR STREAM

the flow is contracted, but the converging of the air stream causes the flow to contract beyond the edge of the constriction, forming a *vena contracta* in which the area of cross-section of the air stream is the minimum immediately following the edge of the constriction. For contraction, therefore, the shock loss is caused by expansion from the vena contracta to the full area following the contraction. Enlargement in area may be considered as a special condition of general expansion following contraction. Fig. 5 illustrates (a) abrupt enlargement and (b) abrupt contraction.

For a sudden symmetrical enlargement a theoretical expression for the loss is:

$$h_e = \left(1 - \frac{A_1}{A_2}\right)^2 \frac{v_1^2}{2g} = \frac{(v_1 - v_2)^2}{2g} \qquad (6)$$

or, for standard air:

$$H_e = \left(1 - \frac{A_1}{A_2}\right)^2 \left(\frac{V_1}{4005}\right)^2 = \left(\frac{V_1 - V_2}{4005}\right)^2 \qquad (7)$$

where

h_e = pressure loss due to sudden enlargement, inches of water.

H_e = pressure loss due to sudden enlargement, based on standard air, inches of water.

V_1 = velocity in the smaller duct, in feet per min.

V_2 = velocity in the larger duct, in feet per min.

A_1 = area of the smaller duct.

A_2 = area of the larger duct.

The loss for a sudden symmetrical contraction, h_c can similarly be expressed as

$$h_c = \left(1 - \frac{A_1'}{A_2}\right)^2 \left(\frac{v_1'^2}{2g}\right) \tag{8}$$

where

v_1' = the velocity at the vena contracta, feet per minute
A_1' = the area of the vena contracta

Introduction of the *contraction coefficient* $\alpha = \frac{A_1'}{A_2}$ and the *loss coefficient* $c = \left(\frac{1}{\alpha} - 1\right)^2$ in Equation (8) gives (with $A_1' \times v_1' = A_2 \times v_2$):

$$h_c = c\,\frac{v_2^2}{2g} \tag{9}$$

or for standard air:

$$H_c = C\left(\frac{V_2}{4005}\right)^2 \tag{10}$$

where

V_2 = velocity in the smaller duct, in feet per min.

Values for C and α are given in following table:

Duct Element	α	C
Sharp corner	0.61 to 0.64	0.41 to 0.314
Broken corner	0.68 to 0.8	0.221 to 0.0625
Corner with small radius	0.9	0.0125
Corner with large radius	0.99	0.0001

Values of α for sharp corners for increasing ratios of A_2/A_1 are given in the following table:

A_2/A_1	0.01	0.1	0.2	0.4	0.6	0.8	1.00
α	0.6	0.61	0.62	0.65	0.7	0.77	1.00

For discharge to atmosphere from a pipe, C = 1.0 in Equation 10, whereas for discharge to atmosphere from an opening in a pipe or wall surface, it is approximately 2.5;

$$C = \left(\frac{1}{\alpha}\right)^2 = \left(\frac{1}{0.63}\right)^2 = 2.5$$

Conditions of expansion and contraction, in addition to abrupt expansion and abrupt contraction, include abrupt contraction followed by either abrupt or gradual expansion, and gradual contraction followed by either abrupt or gradual expansion. Pressure losses for various duct conditions have been determined experimentally, although the available information is generally restricted to symmetrical area changes.[12,13,14]

PRESSURE CHANGES

The fundamental energy equation for standard air flow in a horizontal duct can be written[15]

$$h_1 + \left(\frac{V_1}{4005}\right)^2 = h_2 + \left(\frac{V_2}{4005}\right)^2 + \text{Loss of Pressure (head)} \qquad (11)$$

where

h_1 and h_2 = the static pressure (head) at two given points (1) and (2), inches of water.

$\left(\frac{V_1}{4005}\right)^2$ and $\left(\frac{V_2}{4005}\right)^2$ = the velocity pressure (head) at the same points, inches of water.

Equation 11 states that the mechanical energy at a given point (1) must be equal to the mechanical energy at another point (2), plus any dissipation of mechanical energy to internal energy (loss of pressure). Equation 11 is only valid, if no work is done by or upon the air between the sections (1) and (2) and if there is no heat transfer to or from the air.

In Equation 11, h is a measure of the potential energy and $\left(\frac{V}{4005}\right)^2$ a measure of the kinetic energy or energy of motion. The sum of static pressure and velocity pressure is called total (dynamic, impact) pressure and is a measure of the total energy.

Static pressure and velocity pressure are mutually convertible, that is to say, static pressure may be transformed into velocity pressure, and vice versa. Every change in the cross-sectional area of a duct results in such a conversion of energy which is always accompanied by some loss in efficiency, or loss in total pressure.

In the final analysis of pressure losses in ducts, shock losses are therefore due to accelerations and decelerations of the air stream as a whole. In a *converging* duct, the air velocity will be accelerated; some pressure head will be converted into velocity pressure. This conversion is generally a stable and efficient process; the energy losses are small, and there is no eddy formation.

In an *expanding* duct section on the other hand, the air will be decelerated and an opposing pressure gradient is required to reduce the velocity. If the angle of divergence is appreciable, the flow becomes unstable, there is danger of separation of the flow from the duct wall and large energy losses and eddy formation are possible.[16]

In order to keep losses in an expanding duct section to a minimum and to convert the velocity pressure efficiently into static pressure, the angle of divergence should be kept small.[17] Theoretically, it might seem possible to increase the duct area so gradually that the reduction in velocity and accompanying loss of velocity pressure would occur reversibly and thus permit 100 per cent conversion to static pressure. Such an ideal application of the principle of *static regain* in duct design is, of course, impossible for various reasons[18] such as: the necessity of using sections of uniform diameter because of cost, the need for using ducts of dimensions varying in full inches, the changing of duct sizes mainly at branch connections, and the inevitable loss due to turbulence. The principle of static pressure regain is, however, of importance in the economical design of duct systems.

Fig. 6 shows the application of static pressure regain to a simple fan and

discharge duct.[19] The fan in the upper part of the figure has a free inlet and discharges air through a straight duct, the diameter of which is equal to the fan outlet. The total pressure which must be provided by the fan is therefore the sum of the pressure that is necessary to overcome the friction in the duct (no shock pressure loss), plus the velocity pressure which in this case is the same at any location along the length of the duct.

In arrangement B in the lower part of Fig. 6 a diverging section with after section has been added to the straight duct. The velocity in the diverging section is therefore decreased and velocity pressure converted

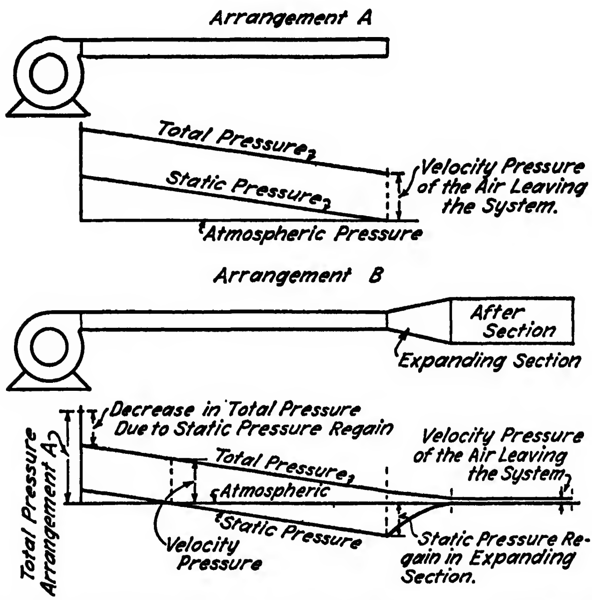


Fig. 6. Application of Static Pressure Regain to a Simple Fan and Discharge Duct

into static pressure before the air is released to the atmosphere. It can be seen that in case B, the total pressure at the fan outlet is less than in case A; and thus a saving in horsepower can be effected.

The regain in static pressure h_r in an abruptly expanded section is the difference in the velocity pressures of the small and the large duct, minus the shock pressure loss (Equation 6):

$$h_r = \left[\frac{v_1^2}{2g} - \frac{v_2^2}{2g}\right] - \left[\frac{(v_1 - v_2)^2}{2g}\right] \tag{12}$$

or simplified

$$h_r = \frac{v_1 v_2 - v_2^2}{g} = \frac{v_2(v_1 - v_2)}{g} \tag{13}$$

where

h_r = regain in static pressure, feet of fluid flowing.

v_1 and v_2 = mean velocities in smaller and larger duct sections respectively, feet per second.

The static pressure regain in a gradually expanding transition followed by an after section may be expressed as:

$$h_r = \left[\frac{v_1^2}{2g} - \frac{v_2^2}{2g}\right] - \left[\frac{c(v_1 - v_2)^2}{2g}\right] \tag{14}$$

or

$$h_r = \frac{v_1^2 - v_2^2 - c(v_1 - v_2)^2}{2g} \tag{15}$$

Curves have been developed showing the static pressure regain and the *theoretical* efficiency of conversion in abrupt expansion and in diverging sections in smooth circular ducts.[13,14]

DUCT DESIGN

The discussion of duct design in this chapter refers to ducts in fan systems for central heating, ventilating and air conditioning systems. Additional data for heating ducts used in residences are to be found in Chapter 21 (Gravity Warm Air Systems) and Chapter 22 (Mechanical Warm Air Systems). The design of ducts in industrial exhaust systems is discussed in Chapter 46.

The following general rules should be followed in the design of a duct system:

1. The air should be conveyed as directly as possible at the permissible velocities to obtain the desired results with greatest economy of power, material and space.

2. Sharp elbows and bends should be avoided. Carefully designed splitters and turning vanes should be used to reduce the elbow or outlet pressure loss. (See section Pressure Loss in Elbows).

3. Diverging transformation pieces should be made as long as practicable. As shown in the section on area changes, losses in sudden enlargements are high and abrupt enlargements should be avoided. The included angle of divergence for enlargements should not exceed 20 deg. Losses in contractions are low, but the included angle of divergence should not be larger than 60 deg.

4. Special care should be taken to avoid restriction of flow in elbows or transformation pieces.

5. Rectangular ducts should be made as nearly square as possible. Good practice limits the ratio between the long side and the short side 3 to 1. In no case should this ratio exceed 10 to 1.

6. Ducts should be constructed of smooth material, such as steel or aluminum sheet metal. For ducts made from other materials, for example masonry, proper allowance for the surface friction coefficient should be made.

Procedure for Duct Design

The general procedure for designing a duct system is outlined in the several items listed herewith:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.

2. Arrange the positions of duct outlets to insure the proper distribution of air.

3. Divide the building into zones and proportion the volume of air necessary for each zone.

4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity and throw. In case of some ceiling diffusers, determine size of outlet for proper throat velocity and radius of diffusion.

5. Calculate the sizes of all main and branch ducts by one of the three methods of sizing air supply systems in common use, the velocity reduction method, the equal friction method or the static regain method.

6. Calculate the losses for the duct offering the greatest resistance to the flow of air, using the A.S.H.V.E. Friction Chart, Fig. 1, and the other data given in this chapter.

Recommended Design Velocities

The air velocities given in Table 4 have been found to give satisfactory results in engineering practice. Where the higher velocities are used, the ducts should be cross-braced to prevent breathing, buckling or vibration, and should be constructed of heavier gage metal. At the higher velocities it is particularly important to design the ducts for minimum resistance. As high velocities at one point offset the effect of proper design in all other parts of the system, emphasis should be placed on the importance of air velocities, elbow design, location of dampers, fan connections, grille and register approach connections, and similar details. For industrial

TABLE 4. RECOMMENDED AND MAXIMUM DUCT VELOCITIES

Designation	Recommended Velocities, fpm			Maximum Velocities, fpm		
	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside Air Intakes[a]	700	800	1000	800	900	1200
Filters[a]	250	300	350	300	350	350
Heating Coils[a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800–1200	1100–1600	1300–2200
Branch Ducts	600	600–900	800–1000	700–1000	800–1300	1000–1800
Branch Risers	500	600–700	800	650–800	800–1200	1000–1600

[a] These velocities are for total face area, not the net free area.

buildings, noise is seldom given much consideration, and main duct velocities as high as 2800 or 3000 fpm are sometimes used, but when these velocities are used due consideration should be given to duct design, resistance pressure, fan efficiencies and motor horsepower. For department stores and similar buildings, 2000 to 2200 fpm are sometimes used in main ducts where noise is not objectionable and space conditions warrant it.

Where high velocity diffusing outlets are used, the duct velocity should be, if possible, equal or somewhat lower than the throat (neck) velocity of the diffuser, in order to utilize the effect of higher static pressure in the duct for equalization of air discharge.

The velocities in main ducts, and particularly in branch ducts and branch risers should be correlated to the throat (neck) velocity of the air outlets and manufacturers data should be consulted for permissible throat velocity for the particular type of application.

If it is necessary to use a duct velocity that is twice the velocity for an outlet mounted on the side of the duct, a collar with directing vanes should be used to straighten the flow of air from the outlet. Sometimes it is desirable to mount the outlet flush with the side of the duct in which case the duct velocity should be kept below twice that of the outlet velocity, and even then an outlet larger than normally required should be used, as

the entire outlet area will not be effective. Manufacturer's selection tables base sizing of outlets on required volume of air, temperature differential, and distance of throw or radius of diffusion. In following their recommendations, maxima should be avoided. See Chapter 40 for a discussion of air outlets.

DESIGN METHODS

The design of the air transmission system is generally the last step in the design of the heating, ventilating or air conditioning system, but it should always be kept in mind that the type of air transmission used will, to some extent, depend on the type of equipment used as well as on the purpose of the system. Zoning and zone control and other factors and their influence on the transmission and air distribution system are briefly discussed in Chapter 43 (Central Systems for Air Conditioning).

The methods used for the design of duct systems reflect, to some degree, certain developments in the arts of heating, ventilating and air conditioning and it took a long time before empirical methods gave way to more refined and scientific calculations. Some engineers prefer speed and simplicity to scientific exactness, but experience is then needed and proper judgment must be exerted to prevent mistakes. Both the *Velocity Reduction Method* and the *Equal Friction Method* take no account of the static regain resulting from the difference between the velocity of fan discharge and velocities of pipe discharge, and are therefore, to some degree, approximate methods. However, they are more easily applied than the static regain method which is based on scientific reasoning but is subject to an assumption regarding the efficiency of conversion from kinetic energy to static regain. This efficiency for ordinary expansion sections is usually taken as 50 per cent.

1. Velocity Reduction Method

When this method is used, arbitrary velocities for the various sections of the ducts are selected, with the highest velocity at the fan outlet and lower velocities down the run as various branch ducts are taken off the main duct. As the quantities of air that are to be delivered through each section of the duct are known, the area of each duct section can be easily determined by using the formula:

$$A = \frac{Q}{V_m} \tag{16}$$

where

A = duct area in square feet.

Q = air quantity in cubic feet per minute.

V_m = air velocity in feet per minute.

To find the total static pressure against which the fan must operate the static pressure loss of each section is calculated separately and the total loss found by adding the individual losses of the sections of the duct which has the highest resistance. This may be the duct with the longest run, but not necessarily so.

The velocity method has the advantage that the duct area can be determined very easily. It should be used only for simple layouts. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Balancing is obtained by use of dampers.

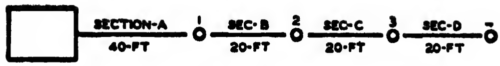


FIG. 7. DUCT LAYOUT FOR EXAMPLE 1

Example 3. (Velocity Reduction Method). A duct layout is shown in Fig. 7. The fan delivers 8000 cfm. Four outlets deliver 2000 cfm each. Find duct dimensions and total pressure loss.

Solution. Select velocity for Section A (1600 fpm) and reduce velocity arbitrarily along run. Find duct areas by using Equation 16. For selection of circular equivalents of rectangular ducts refer to Table 2 and for determination of friction loss in duct refer to Fig. 1 (See *Example 1*). Results are tabulated in Table 5.

2. Equal Friction Method

When the equal friction method of design is used, the duct system is designed for equal friction per foot of length. This prevents one section of the duct from having an excessive resistance compared with another. The usual procedure in this method is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All ducts are then sized for equal friction per unit length by the use of Fig. 1 and Table 2. The equal friction method has the advantage of automatically reducing the velocities in the various sections of the system and also of allowing a quick check of the total duct resistance.

In cases where the fan or factory assembled air conditioning unit can operate against only a limited external resistance, it is necessary to divide the permissible total resistance by the total *equivalent* length of the longest or most complicated run of duct to determine the design resistance per 100 ft and then to size all ducts at this resistance value. This will automatically determine the duct velocities and give the desired total duct resistance. A further refinement, which is sometimes used in large systems, is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

Example 4. Equal Friction Method. A duct layout is shown in Fig. 8. The fan delivers 2500 cfm. Outlets No. 1 and 2 deliver 750 cfm each and outlet No. 3 delivers 1000 cfm. Trunk velocity is assumed as 1500 fpm; the area will be 1.67 sq ft; (240 sq in.); and the size will be 20 x 12 in. Determine sizes of ducts for sections B, C, D and E and find the total pressure loss.

Solution. The equivalent round diameter of a 20 x 12 in., rectangular duct is 16.8 in. (from Table 2). Referring to Friction Chart, Fig. 1, a volume of 2500 cfm through a 16.8 in. duct gives a resistance of 0.2 in. per 100 ft. The amount of air to be handled by each section is known and the corresponding round duct sizes with equal pressure drop for these values can be located on the 0.2 in. friction line. The equivalent rectangular duct sizes are then selected from Table 2.

Results are tabulated in Table 6.

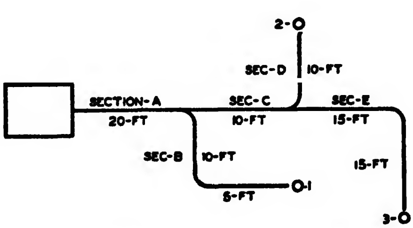


FIG. 8. DUCT LAYOUT FOR EXAMPLE 4

TABLE 5. TABULATION OF RESULTS (EXAMPLE 3)

SECTION	AIR VOLUME cfm	VELOCITY fpm	AREA sq. ft.	AREA sq. in.	DUCT SIZE in.	DIAM. in.	FRICT. PER 100 FT. in. H_2O	FRICT. LOSS in. H_2O
A	8000	2200	3.64	524	26 x 20	24.8	.25	0.10
B	6000	2000	3.00	432	22 x 20	22.9	.23	0.05
C	4000	1800	2.22	320	20 x 16	19.5	.24	0.05
D	2000	1600	1.25	180	12 x 16	15.1	.24	0.05

Total resistance, 0.25

The total pressure loss in the longest run is the friction loss in Sections (A + C + E), plus the shock loss in one elbow and the loss through the outlet (3). The pressure loss in the elbow will be assumed as 12W (Fig. 4), the equivalent length of duct is then 10 ft and the design loss will be 0.02 in.

Friction loss (A + C + E).. 0.12 (Duct length = 20 ft + 10 ft + 15 ft + 15 ft)
Shock loss elbow.......... 0.02
Loss through outlet....... 0.12

Total pressure loss in duct.................. = 0.26 in.

The pressure required at the beginning of the main run is therefore 0.26 in. The fan selected for the duct system must not only deliver the required volume of air against this loss, but also against the losses in all air conditioning apparatus such as washers or spray chambers, heating or cooling coils and filters. The static head required of the fan for the usual air conditioning installation is between 1 and 1.5 in. of water. About one-third of this represents losses in the duct system. The losses in the air conditioning apparatus can be obtained from manufacturers' catalogs.

Resizing of Ducts

In order to equalize the pressure drop in the system the following additional procedure is recommended.

Assume R_1, R_2, R_3 to be the total pressure loss through ducts (1), (2) and (3); r_a, r_b, r_c, r_d, r_e the friction losses in the straight sections of the system; r_{be}, r_{de}, r_{ee} the elbow losses, and r_1, r_2, r_3 the loss through the outlets then becomes

$$R_1 = r_a + r_b + 2r_{be} + r_1$$
$$R_2 = r_a + r_c + r_d + r_{de} + r_2$$
$$R_3 = r_a + r_c + r_e + r_{ee} + r_3$$

If $R_1 = R_2 = R_3 = R$

$$r_b + 2r_{be} = R - r_a - r_1$$
$$r_d + r_{de} = R - r_a - r_c - r_2$$

or with the values from *Example 4*:

$$r_b + 2r_{be} = 0.26 - 0.04 - 0.12 = 0.10 \text{ in.}$$
$$r_d + r_{de} = 0.26 - 0.04 - 0.02 - 0.12 = 0.08 \text{ in.}$$

TABLE 6. TABULATION OF RESULTS (EXAMPLE 4)

SECTION	AIR VOLUME cfm	FRICTION PER 100 FT. in.	DIAM. in.	RECTANGULAR DUCT in.	VELOCITY fpm	FRICTION PER 100 FT. in.	DIAM. in.	RECTANGULAR DUCT in.	VELOCITY fpm
A	2500	0.2	16.8	20 x 12	1500	0.2	17	20 x 12	1600
B	750	0.2	10.7	10 x 9	1190	0.286	10	10 x 8	1350
C	1750	0.2	14.8	15 x 12	1400	0.2	14.5	15 x 12	1450
D	750	0.2	10.7	10 x 9	1190	0.4	9.8	10 x 8	1350
E	1000	0.2	12	10 x 12	1200	0.2	12	10 x 12	1300

The loss in the elbows can be assumed to be 12W or 10 ft, the friction loss of head per 100 equivalent ft is then

for duct (1) $$\frac{0.10}{0.15 + 2 \times 0.10} = 0.286 \text{ in.}$$

for duct (2) $$\frac{0.08}{0.10 + 0.10} = 0.4 \text{ in.}$$

Using Friction Chart Fig. 1, the duct diameter of Section B to carry 750 cfm with a loss of 0.286 in. per 100 ft is found as 10 in. and the duct diameter of Section D to carry 750 cfm with a loss of 0.4 in. per 100 ft is 9.8 in. Equivalent rectangular ducts are 10 × 8 in., the velocity in both ducts is 1350 fpm.

The actual loss in ducts (1) and (2) is:

$$0.04 + 20 \times \frac{0.286}{100} + 15 \times \frac{0.286}{100} + 0.12 = 0.26$$

$$0.06 + 10 \times \frac{0.4}{100} + 10 \times \frac{0.4}{100} + 0.12 = 0.26$$

For final survey of ducts selected see Table 6 (Tabulation of Results).

3. Static Regain Method

When this method is used,[20] the velocity is reduced at each branch or take-off so that the recovery in static pressure due to this reduction will offset the friction in the succeeding section. This method is based on the convertibility of static pressure and velocity pressure as discussed in section Pressure Changes. If no loss from friction or shock occurred, the change in velocity head would be completely converted into a regain in static pressure, that is to say for standard air:

$$h'_r = \left(\frac{V_1}{4005}\right)^2 - \left(\frac{V_2}{4005}\right)^2 \qquad (17)$$

where

V_1 = initial velocity, fpm

V_2 = velocity after reduction, fpm

h'_r = theoretical head recovered (static regain)

Under ideal conditions, 0.7 to 0.8 of the velocity head is actually recovered, but for practical design an average recovery of 0.5 is assumed. The actual velocity head recovered h_r, becomes then

(18)

Charts based upon Equation 18 and designed to facilitate selection of duct sizes by the static regain method are found in Reference 20. See also chart included in Reference 18.

DUCT CONSTRUCTION DETAILS

Straight sections of round duct are usually formed by rolling the sheets to the proper radius and grooving the longitudinal seam. Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam due to lack of equipment. Elbows and transformation sections are generally formed with *Pittsburgh* corner seams because this seam is easier to

lock in place than the double seam, but complicated fittings such as double compounded elbows are usually constructed with double seam corners. The construction of these various seams as well as the types of girth connections are shown in Fig. 9. The application of the various slips and connections is outlined in Table 7. The end slip may be used wherever *S* slips are recommended. Where drive slips are used the end slip may be applied on the narrow side of the duct and the drive slips on only the maximum side. Ducts 25 to 30 in. in size should be reinforced between

TABLE 7. RECOMMENDED SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[a]

U. S. Std. Gage	Maximum Side, Inches	Type of Transverse Joint Connections[b]	Bracing
26	Up to 12	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
24	13 to 24	S, Drive. Pocket or Bar Slips, on 7 ft 10 in. centers	None
	25 to 30	S, Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x $\frac{1}{8}$ in. angles 4 ft from joint
22	31 to 40	Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft.10 in. centers[c]	1 x 1 x $\frac{1}{8}$ in. angles 4 ft from joint
	41 to 60	$1\frac{1}{2}$ in. Angle Connections, or $1\frac{1}{2}$ in. Pocket or $1\frac{1}{2}$ in. Bar Slips with $1\frac{3}{8}$ in. x $\frac{1}{8}$ in. bar reinforcing on 7 ft 10 in. centers[c]	$1\frac{1}{2}$ x $1\frac{1}{2}$ x $\frac{1}{8}$ in. angles 4 ft from joint
20	61 to 90	$1\frac{1}{2}$ in. Angle Connections, or $1\frac{1}{2}$ in. Pocket or $1\frac{1}{2}$ in. Bar Slips 3 ft 9 in. maximum centers with $1\frac{3}{8}$ x $\frac{1}{8}$ in. bar reinforcing	$1\frac{1}{2}$ x $1\frac{1}{2}$ x $\frac{1}{8}$ in. diagonal angles, or $1\frac{1}{2}$ x $1\frac{1}{2}$ x $\frac{1}{8}$ in. angles 2 ft from joint
18	91 and up	2 in. Angle Connections or $1\frac{1}{2}$ in. Pocket or $1\frac{1}{2}$ in. Bar Slips 3 ft 9 in. maximum centers with $1\frac{3}{8}$ x $\frac{1}{8}$ in. bar reinforcing[d]	$1\frac{1}{2}$ x $1\frac{1}{2}$ x $\frac{1}{8}$ in. diagonal angles, or $1\frac{1}{2}$ x $1\frac{1}{2}$ x $\frac{1}{8}$ in. angles 2 ft from joint

[a] For normal pressures and velocities (see Table 4) utilized in typical ventilating and air conditioning systems. Where special rigidity or stiffness is required, ducts should be constructed of metal two gages heavier. All uninsulated ducts 18 in. and larger should be cross-broken. Cross-breaking may be omitted on uninsulated ducts if two gages of heavier metal are used.

[b] Other joint connections of equivalent mechanical strength and air tightness may be used.

[c] Duct sections of 3 ft 9 in. may be used with bracing angles omitted, instead of 7 ft 10 in. lengths with joints indicated.

[d] Ducts 91 in. and larger require special field study for hanging and supporting methods.

Note: Aluminum ducts, well supported, should be 2 (B & S) gages heavier than steel ducts.

the joints, but not necessarily at the joint. Ducts 31 in. and up should be reinforced at the joint and between the joints; if drive slips are used the angles are usually riveted to the duct about 2 in. from the slips. It is good practice to cross-break or kink all flat surfaces to prevent vibration or buckling due to the air flow and accompanying variations in internal pressure. Round ducts are sometimes swedged 1.5 in. from the ends so that the larger end will butt against the swedge and are held in place with sheet metal screws. Where swedges are not used it is general practice to paste the joint with asbestos paper to insure a tight joint.

The construction of elbows and changes of shape cannot be definitely outlined because of the varied conditions encountered in the field, but in

general long radius elbows and gradual changes in shape tend to maintain uniform velocities accompanied by decreased turbulence, lower resistance and a minimum of noise.

Heavy canvas connections (asbestos cloth if there is a fire hazard) are recommended on both the inlet and outlet to all fans. The fan discharge

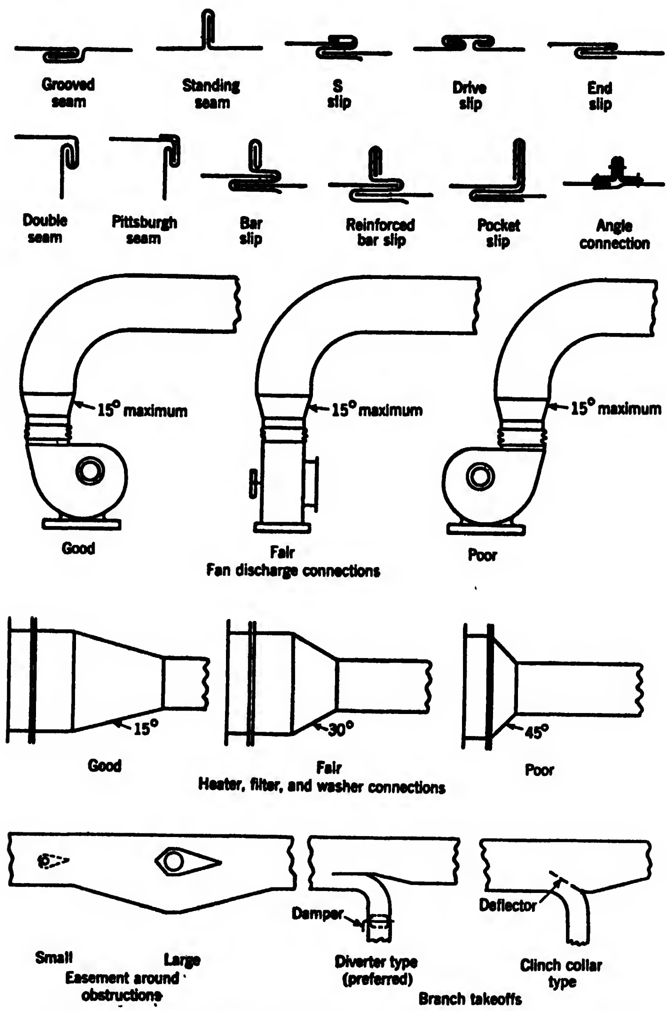


FIG. 9. SHEET METAL DUCT AND ARRANGEMENT DETAILS

connections shown in Fig. 9 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 9 will readily show that uniform velocity through the heater cannot be expected in the diagram noted *poor*. When obstructions cannot be avoided, the duct area should never be decreased more than 10 per cent and then a *streamlined* collar should be used. Larger obstructions require an increase in the duct size in order to maintain as nearly uniform velocity as possible. Branch take-offs should always be

Table 8. Weights of Sheet Metal Used for Duct Construction

U. S. Std. Gage	Black Sheets				Galvanized Sheets[a]			
	Approximate Thickness, In.		Weight Per Square Foot		Approximate Thickness, In.		Weight Per Square Foot	
	Steel	Iron	Ounces	Pounds	Steel	Iron	Ounces	Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0.0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

[a]Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

arranged to cut or slice into the air stream in order to reduce as far as possible the losses in velocity head.

The recommended gages for sheet metal duct construction are given in Table 7. Weights of sheet metal per square foot of surface for different gages are given in Table 8. The weights of various gages and the areas for any length of run of rectangular sheet metal ducts may also be determined from Fig. 10. The bottom scale represents the sum of the two sides of the duct and the oblique lines give the length of run in feet. Proceeding horizontally to the right from the intersection of vertical and oblique lines on the chart, the area of the duct may be determined in the first vertical scale.[21] The scales to the right give the weights of the duct run for different gages of metal. In calculating the weights of duct, it is considered good practice to allow 20 per cent additional for weights of joints and bracings. Various weights and thicknesses of standard copper sheets will be found in Table 9 and of 2 *S* aluminum in Table 10.

HEAT LOSSES FROM DUCTS

In designing duct systems, the heat gains or losses of ducts should not be neglected. Heat gains in large duct systems can be quite considerable,

Table 9. Weights and Thicknesses of Standard Copper Sheets[a]
Rolled to Weight

Weight per Square Foot		Thickness, Inches		Nearest Gage No.		
Ounces	Pounds	Decimal Equivalent	Nearest Fraction	B. & S.	Stubs	U. S. Std.
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[a]Variations from these weights must be expected in practice.

not only if the duct passes through unconditioned space, but also on long duct runs within conditioned space. Proper insulation will remedy this situation considerably, but sometimes a redistribution of the supply air to the various supply outlets is necessary in order to compensate for the heating effect of the duct surface.

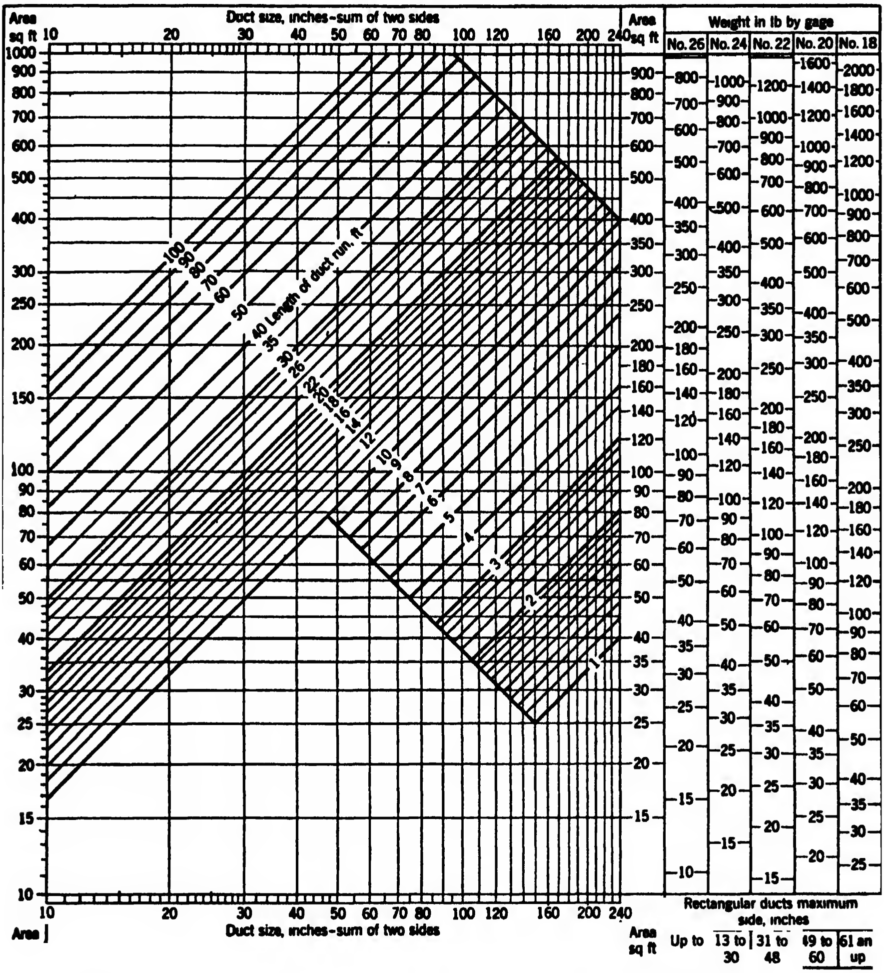


FIG. 10. AREA AND WEIGHT OF RECTANGULAR SHEET METAL DUCTS

The thermal transmittance U for ducts can be found as follows:

For uninsulated metal duct, $$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o}} \qquad (19)$$

For uninsulated non-metallic ducts, $$U = \frac{1}{\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o}} \qquad (20)$$

where

- U = overall coefficient of heat transfer, Btu per (hour) (square foot) (degree Fahrenheit).
- f_i = surface conductance (inside) Btu per (hour) (square foot) (degree Fahrenheit).
- f_o = surface conductance (outside) Btu per (hour) (square foot) (degree Fahrenheit).
- x = thickness, inches
- k = unit conductivity of material, Btu per (hour) (square foot) (degree Fahrenheit per inch thickness).

TABLE 10. WEIGHTS AND THICKNESSES OF 2S ALUMINUM (DENSITY 0.098 LB/CU IN.)

B. & S. Gage	Thickness, Inches		Weight per Square foot	
	Decimal	Nearest Fraction	Ounces	Pounds
28	0.012	1/64	2.7	.169
26	0.016	1/64	3.6	.226
24	0.020	1/64	4.5	.282
22	0.025	1/32	5.4	.353
20	0.032	1/32	7.2	.452
18	0.040	3/64	9.0	.563
16	0.051	3/64	11.5	.720
14	0.064	1/16	14.4	.903

Where x is small and k is large, however, this factor $\frac{x}{k}$ is of little importance and may be neglected.

Film conductance f_i for air flowing in ducts apparently depends only on the velocity of the air and the diameter of the duct. A fairly reliable inside coefficient can be calculated from Schultz's modified equation:

$$f_i = \frac{0.32V_a^{0.8}}{D^{0.25}} \tag{21}$$

where

- V_a = velocity of air in duct, feet per second.
- D = inside diameter of duct, feet.

Film conductance f_o depends on a number of variables including temperature, diameter, and emissivity of the outer surface and can be calculated from data in Chapter 5. From this explanation, it is seen that it is unwise to recommend a given value of U for all uninsulated metal ducts.

The heat loss from a given length of duct can be expressed by:

$$Q_w = UPl\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] \tag{22}$$

where

- Q_w = heat loss through duct walls, Btu per hour.
- P = perimeter of duct, feet.

l = length of conduit, feet
t_1 = temperature of air entering duct, Fahrenheit degrees.
t_2 = temperature of air leaving duct, Fahrenheit degrees.
t_s = temperature of air surrounding duct, Fahrenheit degrees.

The heat given up by the air in the duct is:

$$Q_v = 0.24W(t_1 - t_2) = 14.4AV_m\rho_v(t_1 - t_2) \quad (23)$$

where

W = weight of air through duct, pounds per hour
A = cross-sectional area of duct, square feet
V_m = velocity of fluid, feet per minute
ρ_v = density of air at specified temperature at which velocity V_m, is measured, pounds per cubic foot.

Equating (22) and (23):

$$\frac{t_1 + t_2 - 2t_s}{t_1 - t_2} = \frac{28.8AV_m\rho_v}{UPl}$$

Let $y = \frac{28.8AV_m\rho_v}{UPl}$ for rectangular ducts, and $\frac{7.2DV_m\rho_v}{Ul}$ for round ducts, and solve for t_1 and t_2 :

$$t_1 = \frac{t_2(y + 1) - 2t_s}{(y - 1)} \quad (24) \qquad t_2 = \frac{t_1(y - 1) + 2t_s}{(y + 1)} \quad (25)$$

For low velocities and long ducts of small cross-section, a somewhat more accurate formula may be used as follows:

$$t_2 = \frac{t_1 - t_s}{e^{\left(\frac{UPl}{14.4A\rho_v V_m}\right)}} + t_s \quad (26)$$

where

e = Naperian base of logarithms = 2.718.

In using Equations 24, 25, and 26, one of the duct air temperatures will be unknown and will be obtained by substitution of the other known or assumed values.

Heat loss coefficients for insulated ducts with various conductivities are given in Fig. 11. The conductivities of various materials, which are based on mean temperatures, ranging from about 70 to 90 F, will be found in Table 2 of Chapter 6. For cases where the mean temperature is other than that at which the test was conducted, a correction should be made. However, in most cases the effect of this factor will be small and may be neglected.

Example 5. Determine the entering air temperature and heat loss for a duct 24 × 36 in. cross-section and 70 ft in length, insulated with ½ in. of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution. Referring to Fig. 11, the over-all heat transmission coefficient is found to be 0.49 Btu. From Table 1, Chapter 3 the density of air at 70 F and 29.921 in. Hg

is found to be 1/13.348 = 0.0749 lb per cubic foot. Substituting these and the other given values in Equation 24:

$$y = \frac{28.8 \times 6 \times 1200 \times 0.0749}{0.49 \times 10 \times 70} = 45.3$$

$$t_1 = \frac{120(45.3 + 1) - 80}{45.3 - 1} = 123.7$$

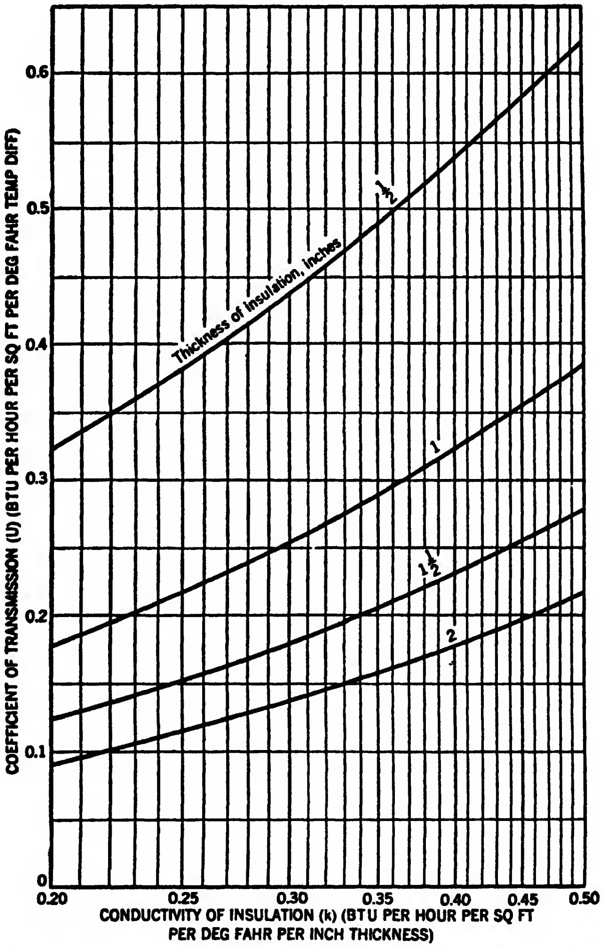


FIG. 11. HEAT LOSS COEFFICIENTS FOR INSULATED DUCTS[a]

[a] For round ducts less than 30 in. diameter, increase heat transmission values by the percentages shown.

THICKNESS OF INSULATION (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

Substituting in Equation 22:

$$Q = 0.49 \times 10 \times 70 \left[\left(\frac{123.7 + 120}{2} \right) - 40 \right] = 28{,}100 \text{ Btu per hour.}$$

For special considerations which apply to insulation of ducts in marine installations see Chapter 49.

MAINTENANCE

Ducts should be designed in such a manner as to enable easy maintenance.[22] They should have enough access doors, not only to enable inspection, but also to facilitate cleaning of the ducts.[23] The periodic cleaning of the ducts should be part of the regular maintenance schedule. It should be done efficiently and competently to avoid difficulties or hazards in the operation of the system.[24,25]

LETTER SYMBOLS USED IN CHAPTER 41

α = coefficient of contraction.

ϵ = absolute roughness, feet.

ρ_o = density of air under actual (operating) conditions, any consistent units.

ρ_s = density of air under standard conditions, any consistent units.

ρ_v = density at which V_m is measured, pounds per cubic foot.

A = cross-section area of duct, square feet.

A_1 = area of smaller duct.

A_2 = area of larger duct.

a = length of one side of rectangular duct, feet or inches. (Other side is b.)

b = length of one side of rectangular duct, feet or inches. (Other side is a.)

C = shock loss coefficient for standard air.

c = shock loss coefficient.

D = inside diameter of conduit, feet.

D_e = circular equivalent of a rectangular duct for equal friction and capacity, inches.

d = equivalent diameter, feet or inches.

e = Naperian base of logarithms = 2.718.

f = non-dimensional friction coefficient.

f_i = surface conductance (inside) Btu per (hour) (square foot) (Fahrenheit degree).

f_o = surface conductance (outside) Btu per (hour) (square foot) (Fahrenheit degree).

g = acceleration due to gravity, 32.17, feet per (second) (second).

H_c = pressure loss due to sudden enlargement, based on standard air, inches of water.

H_e = pressure loss due to sudden enlargement, based on standard air, inches of water.

H_v = total shock pressure loss for standard air, inches of water.

h_1 and h_2 = static pressure head at given points (1) and (2).

h_o = friction loss under actual (operating) conditions, any consistent units.

h_c = pressure loss due to sudden contraction, inches of water.

h_e = pressure loss due to sudden enlargement, inches of water.

h_f = friction loss, feet of fluid flowing.

h_r = regain in static pressure, feet of fluid flowing.

h_s = friction loss under standard conditions, any consistent units.

h_v = total shock pressure loss, inches of water.

k = conductivity, Btu per (hour) (sq ft) (Fahrenheit degree per inch).

l = length of conduit, feet.

P = perimeter of duct, feet.

Q_w = heat loss through duct walls, Btu per hour.

t_1 = temperature of air entering duct, Fahrenheit degrees.

t_2 = temperature of air leaving duct, Fahrenheit degrees.

t_3 = temperature of air surrounding duct, Fahrenheit degrees.

U = thermal transmittance coefficient, Btu per (hour) (square foot), (Fahrenheit degree).

V = fluid or air velocity, feet per second.

V_a = velocity of air in duct, feet per second.

V_m = velocity of air or fluid, feet per minute.

V_1 = velocity in smaller duct, feet per minute.

V_2 = velocity in larger duct, feet per minute.

v = velocity of air, feet per second.
v_1 = mean velocity in smaller duct section, feet per second.
v_2 = mean velocity in larger duct section, feet per second.
W = weight of air through duct, pounds per hour.
x = thickness, inches.

REFERENCES

[1] A.S.H.V.E. RESEARCH PAPER—A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 303).

[2] Friction Factors for Pipe Flow by L. F. Moody (*A.S.M.E. Transactions*, Vol. 66, 1944, p. 671).

[3] See Chapter 1 for definition of standard air.

[4] Friction Charts for Gases Including Correction for Temperature, Viscosity and Pipe Roughness, by R. D. Madison and W. R. Elliot (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1946).

[5] Friction Equivalents for Round, Square and Rectangular Ducts, by R. G. Huebscher, (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1947).

[6] Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne and W. A. Grant, 1940, p. 234 (Pitman Publishing Corp.).

[7] Heating, Ventilating and Air Conditioning, by L. A. Harding and A. C. Willard, 1932, p. 673 (John Wiley and Sons).

[8] Pressure Losses Due to Bends and Area Changes in Mine Airways, by G. E. McElroy, (*U. S. Bureau of Mines*, Information Circular I.C. 6663, p. 4).

[9] A.S.H.V.E. RESEARCH REPORT No. 1211—Pressure Loss Caused by Elbows in 8-inch Round Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 335).

[10] Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 366).

[11] Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, p. 365, August, p. 427, September, p. 483, 1936).

[12] Engineering Factors in the Ventilation of Metal Mines, by G. E. McElroy, (*U. S. Bureau of Mines*, Bulletin 385, pp. 55–68).

[13] Pressure Losses Resulting from Changes in Cross-Sectional Area in Air Ducts, by A. P. Kratz and J. R. Fellows, (*University of Illinois Engineering Experiment Station*, Bulletin No. 300).

[14] Fan Engineering, (R. D. Madison, Editor) 5th ed., 1948, pp. 124–128 (Buffalo Forge Company).

[15] Air Conditioning Principles, by C. O. Mackey, 1941, p. 158. (International Textbook Company).

[16] Fluid Mechanics, by R. C. Binder, 1943, p. 152. (Prentice-Hall).

[17] Modern Diffuser Design by G. N. Patterson. (*Aircraft Engineering*, Sept., 1938, p. 267) and The Design of Airplane Ducts by G. N. Patterson (*Aircraft Engineering*, July, 1939, p. 263).

[18] Duct Sizing with Partial Static Regain, by R. H. Anderegg and F. W. Hutchinson, (*Heating, Piping and Air Conditioning*, August, 1946, p. 97).

[19] Power Savings Through Static Pressure Regain in Air Ducts, by J. R. Fellows, (*Heating, Piping and Air Conditioning*, April, 1939, p. 219).

[20] Reference 6, p. 240.

[21] Method of Determining Rectangular Equivalents and Weights of Ducts, by P. Franck, (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning* December, 1940, p. 743).

[22] Design of Air Conditioning Systems for Low Maintenance, by M. G. Kershaw, (*Refrigerating Engineering*, October, 1946, p. 315).

[23] Standards of the *National Board of Fire Underwriters*, (N.B.F.U. Pamphlet No. 90, p. 21).

[24] Clean Your Ducts, by G. L. Candler, (*Heating, Piping and Air Conditioning*, July, 1947, p. 83).

[25] Duct Leakage and Job Revision, by E. L. Schulz, (*Heating and Ventilating*, October, 1944, p. 83).

BIBLIOGRAPHY

FLUID FLOW (See Chapter 4)

Engineering Applications of Fluid Mechanics, by J. C. Hunsaker and B. G. Rightmire, 1947 (McGraw-Hill Book Company, Inc.)

Elementary Mechanics of Fluids, by Hunter Rouse 1946. (John Wiley & Sons, Inc.)

The Flow of Liquids, by W. H. McAdams, (*Refrigerating Engineering*, February, 1925, p. 279).

The Flow of Fluids in Closed Conduits, by R. J. S. Pigott, (*Mechanical Engineering*, Vol. 55, 1933, p. 497).

A Study of the Data on the Flow of Fluids in Pipes, by E. Kemler, (A.S.M.E. Transactions, Vol. 55, 1933, Hydraulics, p. 7).

Mechanical Similitude and Turbulence, by T. von Karman (translated and reprinted as Technical Memorandum N.A.C.A. No. 611, 1931).

Turbulent Flow in Pipes, with Particular Reference to the Transition Region Between the Smooth and Rough Pipe Laws, by C. F. Colebrook, (*Journal, Institute of Civil Engineers*, Vol. II, 1938-39, p. 133).

Evaluation of Boundary Roughness, by H. Rouse (Proceedings Second Hydraulics Conference, *University of Iowa*, Bulletin 27, 1943).

FLOW OF AIR IN DUCTS

Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghton, J. B. Schmieler, J. A. Zalovcik, and N. Ivanovic (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 35).

Analysis of Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten, and H. T. Olson, (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 193).

The Flow of Air In Ducts, by E. Kemler, (Heating and Ventilating, May 1936, p. 38).

Total, Static and Velocity Pressure, by A. A. Berestneff, (*Heating, Piping and Air Conditioning*, March 1932, p. 195).

PRESSURE LOSS IN ELBOWS

Flow of Fluids Through Valves, Fittings and Pipe, (Crane Co., Technical Paper No. 409).

Pressure Losses for Fluid Flow in 90° Pipe Bends, by K. H. Beij, (*National Bureau of Standards*, Research Paper 1110, Journal of Research July 1938, p. 1).

Pressure Loss in Ducts with Compound Elbows, by J. R. Weske, (N.A.C.A. Wartime Report, Originally issued as A.R.R. Feb. 1943).

DUCT DESIGN

A Rational Method of Duct Design, by L. G. Miller, (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 71).

MISCELLANEOUS

Performance Tests of Asbestos Insulating Air Ducts, by R. H. Heilman and R. A. McArthur (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 197).

CHAPTER 42

SOUND CONTROL

Unit of Noise Measurement, Apparatus for Measuring Noise, General Problem, Kinds of Noise, Noise Transmitted Through Ducts, Design Room Noise Level, Noise Generated by Fans, Natural Attenuation of Duct System, Duct Sound Absorbers, Air Supply Noises, Grille Selection, Cross Transmission Between Rooms, Controlling Vibration from Machine Mountings

IN ventilating and air conditioning a building or a room, consideration must be given to the effect of the mechanical system on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

According to an international standard, two terms are used for noise measurement. The *decibel* (db) is the physical unit for expressing intensity or pressure levels. The *phon* is the unit of loudness level. The loudness level, in phons, of any sound is by definition equal to the intensity level in decibels of a thousand cycle tone which sounds equally loud. The decibel is defined by the relation $N = 10 \log_{10} \frac{I_i}{I_0}$, where N is the number of decibels by which the intensity flux I_i exceeds the intensity flux I_0. The intensity flux is the measure of the intensity of a sound wave and is defined in terms of watts per square centimeter passing through a unit area of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitrary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose a reference intensity of 10^{-16} watts per square centimeter has been selected. This intensity is slightly less than the threshold of audibility for the average ear at a frequency of 1,000 cycles per second. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter for sound in air at usual room temperatures.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is based on a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary

with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in Standards[1] published by the *American Standards Association*.

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying response of the ear at different frequencies in designing noise measuring equipment. It is most satisfactory to measure noise by means of a sound level meter, which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the standard reference level and usually contains a weighting network to make it less sensitive at those frequencies where the ear is less sensitive. Three types of weighting networks may be provided. The "A" network is intended to provide relative frequency sensitivity corresponding to the characteristics of the ear at a loudness level of 40 decibels. The "B" network is weighted to correspond to the ear characteristic at a loudness level of 70 decibels. The meter may also be provided with a "flat" network which provides no weighting by frequency. For complete details on standards for sound level meters, refer to the information[2] published by the *American Standards Association*.

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the problem confronting the air conditioning engineer is that of designing a system which will operate without increasing the noise level in the conditioned space. To be sure that this is accomplished, it is necessary:

1. To determine the noise level existing without the equipment.
2. To ascertain the noise level which would exist if the equipment were installed without sound control.
3. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in Item 1.

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.
2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.
3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have sufficient information to predict the levels produced by noises which may be transmitted by the duct system from one conditioned space to another or from an outside space to the conditioned space. In either case, the designer must know the probable noise level at the point where the noise originates. From this he can compute the attenuation or transmission loss required in order to bring this level down to that required in the conditioned space. If there is likelihood of direct transmission through a duct, the attenuation required may be computed as shown in section Noise Transmitted Through Ducts. If the transmission is through dividing walls, it will be necessary to refer to published data on losses through standard building constructions[3].

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation, although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products. A sound test code for fans has been developed by the *National Association of Fan Manufacturers* which uses the flat response network of the sound level meter. However, when determining the noise generated by an air distribution system, it is customary to use the noise level of the fan as determined by the weighting network most nearly approaching the noise level of the space. In most cases this will be the noise level of the fan as determined on the 40 decibel weighting network. Refer to Table 1 for typical noise levels.

KINDS OF NOISE

In solving a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source, and instead, places the emphasis on ascertaining the level at the point where the sound enters the room.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two general kinds depending on how it reaches the room:

1. Noise transmitted through the ducts.
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From air currents, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system.
   e. Noise produced by the grilles.
2. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE TRANSMITTED THROUGH DUCTS

Operation of an air distribution system results in the generation of noise which may be transmitted through the ducts to the ventilated or conditioned room. The transmission of this noise may be controlled by the proper application of sound absorptive material within the ducts. The application of the absorptive material is a problem in balancing the room noise level requirements against the intensity of the noise generated. The four steps in the problem are:

1. Determination of acceptable room noise level resulting from the operation of the equipment.

2. Determination of noise level generated by the equipment.

The difference between items 1 and 2 in decibels is the over-all noise reduction required between the equipment and the room. In the discussion which follows reduction of noise will be referred to as *attenuation* of noise.

3. Determination of the natural attenuation of the duct system.

4. Selection of the proper sound treatment for the duct system.

The difference in decibels between the over-all attenuation required and the natural attenuation (3) is the additional sound attenuation to be provided by ab-

sorptive materials installed in the duct system or by special constructions designed to absorb sound. Experience has shown, for example, that where ventilating requirements permit, introduction of an expansion chamber or a change in area in the duct will frequently provide further reduction in low frequency noise.

DESIGN ROOM NOISE LEVEL

Measurements of noise levels have been observed by several investigators in various rooms and locations and are listed in Table 1. The

TABLE 1. TYPICAL NOISE LEVELS[a]

Rooms	Noise Level in Decibels to be Anticipated		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Post Offices, etc	45	55	60
Court Rooms	30	35	45
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[b]	70	80
Pullman Car	55[b]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	75	80	90

[a]These values are tentative. More detailed measurements by D. F. Seacord, Bell Telephone Laboratories (*Journal Acoustical Society of America*, Vol. 12, pp 183-187, 1940) give average values and standard deviations of room noise in residences, offices, stores, factories, etc., in large American cities.

[b]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

values given were determined with the air conditioning or ventilation equipment out of operation, and with all windows and doors closed simulating the conditions of an actual installation. This is an important consideration, since, in offices or stores adjacent to busy thoroughfares the difference between the typical noise level in the space with the windows and doors open and closed may be as high as 10 db. Minimum, representative, and maximum levels are given for each type of space. The values are intended to give the variation with respect to location and not to time, and may be roughly classified as shown in the following paragraphs.

Minimum loudness refers to: Spaces of expensive construction, typified by double windows, carpeted floors, heavy upholstered furniture, or acoustically treated walls and ceilings.

Representative loudness refers to: Spaces of average construction and furnishings which are exposed to external noises typical of the locality in which the space is usually found.

Maximum loudness refers to: (1) Any space of inexpensive construction, and bare furnishings where noise is not an important factor. (2) Spaces in close proximity to very intense street traffic or to intense factory noise.

In general, if the noise level in the space resulting only from the operation of the air conditioning equipment is equivalent to or less than the typical level given in Table 1, the installation will prove satisfactory. If the typical level and the equipment level are equal and heard together the resultant level will be 3 db higher than either of them alone.

In some cases it is desirable to keep the equipment noise level in the ventilated or conditioned room at such a value that it actually will not increase the noise level in the room to any measureable degree. This can be accomplished if the equipment noise at the room can be kept 10 db below the noise levels shown in Table 1.

NOISE GENERATED BY FANS

Noise generated by fan wheels may be divided into two classifications, rotational noise and vortex noise. In ventilation and air conditioning work, where the maximum ratio between the fan tip speed and the velocity of sound is not greater than 0.12, vortex noise is by far the most important. The rotational noise may be described as that due to the thrust and torque applied to the air. Vortex noise is that due to the shedding of vortices from the blade and is dependent on the angle of attack, velocity, air turbulence, and blade shape. Vortex noise is due to pressure variations on the blade as a result of variations of air circulation. Given the noise level at the outlet or inlet of one type of fan construction under specific conditions of size, tip speed, and total pressure, noise levels at other values of tip speed, total pressure, and size may be approximated by the relationships:

1. For constant size and point of rating, the noise level of a fan will increase with increasing speed.

$$db\ (\text{change}) = 55 \log_{10} \left(\frac{RPM_2}{RPM_1} \right) \tag{1}$$

2. For constant pressure and tip speed, the noise level of a given type of fan will increase with increasing fan size.

$$db\ (\text{change}) = 20 \log_{10} \left(\frac{\text{Size}_2}{\text{Size}_1} \right). \tag{2}$$

Fan size refers to wheel diameter, housing height or some dimension that is directly proportional to lineal units. Fan sizes based on arbitrary systems or systems of preferred numbers have no significance.

The noise of a given fan is not constant at constant speed if the air delivery changes due to change of resistance. In general, a backward curved blade fan is lowest in noise at or near the point of maximum efficiency; a forward curved blade fan at or between the point of maximum efficiency and shut-off; an axial flow fan at or between the point of maximum efficiency and free delivery. The noise level of a double width fan may be

TABLE 2. ATTENUATION IN STRAIGHT SHEET METAL DUCT RUNS

DUCT	SIZE, IN.	ATTENUATION PER FT, db
Small	6 x 6	0.10
Medium	24 x 24	0.05
Large	72 x 72	0.01

taken as 3 db higher than a similar single width fan operating under the same conditions of speed and pressure.

NATURAL ATTENUATION OF DUCT SYSTEM

Straight Sheet Metal Ducts. The attenuation of sound in straight sheet metal ducts is a function of the length, shape, and size of the duct[4]. Attentuation values are given in Table 2. In general this attenuation is so negligible except for long runs that it may be disregarded for all practical purposes.

Elbows and Transformations. Due to reflective interference, attenuation will take place at elbows and transformations. The magnitude of the attenuation will depend on the size and abruptness of the elbow or transformation as shown in Table 3.

When the area of a duct increases abruptly, an attenuation of noise level takes place in the duct. In duct design practice the total area of the branch ducts is greater than the supply duct. Similarly with outlets, the area of the outlet plus the area of the duct after the outlet is greater than the duct area before the outlet. Therefore in an outlet run, attenuation occurs in the duct as it passes each outlet. Table 4 gives the *db* reduction for various ratios of total branch duct and outlet area to supply duct area.

Grilles to Room. The large abrupt change in area between the grilles and the surfaces within a room results in an appreciable noise attenuation. This attenuation is a function of the total grille area (supply and return) and the total sound absorption of the room in *sabins*. (The sound absorption of a room in *sabins* is the summation of the products of each surface of the room measured in square feet multiplied by its corresponding absorption coefficient. The *sabin* is a unit of sound absorption equivalent to the

TABLE 3. ATTENUATION OF ELBOWS[a]

ELBOW	SIZE, IN.[b]	ATTENUATION PER ELBOW, db
Very small	2 wide	3
Small	3 to 15	2
Medium	15 to 36	1.5
Large	36 plus	1

[a]The attenuation in vaned elbows should be considered the same as in elbows having the same dimensions as the radius of curvature of the vanes. If the vanes are lined for the purpose of damping any vibrations in them, one third may be added to the attenuation values listed.

[b]These attenuation values are based on elbows having a center line radius 1.5 to 2 times the diameter or width of the duct. The attenuation will be greater if the ratio is less than 1.5 and less when the ratio is greater than 2.

TABLE 4. ATTENUATION AT DUCT BRANCHES OR OUTLETS

RATIO $\frac{\text{Branch Duct + Outlet Area}}{\text{Supply Duct Area}}$ OR $\frac{\text{Sum of Branch Areas}}{\text{Supply Duct Area}}$	ATTENUATION PER TRANSFORMATION, db
1.00	0.0
1.20	0.8
1.35	1.3
1.50	1.8
1.75	2.5
2.00	3.0

absorption of one square foot of a totally sound-absorbent surface). The attenuation is given in Equation 3 as:

$$db\begin{pmatrix}\text{Attenuation between}\\ \text{grilles and room}\end{pmatrix} = 10\log_{10}\frac{\text{Total Room Absorption in Sabins}}{\text{Total Grille Area}} \tag{3}$$

Values in Table 5 approximate the attenuation for various rates of air change, and general types of room surfaces.

DUCT SOUND ABSORBERS

The difference between the required sound attenuation and the natural attenuation is that which must be supplied by the proper sound treatment of the ducts.

Selection of the Absorptive Material

When a sound wave impinges on the surface of a porous material, a vibrating motion is set up within the small pores of the material by the

TABLE 5. APPROXIMATE ATTENUATION BETWEEN GRILLES AND ROOM

OUTLET VELOCITY FPM	AIR CHANGE MIN.	LIVE ROOM[b] $a^a = 0.05$ db	MEDIUM ROOM[c] $a = 0.15$ db	DEAD ROOM[d] $a = 0.25$ db
500	5	11	16	18
	10	14	19	21
	15	16	21	23
	20	17	22	24
750	5	13	18	20
	10	16	21	23
	15	18	23	25
	20	19	24	26
1000	5	14	19	21
	10	17	22	24
	15	19	24	26
	20	20	25	28
1250	5	15	20	22
	10	18	23	25
	15	20	25	27
	20	21	26	28

[a]Average absorption coefficient for the room.
[b]Live room-average absorption coefficient 0.05. Bare wood or concrete floor—hard plaster walls and ceiling—minimum of furniture.
[c]Medium room-average absorption coefficient 0.15. Carpeted floor, upholstered furniture, hard plaster walls and ceiling or bare room with acoustically treated ceiling.
[d]Dead room-average absorption coefficient 0.25. Heavy carpeted floor. Walls and ceiling acoustically treated. Upholstered furniture.

alternating sound waves. As the ratio of the cross-sectional area of the pores to their interior surface is small, the resistance to the movement of air in the pores is large. This viscous resistance within the pores of the material converts a portion of the sound energy into heat. The decimal fraction representing the absorbed portion of the incident sound wave is called the absorption coefficient. Considerable absorption may also result, particularly in the low frequency range, from the flexural vibrations of the duct. In the selection and application of the absorptive material, several points should be considered.

1. For the absorption of the low frequencies below 500 cycles per second the material should be at least 1 to 2 in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb the high frequencies and reflect the low.

2. In order to provide as much low frequency noise absorption as possible by means of flexural vibration, it is desirable to fasten the absorptive panels discontinuously. This result may be attained to some extent by spot cementing, but better results are obtained when it is possible to fasten the absorptive panels to furring strips, leaving an air space behind. However, the exact resonance characteristics of the panels, and thus their absorption, are so unpredictable that flexural vibration cannot be relied upon for a specific value of attenuation.

Requirements for a good sound absorption material are: (1) high absorption at low frequencies[5], (2) adequate strength to avoid breakage, (3) fire resistance and compliance with national and local code requirements, (4) low moisture absorption, (5) freedom from attack by bacteria and algae, (6) low surface coefficient of friction, (7) particles should not fray off at the higher design velocities, and (8) freedom from odor when either dry or wet.

With every application, the use of sound absorptive material should be considered in the dual function of insulation and sound absorption. It has been shown theoretically[6] that the reduction, in decibels per linear foot, of sound transmitted through a duct lined with sound absorbing material is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental evidence likewise indicates that there is no simple formula involving the variables which will apply accurately to all cases. However, it may be stated generally that the attenuation in decibels at a given frequency is directly proportional to the length of lined duct. It decreases as the cross-sectional area increases, and increases as the aspect ratio is increased.

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, consideration should be given both to the comparative efficiency of the duct lining material at different frequencies, and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based on the frequency 256 cycles, since most of the noise energy is in the region of this frequency. In quieting noise due to air turbulence and eddy currents, in which the high frequencies predominate, the frequency 1024 cycles should be used.

Since ventilating system noise contains many frequencies, an exception should be noted to the statement above that attenuation in decibels is directly proportional to length of duct. Most sound absorbent materials are more efficient at high frequencies than at low frequencies. In consequence, the attenuation in the first five or ten feet of lined duct will be greater because the high frequencies are being absorbed. Thereafter, since low frequencies will be predominant, the over-all noise attenuation per foot will gradually be less.

Acoustic Impedance of Absorptive Materials

In the past five years considerable literature has collected describing methods of determining the acoustic impedance of sound absorbent materials and methods of utilizing this quantity for predicting the acoustics of rooms[7] and the attenuation of sound in ventilating ducts[8]. Acoustic impedance as a concept is derived from electrical circuit theory. The effect of the sound absorbent material upon incident sound waves is described in terms of a resistive and a reactive component which may be determined by specifically devised apparatus[9].

Generally speaking, however, the use of acoustic impedance theory involves rather elaborate mathematical calculations, and the improved accuracy obtained is largely off-set by variations in the materials themselves and in their methods of mounting. It has been difficult to measure the acoustic impedance of large areas of material mounted in a manner typical of standard construction. P. E. Sabine[7] concludes that the assumptions required by acoustic impedance theory make this method of calculation of no immediate practical advantage in the measurement of sound absorption coefficients.

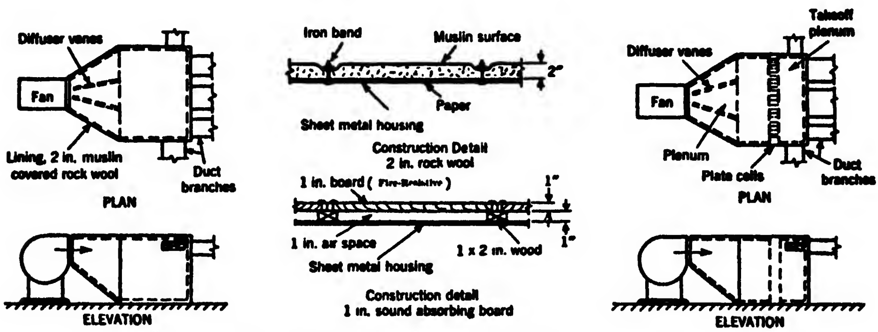


Fig. 1. Absorption Plenums With and Without Sound Cells

Beranek[8] compares results of sound attenuation observed for rectangular ducts lined with absorbent material computed by acoustic impedance theory with data reported by H. J. Sabine[10], using the methods of this chapter. Beranek concludes that the conventional P/A relation is valid for rectangular ducts not too far from square. His analysis indicates that other cases require more exact theory. However, it seems questionable whether the improvement in accuracy offered by the impedance method overbalances the additional computation time required and outweighs other sources of error such as variations between samples of material.

Plenum Absorption

In systems, where individual ducts are directed to a number of rooms and sound treatment is required in every duct, a sound absorption plenum on the fan discharge as shown in Fig. 1 will often prove the most economical arrangement. The absorption in the plenum may be approximated by Equation 4.

$$db\,(\text{Attenuation}) = 10 \log_{10} \frac{\text{Plenum Absorption in Sabins}}{\text{Area Fan Discharge}} \tag{4}$$

The area of the plenum should be at least ten times as great as the fan discharge area. The plenum should be lined with 2 in. of muslin covered

TABLE 6. END REFLECTION OF PLATE ABSORBERS

PERCENTAGE FREE AREA OF ABSORBER	ATTENUATION, db
50	1
40	2
30	4
25	5
20	6

rock wool blanket or 1 in. sound absorbing board preferably nailed to wood strips on the inside of the plenum. With such a lining the plenum is particularly effective in reducing low frequency fan noise. The absorption of the plenum in sabins is the sum of the products of each interior area of the plenum measured in square feet multiplied by its corresponding absorption coefficient.

Plate Cells

One of the most economical methods of applying sound absorbent material from the standpoint of both labor and material is the plate cell. The plate cell consists of ½ or 1 in. sound absorbent board, spaced on 2, 3 or 4 in. centers. The attenuation due to the plate cell may be divided into two parts. There is reflection at each end due to the change in area and the absorption at the ends. Values for this attenuation are given in Table 6 which depend on the spacing. There is also attenuation due to absorption of sound within the passages of the cell, which depends on the length and the spacing. The attenuation within the cell for 1 in. board neglecting the end effect is given approximately by Equation 5.

$$R = \frac{10La^{1.4}}{S} \tag{5}$$

where

R = attenuation, decibels.
L = linear length of duct, feet.
S = spacing in inches between plates up to 3 in.
a = absorption coefficient for the full thickness of the cell material. For typical value of a see Table 7.

An important objection to the plate cell is the increase in duct cross-sectional area required. Often on the fan discharge, particularly with unitary equipment, where a number of branch ducts take off, the plate cell may be installed with little or no difficulty.

TABLE 7. ATTENUATION FORMULAE FOR 1 IN. THICK TYPICAL DUCT LINING BOARD

FREQUENCY	ABSORPTION COEFFICIENT	ATTENUATION REDUCTION, db
256	0.37	3.0 L P/A
512	0.69	7.5 L P/A
1024	0.78	9.5 L P/A
2048	0.78	9.5 L P/A

Outlet Sound Absorbers

Outlet sound absorbers are rectangular or plate cells installed directly behind an outlet or they may be the lining of a pan or plaque outlet. They are particularly effective in the elimination of high frequency whistles which are generated by air flow in the ducts. They are also employed in large systems with long runs where only a few outlets near the fan require treatment. Frequently outlet cells are the only means of correcting existing noisy installations, as the duct sections directly behind the outlets may be the only sections accessible for treatment. (See Fig. 2).

Duct Lining or Rectangular Cells

One series of experiments[10] made on a commonly used type of duct lining material (1 in. rock wool sheet) has shown that, subject to certain restrictions, the attenuation of single-frequency sounds may be expressed by the approximate Equation 6. This equation is accurate within plus or minus 10 per cent for duct sizes ranging from 9 x 9 in. to 18 x 18 in., for

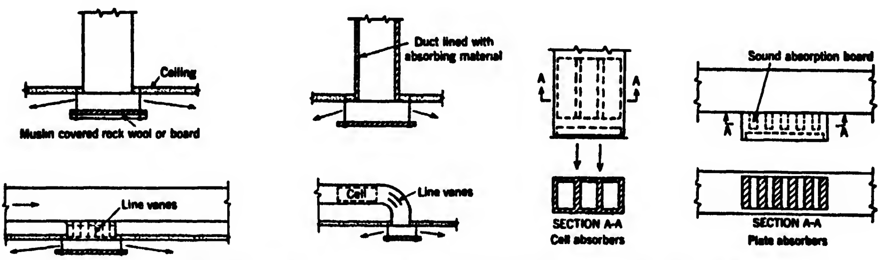


FIG. 2. OUTLET CELLS FOR PAN OUTLETS OR GRILLES

cross-sectional dimension ratios of 1:1 to 2:1, for frequencies between 256 and 2048 cycles, and for absorption coefficients between 0.20 and 0.80.

$$R = 12.6L\frac{P}{A}a^{1.4} \tag{6}$$

where

R = attenuation, decibels.

L = length of lined duct, feet.

P = perimeter of duct, inches.

A = cross-sectional area of duct, square inches.

a = absorption coefficient of lining.

In Table 7, the absorption coefficients at different frequencies of a material of the previously mentioned type are listed, together with the corresponding values for Equation 6.

Results of other experiments indicate, however, that Equation 6 may be in error when applied to other types of duct lining material and to duct sizes and shapes outside of the range specified. An empirically derived chart[11] representing the *average* experimental data on a number of different types of materials including the rock wool sheet mentioned as applicable to Equation 6 is shown in Fig. 3. Since individual materials vary, the curves in Fig. 3 are given only as representing the best available averages

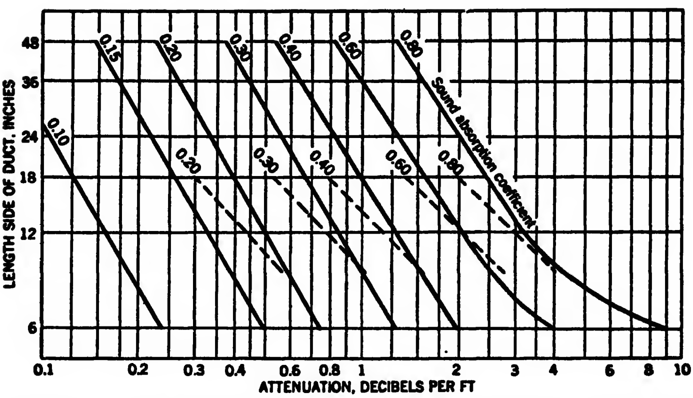


FIG. 3. SOUND ATTENUATION FOR VARIOUS ABSORBING DUCT LINERS

for duct sizes of square cross-sections from 6 x 6 in. to 48 x 48 in. As an illustration, the dotted lines in the chart show values calculated from Equation 6 which indicate that the slope for this particular material is somewhat different than from the average curves. The curves in Fig. 3, as well as Equation 6, show that the attenuation in decibels is directly proportional to the length of duct lined, and that the larger the duct the greater will be the length which must be lined in order to obtain a given noise reduction.

If the length of duct from the main duct to a grille is shorter than the length of lining indicated by the calculations, this duct may be subdivided into smaller ducts, as shown in Fig. 4. The increase in noise reduction thus obtained may be calculated from Equation 7, provided the splitters are installed parallel to the long side of the duct:

$$R_s = R_o \frac{a + bn}{a + b} \tag{7}$$

where

R_s = reduction with splitters, decibels.

R_o = reduction in same length of duct, without splitters, decibels.

a = dimension of short side of duct, inches or feet.

b = dimension of long side of duct, inches or feet.

n = number of channels formed by splitters.

Example 1. An air conditioning installation is to be installed in a small theater. Determine the necessary sound treatment for the air distribution system to provide a satisfactory noise level in the theater utilizing these conditions:

Fan tip speed 4000 fpm, total pressure 1.25 in.	77 db
Acceptable room noise level (Table 1)	40 db
Required attenuation	37 db

Solution: Natural attenuation of supply duct.

Sheet metal duct 50 ft long 48 in. x 36 in. (Table 2) 50 x 0.01	0.5 db
Elbows, two size 48 in. x 36 in. (Table 3) 2 x 1	2.0 db
Attenuation grilles to theater air change 10 min (Table 5) outlet velocity 1000 fpm	22.0 db
Total natural attenuation	24.5 db

Difference between required and natural attenuation, 37 minus 24.5, is 12.5 db. This attenuation must be supplied by sound treatment in the duct, either in the form of duct lining or plate cells.

A similar analysis of the return duct system shows that 15 db attenuation is to be furnished by absorptive material. An inspection of the installation shows that the lining of the plenum on the suction side of the fan would prove the most economical, where it would secure the dual function of heat insulation and sound absorption.

Example 2. A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily, using a lining material of a type to which Equation 6 applies, and having an absorption coefficient of 0.40 at 256 cycles. Assume that the duct is only 12 ft long as shown in Fig. 4, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters), From Equation 6,

$$R_o = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 12.6 \text{ db}$$

Case 2. (Two splitters, three channels), From Equation 7,

$$R_s = 12.6 \times \frac{10 + (20 \times 3)}{10 + 20} = 29.6 \text{ db}$$

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate[12]. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db.

Grille noise is similar in character to fan vortex noise. Knowing the noise level at the face of a grille for a given grille blade setting, the noise will vary as given in Equation 8 where V is the velocity of the air through the grille.

$$db \text{ (change)} = 55 \log_{10} \left(\frac{V_2}{V_1} \right) \tag{8}$$

For a change in blade setting Equation 9 applies and in this case the total pressure is measured directly behind the face of the grille. For a typical air conditioning grille the noise level at the grille face may be approximately 48 db with a total pressure behind the grille of 0.1 in.

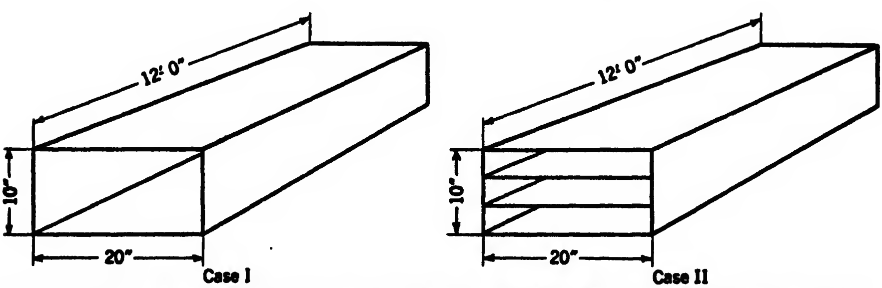


FIG. 4. DIAGRAM OF BRANCH DUCT TREATMENT WHERE LENGTH IS INSUFFICIENT FOR ADEQUATE ABSORPTION

$$db\ (\text{change}) = 27.5 \log_{10} \left[\frac{(\text{Total Pressure})_2}{(\text{Total Pressure})_1} \right] \tag{9}$$

The resultant room noise level can be approximated by Equation 10.

$$\text{Room Level} = \left[\begin{matrix} \text{Noise Level at} \\ \text{Face of Grille} \end{matrix} \right] - 10 \log_{10} \frac{\text{Total Room Absorption in Sabins}}{\text{Total Grille Area}} \tag{10}$$

Grille Selection

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. In comparing sound ratings of various grilles several factors must be known if the information is to be properly applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level: if not, the room level (without grille noise) must be known.
6. Methods used for recording data. (Characteristics of sound meter).

Since total loudness and air flow are both functions of velocity and area, the solution of the problem implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves, as illustrated in Fig. 5, which are based on these assumptions:

1. Threshold intensity = 10^{-16} watts per square centimeter[1].
2. Microphone location 5 ft from lower edge of supply opening on a line downward at 45 deg and in a plane bisecting the supply opening perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabins absorption.
5. Plotted data are loudness levels of *supply openings only*, correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure the total sound level of supply openings of more or less than one square foot area from Equation 11.

$$\text{Decibel Addition} = 10 \log_{10} A \tag{11}$$

where

A = core area, square feet.

With Fig. 5 it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular design of air supply opening. A correction chart is shown in Fig. 6 for a room having a sound absorption other than 100 sabins.

Example 3. Determine the core area (see Chapter 40) of an air supply grille which will maintain a noise level of not more than 40 db in a room having 100 sabins of sound

absorption, if an air volume of 2400 cfm is required to maintain the proper air conditioning.

Solution. **Assuming a grille noise rating of at least 5 db below the noise level of the room, Fig. 5 shows that the limiting grille velocity for a total loudness of 35 db is about 725 fpm and the core area becomes fixed at 2400 ÷ 725 or 3.31 sq ft.**

If the room absorption had been greater, the previously selected velocity of 725 fpm would be safe, since the loudness reduces. If the room absorption had been 200 sabins a correction of plus 1.3 should be made by reference to Fig. 6, and the permissible velocity becomes that corresponding to a total loudness of 36.3 or approximately 800 fpm.

If the room had been highly reflective with an absorption of less than 100, the correction would be much more important. For instance, for a room of 35 sabins a correction of minus 3 db should be made and the maximum velocity corresponding to the 32 db total loudness would be approximately 600 fpm.

Where more than one supply opening must be considered, the problem is more complicated. If a similar supply opening is added in a far corner

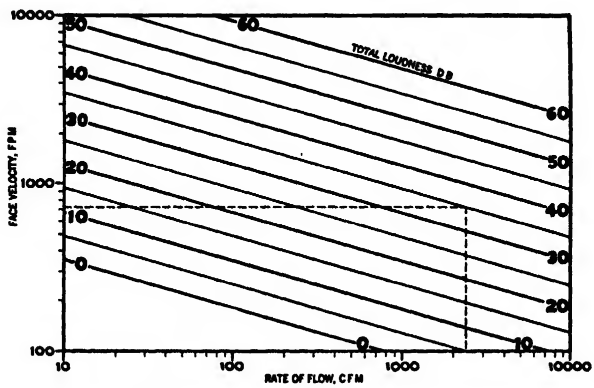


FIG. 5. AIR FLOW AND LOUDNESS CHART

of a highly absorbent room, the change in noise level at the 5 ft station at the first supply opening is small; however, if the room is small, or highly reverberant or both, the intensity at the 5 ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem is to treat the room as though all the air were being supplied by one supply opening. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with Fig. 5. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply openings. That is, if 1000 cfm are supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 5.

CROSS TRANSMISSION BETWEEN ROOMS

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide

ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts, serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct into the air stream and thereby carried into the room. In such cases, that portion of the duct which is exposed to

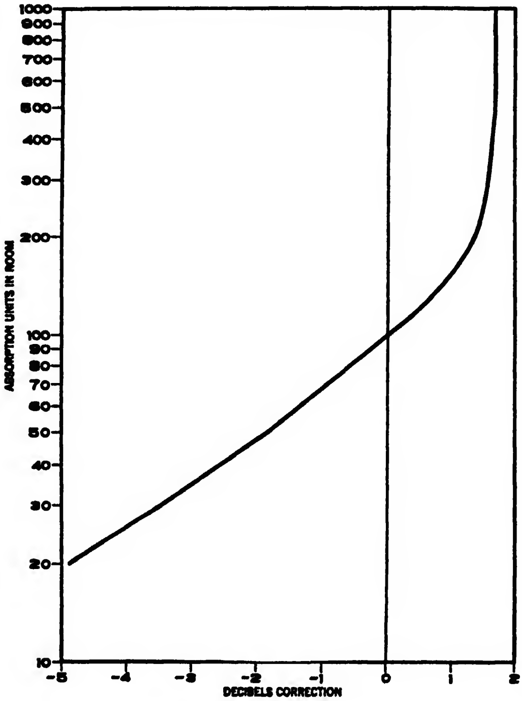


FIG. 6. ROOM ABSORPTION CORRECTION CHART

the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Standard reference books should be consulted for sound insulating properties of various materials. Inside lining material used in the case previously mentioned would serve as an

absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

CONTROLLING VIBRATION FROM MACHINE MOUNTINGS

It is impossible to select equipment which will operate without producing some mechanical noise and, since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the energy transmitted between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the energy transmitted between the equipment and the floor upon which it is supported.

In the proper isolation of vibration, which is usually in the lower range of frequencies and does not include the *airborne* vibrations known as sound, there is one basic formula which is important in the solution of the problem. It is the formula of transmissibility as governed by the equation:

$$T = \frac{1}{1 - \left(\frac{f}{f_n}\right)^2} \tag{12}$$

where

T = transmissibility of the support.
f = frequency of the vibratory force.
f_n = natural frequency of the machine unit on its support (Damping = 0).

Equation 12 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased the transmissibility is also increased until at the resonant frequency, where $f = f_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the resonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency the transmissibility becomes a smaller quantity and at the value of $f/f_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation is first accomplished. At a ratio of 3 to 1 for f to f_n the isolation is effective enough for practical application, and

experience and economical design have shown that a ratio of 5 to 1 is good. For high speeds, higher ratios for f to f_n are easily attained and give better results for effective vibration control, but for the lower speeds as experienced with compressor work the higher ratios become uneconomical.

For a given installation the speed of the compressor is fixed by the specifications, therefore the value of f is fixed. That leaves only f_n to be determined and that is accomplished by the choice of mounting material and design for the support of the machine. It is well to keep in mind that when trying to isolate vibration, no attempt should be made to isolate the driving and driven piece of equipment separately. The two should be mounted on a rigid frame and then the entire assembly isolated according to the rules presented in this chapter.

The value of f_n can be controlled by the flexibility of the machine support, and when the deflection of the machine support is proportional to the load applied (such as with springs or nearly so with rubber in shear) the value of f_n can be determined by Equation 13.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{d}} \tag{13}$$

where

g = gravitational constant.
d = static deflection of supporting material.
f = frequency of the vibratory force.
f_n = natural frequency of the machine unit on its support (damping = 0).

By the use of Equation 13 a set of curves may be plotted as shown in Fig. 7. The first line AB, plotted as the critical frequencies for the various static deflections, is a curve showing the worst possible conditions or resonant conditions.

Plotting another curve CD, which is $\sqrt{2}$ times curve AB, shows the area $MCDN$ in which the resilient material or mounting does more harm than good. Plotting two more curves EF, 3 times curve AB, and GH, 5 times curve AB, shows area $EGHF$ which represents efficient and economical isolation. Area $GPOH$ is excellent isolation, but for all except the highest speeds becomes rather uneconomical because of the large deflections required.

Example 4. An electric motor driven compressor unit is to be isolated. The compressor is partially balanced and operates at a speed of 360 rpm. The speed of the motor is 1160 rpm and is belt connected to the compressor. Total weight of the compressor and motor is 4500 lb.

Solution: The minimum disturbing frequency to be isolated is 360 cycles per minute. Assume that the desired ratio of forced to natural frequency is 3 as a minimum and that 5 is desired. The desired natural frequency of the mounting is 360 ÷ 5 = 72 cycles per minute.

From Fig. 7 a deflection of 7 in. is required to attain a natural frequency of 72 cycles per minute. This value may be obtained from critical curve *AB* for 72 cycles or from curve *GH* (5 times critical) for 360 cycles. For the minimum ratio of 3 the deflection would be 2.5 in.

The next step is to determine the total weight to be supported by the springs. For low speed partially balanced compressors, it has been found necessary to add a foundation weighing 2 to 3 times the weight of the motor and compressor, in order to maintain the machine movement below 0.03 in.

Compressor and motor	4,500 lb
Concrete foundation	9,000 lb
Total	13,500 lb

Practical application dictates the number of springs to be used, which is based on the design of the machine foundation and the supporting floor structure. However, it is desirable to design for at least 8 springs and one or two spares for cases of unknown weights. As many as 50 springs have been used on one installation. The distribution of the springs must be balanced against the masses to be supported, otherwise the foundation design and supporting structure determine the location of the springs.

The choice of the material used in the design of the resilient mounting is also important. For the slow-speed type compressor a common speed found in practice is 360 rpm. For speeds below this, isolation should not be attempted except under careful supervision. Referring to Fig. 7 , it is

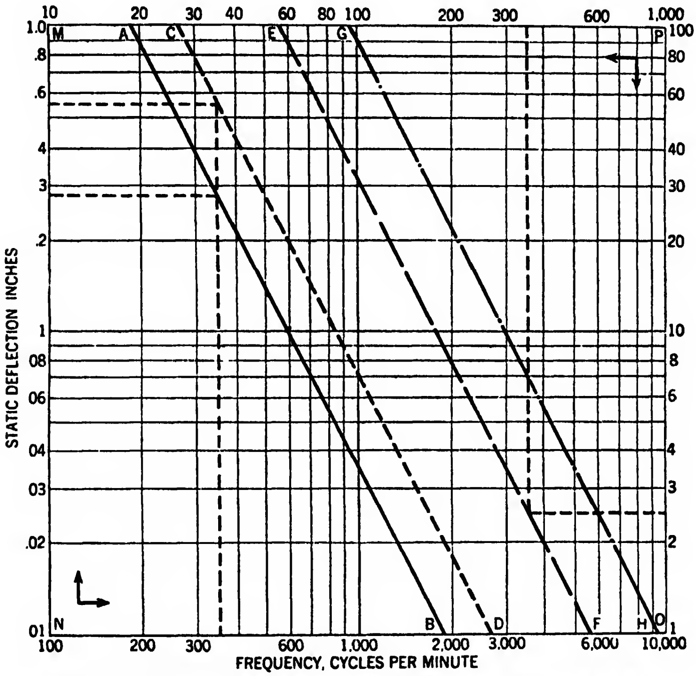


Fig. 7. Static Deflection for Various Frequencies

found that for 360 rpm the static deflection required for a ratio of f/f_n of 3 to 1 (line *EF*) is 2.5 in. and for a ratio of 5 to 1 (line *GH*) it is 7 in. For these values of deflection the only choice of material is the coil spring. This is also true for speeds up to about 700 rpm. In consideration of the transverse spring constant (so as to maintain good ratios among the various degrees of freedom) experience has shown that the spring should be designed with a *working height* equal to 1.0 to 1.5 times the outside diameter. A long spring of small outside diameter has very low transverse rigidity and therefore requires some additional means of preventing side drift of the unit and on very sensitive applications this may tend to destroy the isolation efficiency. For speeds of 700 to 1200 rpm the required deflections range from 0.22 in. to 1.75 in. For these conditions rubber in shear serves as a rather satisfactory material if protected from oil. For speeds higher than 1200 rpm cork specially made for vibration damping can be applied with good results. These limitations are by no means absolute, because certain

liberties may be taken without impairing the result if all possible degrees of freedom have been taken into account in the design of the installation.

When a machine unit is properly isolated it will have a definite amount of movement which is determined by the ratio of the unbalanced forces to the total mass of the machine. If this resultant machine movement is too great for the necessary connections or the satisfaction of the customer it can be reduced only in two ways without destroying the quality of the isolation; first, adding mass or dead weight to the machine (such as concrete) common in the application of low speed, partially balanced machinery; second, accurately balancing (both statically and dynamically) all moving parts so as to eliminate the vibration at the source. This latter method is the best engineering practice and is the modern trend. However, even with well balanced machinery, installed in the vicinity of quiet offices it is usually necessary to isolate properly the equipment to prevent the transmission of vibration likely to cause complaints.

Where limitation of machine movement is desired during the starting and stopping periods, the application of friction or hydraulic damping will serve without seriously interfering with the efficiency of the isolation.

REFERENCES

[1] American Standards for Noise Measurement, Z24.2-1942, *American Standards Association.*

[2] American Standards for Sound Level Meters for Measurement of Noise and Other Sounds, Z24.3-1944, *American Standards Association.*

[3] Sound Insulation of Wall and Floor Constructions (*U. S. Department of Commerce, National Bureau of Standards*, Building Materials and Structures Report BMS17 and Supplement).

[4] A.S.H.V.E. Research Report No. 1205—Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and R. F. Simons (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 267).

[5] For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Materials Association*, 919 No. Michigan Ave., Chicago, Ill.

[6] Sound Propagation in Ducts Lined with Absorbing Materials, by L. J. Sivian (*Journal Acoustical Society of America*, Vol. 9, 1937–38, pp. 135–140).

[7] Specific Normal Impedance and Sound Absorption Coefficients of Material, by P. E. Sabine (*Journal Acoustical Society of America*, Vol. 12, pp. 317–323, January, 1941).

Application of Wave Theory of Room Acoustics to the Measurement of Acoustic Impedance, by C. M. Harris (*Journal Acoustical Society of America*, Vol. 17, pp. 35–46, July, 1945).

[8] Sound Absorption in Rectangular Ducts, by L. L. Beranek (*Journal Acoustical Society of America*, Vol. 12, pp. 228–237, October, 1940).

The Transmission of Sound Inside Pipes, by Philip M. Morse (*Journal Acoustical Society of America*, Vol. 11, pp. 205–210, October, 1939).

[9] Precision Measurements of Acoustic Impedance by L. L. Beranek (*Journal Acoustical Society of America*, Vol. 12, pp. 3–14, July, 1940).

Notes on Acoustic Impedance Measurement, by H. J. Sabine (*Journal Acoustical Society of America*, Vol. 13, pp. 143–150, October, 1942).

[10] The Absorption of Noise in Ventilating Ducts, by Hale J. Sabine (*Journal Acoustical Society of America*, Vol. 12, p. 53, 1940).

[11] The Prediction of Noise Levels from Mechanical Equipment, by J. S. Parkinson (*Heating and Ventilating*, March, 1939, pp. 23–26).

Methods of Rating the Noise from Air Conditioning Equipment, by J. S. Parkinson (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, July, 1940 p. 447).

[12] The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 81).

CHAPTER 43

CENTRAL SYSTEMS FOR AIR CONDITIONING

Features of Systems, Zoning, Apparatus Dew-Point, Cooling Load, Heating Load, Air Quantity and Effective Temperature Difference, Low and High Pressure Induction Convectors, Evaporative Cooling, Precooling, Sensible Cooling with Unwetted Coils, Run-Around System, Selection of Type of System, Location of Apparatus, Design Procedure

THE term, central, applied to an air conditioning system implies that the equipment such as fans, coils, filters and their encasement are designed for assembly in the field rather than in a factory as a unit. As a central system usually serves several different rooms, individual controls are required for each room.

FEATURES OF CENTRAL SYSTEMS

One advantage of a central air supply system is that one apparatus serving many rooms may involve a lower investment cost than that for a number of self-contained plants, each serving a single room. A central system may occupy basement or attic space that is relatively unimportant, whereas individual factory-assembled apparatus placed in each room may occupy otherwise valuable space. Another advantage of a central system is accessibility for servicing, since it is possible to provide doors in the encasement for cleaning and inspecting all of the component parts in a manner usually superior to that practicable with compact factory-assembled equipment.

Central air conditioning systems usually are connected by ducts with the various rooms served, and preferably have exhaust fans that may effect complete removal and disposal of any desired proportion of the air. The exhaust fan may return air to the supply system for recirculation, as a measure of economy of fuel or refrigeration.

Central air conditioning systems are served by heating and refrigerating equipment which may be located at some distance from the air supply apparatus and which may serve one or more central air supply systems.

Year-Round Air Supply System

Fig. 1 is a plan of a year-round air supply system. Outside air may enter from the left at A, desirably from an intake on the side of the building least exposed to solar heat and not close to the ground or to a sun-heated or dust-gathering roof. A damper B for proportioning the volume of outside air, is interlocked with the return air damper C in such manner that as the outside air volume increases the return air volume decreases. The return air duct D, shown diagrammatically, comes from the exhaust fan. All the air, it will be observed, must pass through the filters E and there is ample room on both the inlet and outlet sides of the filters for servicing them. The filters may be of mechanically cleaned type, of replaceable cell type, or may be electronic, as described in more detail in Chapter 33.

The cleaned air passes to the equipment that changes its temperature and humidity. Except in very warm climates, a heating or tempering

coil F is required to warm the air to a temperature above freezing. Usually, the heat is supplied by means of hot water or steam. During many hours of most days it is practicable to recirculate enough of the air so that the air drawn from outside, after mixing with the relatively warm return air, will not be cold enough to freeze the water in the humidifier.

Upon leaving the tempering coil, the air enters the humidifier G. This may be a spray of warmed water, circulated by a small pump from a water tank under the spray chamber, or may be other means of supplying water vapor. The supply of moisture must be under automatic and very reliable control. Following the humidifier a heater I is required, for controlling the temperature of the air entering the supply fan.

The second group of heat transfer devices in a year-round system includes an air cooling component H, for use in warm weather. Its surface may be chilled by direct expansion of an approved refrigerant within

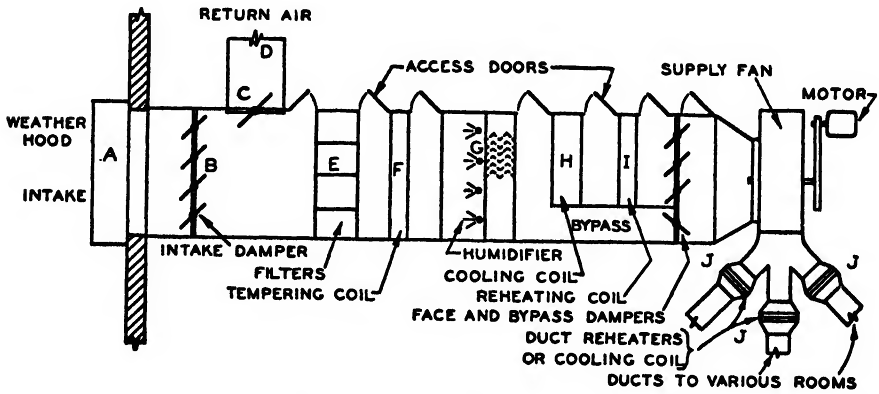


Fig. 1. Arrangement of Equipment for Year-Round Air Supply System

its tubes, or the surface may be cooled by a pump-circulated liquid such as water or brine. This device must be sufficiently cold to cool the summer air to a temperature below the existing dew-point, and may be expected to be wetted constantly by the moisture condensed from the air. A water-tight drainage tank must be installed under the cooling coil and should extend for a distance toward the fan. Water should be drained by means of a trapped waste through a vented air-break. The second group of heat transfer devices also includes an air-heating component (reheater) similar to the tempering coil and capable of warming the chilled, saturated air leaving the cooling surface, to a temperature sufficiently high to prevent complaint of drafts, when the air is delivered into the rooms.

ZONING AND ZONE CONTROL

It is apparent that while an apparatus like that of Fig. 1 would be very desirable for any single room, since in that case the air could be delivered at optimum conditions, the cost of a complete individual system for each room and the space required for the equipment generally would be prohibitive. Economy is favored if the varying requirements of numerous rooms or zones can be simultaneously satisfied by air from a single central supply system.

Various methods are practicable for controlling the temperature, humidity and air movement in various rooms or zones. A measure of control is attainable merely by proportioning the flow of air to each room, though usually such control by throttling dampers is difficult to maintain and should be avoided when possible.

Another scheme is to install a properly proportioned coil in the branch air supply duct serving each room to warm the air to suit the individual need. The air, for example, leaving the fan that serves several rooms, may be cooled before entering the fan, to the condition favorable for one room, and the air for each other room may be reheated by the branch duct coil to the required temperature. It is also possible to circulate a heat absorbing medium in the branch duct coils to reduce the temperature of the air passing to rooms that would be overheated if they received air at the condition leaving the central air supply system. In Fig. 1 such coils J are indicated in the three branch ducts leaving the supply fan.

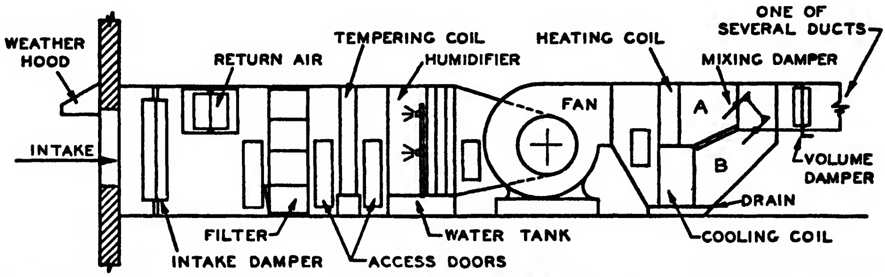


FIG. 2. ALTERNATE ARRANGEMENT OF EQUIPMENT FOR CONTROLLING AIR CONDITION IN CENTRAL AIR SUPPLY SYSTEM

When heat transfer devices are placed in branch ducts for improved temperature control, mechanically circulated water gives excellent results as a heat carrier. The water usually is warmer than the air but it is possible to use water colder than the air.

It is practicable also to use single central air conditioning equipment similar to that shown in Fig. 1, in conjunction with several fans; one for each room or zone. In such cases there may be a separate reheater on the suction side of each relatively small supply fan.

The designer must remember that the various supply fans will compete with each other for air, against the resistance interposed by the filters, coils, etc., that are used in common under such circumstances, and that, therefore, unless the fans are of backward-curved blade, non-overloading type, they may alternate in carrying more than their share of the air, and thereby cause the air distribution to be chaotic and unsatisfactory.

Another method of controlling temperature in various rooms served by a central air supply system is shown in the sectional elevation, Fig. 2. The supply fan is placed immediately after the humidifier. When cooling the air in hot weather, the humidifier is not operated. The fan will deliver the air through the heating coil and through the cooling coil to the two air pressure chambers A and B at the right of these coils. From these chambers many separate ducts, one of which is shown, each with a double-blade mixing damper, may convey the air to the various rooms. The mixing dampers, one of which is shown, are interlocked so that as the

upper one closes the lower one opens; selecting between them, air in the required quantity from either the warmer chamber A or the cooler one B. In cold weather no refrigerant is required in the cooling coil and in hot weather no heating medium is circulated in the heating coil. With this scheme, the control of relative humidity in warm weather is not always sufficiently precise to meet requirements, since the untreated air delivered through the upper coil may be so high in relative humidity that it cannot sufficiently compensate for the nearly saturated air leaving the lower coil. A reheater could be placed if desired, to the right of the lower coil to bring the air in the lower chamber to the desired relative humidity. The simple arrangement of Fig. 2 is admirable in winter and, except where close control of relative humidity is important, may be acceptable in summer.

Another method of attaining temperature control in individual rooms with a year-round central air supply system is to install a booster fan

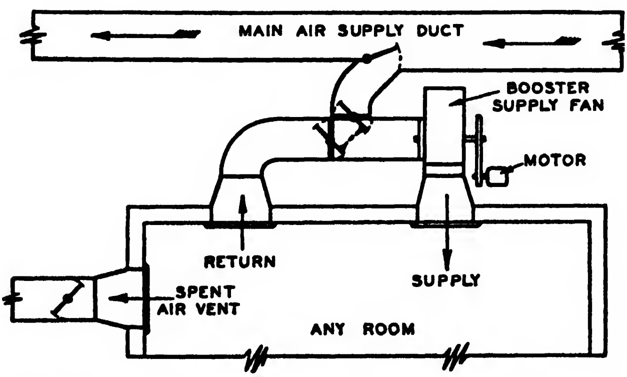


Fig. 3. Arrangement for Individual Room Temperature Control with Central Air Supply System

between the main air supply duct and the air delivery opening to each zone or room, as shown in Fig. 3. Air can then be delivered from the central supply fan through the main duct at some desired condition, for instance, 60 F, 45 per cent relative humidity. A double mixing damper near the intake opening of the booster fan, controlled by a thermostat in the room or zone that is served by the fan, is interlocked with an outlet exhaust damper in the spent air opening, so that as more of the room air is recirculated, and as less new air from the main air supply duct is delivered into the room, the spent air outlet is throttled in proportion. In many large installations this principle is applied successfully for zoning different stories in multi-story office buildings, the main supply fan being on the roof, and each booster fan used for supplying the rooms of one orientation of each story. In other cases the booster fans serve only single offices and therefore are small enough to be concealed above ceilings alongside the main supply duct.

There may be installations in which the use of recirculated air for mixing with new refrigerated and nearly saturated air to control temperature and relative humidity is objectionable. In such cases the general recirculation arrangements of Fig. 1 may be omitted, and the heat transfer coils located in the ducts may be used. In some cases where general recircu-

lation is not acceptable, as for all the rooms in an entire building, use of the local circulation of Fig. 3 may solve the problem.

APPARATUS DEW-POINT

In ordinary practice, with commercial apparatus, complete saturation of the air is seldom obtained. Four-row finned cooling coils contact approximately 80 per cent of the air, whereas six-row finned coils contact approximately 95 per cent of the air. In spray type dehumidifiers of good design the air leaves the dehumidifier at 1 to 2 deg higher wet-bulb temperature than the spray water leaving the dehumidifier, and the difference between the dry-bulb and wet-bulb temperatures leaving the dehumidifier may be as low as 1 deg. A spray type dehumidifier having sufficient length of spray chamber and density of spray, together with proper arrangement of nozzles, may approach saturation very closely.

As explained in Chapter 3, the slope of the line on the psychrometric chart connecting the room condition with the apparatus dew-point on the saturation line, determines the ratio of sensible heat absorbing capacity to the moisture absorbing capacity of the supply air. Therefore the room condition can be maintained as long as the supply air temperature lies on this line, but a greater volume of supply air must be used to satisfy the room load if the cooling coil does not contact 100 per cent of the air. For a given room load, the same apparatus dew-point will be required whether the cooling appliance contacts all the air or only part of the air.

From the point of view of satisfying the given cooling load requirements, the air passing through the apparatus without being cooled below the dew-point temperature produces two effects:

1. The air quantity which must be passed through the dehumidifier must be increased. Thus, if 20 per cent of the air passing is contacted, then 25 per cent ($0.20 \div 0.80 \times 100$) more air must be used than would be necessary if all of it were contacted.

2. Passing untreated air may change the room cooling load, which in turn may change the sensible heat factor. If return air only is passed through the dehumidifier or if room air only is by-passed, the room load would not change, but if some outside air is passed through, the room sensible heat gain and room latent heat gain will be changed due to the addition of untreated outside air, which changes the sensible heat factor. When a load calculation is made, it is necessary to know the percentage of air affected in the dehumidifier and calculation must be made accordingly.

If the ventilation air is drawn through the dehumidifier before it goes into the room, only that portion of the air not saturated must be included in the room load for the purpose of determining the apparatus dew-point and supply air quantity. It should be noted when evaluating the load added by untreated outside air that the temperature difference between room air and outside air and the moisture content difference between room air and outside air should be used, rather than the difference between outside air and apparatus dew-point, since the rise from the apparatus dew-point to room condition is charged against the dehumidifier as the cooling and dehumidifying load.

In winter, room relative humidities in excess of 30 per cent are seldom required in a system designed for comfort conditioning only, and a low saturating efficiency may be desirable, or even necessary, especially if the same volume of air is handled as in summer. With a spray type dehumidifier the main sprays may be shut off and only the eliminators need be

flooded; which may give sufficient moisture. In other cases, such as those in which cooling coils are sprayed, the spray water supply may be throttled. If the saturation efficiency of the sprays is too low, the spray water may be heated. The amount of heat put into the spray water by open or closed water heaters will be equal to that required to bring the dew-point temperature of the air entering the sprays up to that required before entering the preheater. It is possible, where clean steam is available, to introduce steam directly into the air stream to produce the desired dew-point temperature of supply air. However, the steam must be exceptionally clean, or objectionable odors will result.

It should be noted that the quantity of outdoor air to be introduced is affected by infiltration and leakage. Infiltration will reduce the quantity to be introduced by the system, while leakage may have to be offset by an increase in the quantity of outdoor air.

COOLING LOAD

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 15. As pointed out therein, many of the items of heat gain are variable and do not reach their maximum values simultaneously. Proper consideration of these peaks and the avoidance of pyramiding these peaks in the cooling load calculations are stressed. Maximum solar heat gain on an east exposure is seldom coincident with the maximum outdoor wet-bulb.

A large difference in the time-incidence of the peaks between various spaces or parts of the same space indicates the necessity for zoning. In a building having an east and west exposure where solar heat gain forms a fair share of the cooling load, the times of their individual peaks are apt to be some hours apart, and the peak load of one plus the off-peak load of the other will be substantially less than their combined peak loads. Proper zoning will permit operation to take full advantage of this condition or of similar conditions of non-simultaneous peaks and will result in a lower total load and in savings in equipment.

A factor, similar in effect and closely related to the non-simultaneous occurrence of peak loads, is diversity. Typical of this is the case of a large department store where the air handling equipment serving a certain space must be sufficient to handle the load created by the throngs of people attending sales in that space. Under such a condition the number of people in other spaces is usually normal or below. While this means that the air handling equipment for certain departments must be large enough to cope with the situation, the refrigeration equipment need be only large enough to handle the average maximum. If a system employing zone recirculating fans and a single central fan and dehumidifier were used, the saving would be reflected in the capacity of the central fan and dehumidifier. Another example of this diversity is found in an office building having restaurants and stores of certain types in the first story and basement. At noon, when the restaurants and stores are crowded, the offices are below normal occupancy.

Heat lag should be carefully considered in the cooling load calculations. In certain types of buildings the effect of solar radiation is still apparent several hours after the sun has shifted from that exposure. In other types having a much lighter construction, the heat gain due to solar radiation decreases markedly with the passing of the sun. Some walls, having been warmed by the sun, may radiate heat long after the passing

of the sun, thus requiring lower inside temperatures to offset the radiant energy.

Buildings have considerable heat storage capacity which can often be utilized to great advantage and which has more than once provided an unexpected safety factor. If a space is kept below the design inside temperature for some time, the interior walls, floors, furniture and fixtures begin to assume the temperature of the space. Where the time is sufficient, the entire mass rather than merely its surface may reach the room temperature. Thus, when a space has been precooled below the design maximum temperature for a period of time prior to the advent of the peak load, and the heat gain begins to increase the peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data this allowance is often a matter of experience rather than calculation.

Where air conditioning supply and return ducts pass through unconditioned spaces, there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 41). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in the space through which the duct passes, this is not a loss and usually can be compensated for by adjustment of air quantities between the various spaces.

HEATING LOAD

Methods of calculating the heating load are shown in Chapter 14. Many of the factors outlined previously under Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirements instead of the cooling requirements. However, these factors affect the heating load from the standpoint of control of inside conditions, over-all performance, and economy of operation more than from a capacity of equipment standpoint. It is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, and when there are practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly when sudden cold follows relatively warm weather, as may occur after a week-end or holiday shutdown. However, in normal operation during week-ends and holidays, buildings are usually kept at a holding temperature to prevent the freezing of services. In many cases, less fuel is required to continue operation of the heating plant at a near-normal rate and maintain the building or space at a temperature of 50 to 65 F for some time than to shut the system down and then bring the temperature back to normal through forced operation of the heat generating equipment with a consequent loss in efficiency.

AIR QUANTITY AND EFFECTIVE TEMPERATURE DIFFERENCE

The difference between the room air temperature and the supply air temperature at the outlet to the room is known as the effective tempera-

ture difference. In the theoretical case of a dehumidifier having 100 per cent saturating efficiency and where this air is delivered directly to the room without temperature increases due to heat gain, then the effective temperature difference is the difference between room temperature and apparatus dew-point temperature. If duct heat gains are considered a part of the room load, this still holds true. The apparatus dew-point, as outlined previously, is fixed by the latent and sensible loads of the space, but in many cases, it is desirable to deliver more air to the spaces than is indicated by the difference between the room temperature and the apparatus dew-point.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated, the relationship is modified in direct proportion, and that if room air is passed through untreated, no effect on the heat balance results. Similarly, if room air is passed around the dehumidifier and mixed with the treated air the heat balance is not adversely affected. Therefore, if the quantity of air passed through the dehumidifier is determined by the usual methods, room air can be passed around the dehumidifier and mixed with the dehumidified air, increasing the supply air quantity and temperature and decreasing the effective temperature difference. Thus if the difference between the room temperature and the apparatus dew-point indicates that 10,000 cfm at 30 deg below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 deg below the room temperature, then mixed with 10,000 cfm of room air; resulting in a supply air quantity of 20,000 cfm and an effective temperature difference of 15 deg instead of 30 deg. Air supply outlets and grilles that have a high induction ratio are available and cause a large amount of room air to be mixed with the air leaving the outlet within a short distance of the outlet through the induction effect of the air stream. A proper selection of outlets may make it possible to introduce air at low temperatures and high velocities without causing objectionable drafts or cold spots, but care must be used to see that too little air motion is not a result. Low effective temperature differences may be required for this reason. While the use of a high effective temperature difference results in a saving in initial cost of fans and ducts and in the operating cost of fans, this difference should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, low effective temperature differences should be considered, since systems employing low effective temperature differences require less precision in controls.

Reduction of air quantity by slowing down the fans for the winter season and increasing the temperature difference often is feasible. A saving in fan power can thus be effected, provided the air distribution remains adequate.

Extremes should be avoided in all cases. For summer air conditioning, low supply air temperatures result in larger heat gains to the air passing through the ducts, as well as in poor control. Too high a supply air temperature may result in excessive initial and operating costs. Suggested limits for the effective temperature difference are from 12 to 20 deg; the actual selection being based on the requirements of the particular case. For winter air conditioning, too high supply air temperatures result in excessive heat losses from the ducts and stratification within the room unless thorough mixing is assured, while too low supply air temperatures may cause drafts, high operating costs, etc. Suggested limits are from

15 to 35 deg. There can be no set rule and each case should be judged according to its particular requirements of the installation.

Reference may be made to Chapter 40 for further discussion of the most satisfactory design difference between the entering air temperature and volume in relation to the desired room condition.

INDUCTION CONVECTORS—LOW PRESSURE TYPE

Induction convectors, located in the room that is to be served, utilize a jet of conditioned air (or primary air) to induce a flow of room or secondary air which mixes with the primary air Fig. 4. The mixture is discharged into the room through a grille at the top of the convector. Heating coils are located in the secondary air stream. The output is controlled either by manually or automatically throttling the air jet. Heat may be supplied to the coil in summer as well as in winter. These induction con-

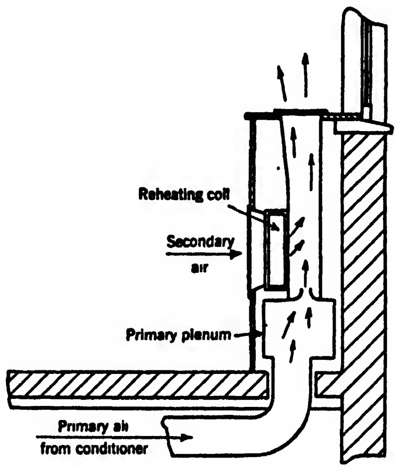


FIG. 4. INDUCTION UNIT (LOW PRESSURE TYPE)

vectors present several advantages. Since the secondary air stream is thoroughly mixed with the high velocity low temperature air stream before leaving the discharge outlet of the device, the resultant temperature of the mixture is satisfactory even though the primary air is introduced at a temperature too low for ordinary methods of distribution. One of these devices usually is provided under each window in place of the customary direct radiator, and combines the air distribution system with the heating system. With a conventional system it may be necessary to provide supplementary heating in the form of direct radiation. Induction convectors may be selected with heating coils having sufficient capacity under gravity conditions (that is, with the fan system shut off and no primary air entering the device) to maintain the room at a reasonable temperature in winter. The use of low temperature, dehumidified air which has not been reheated or mixed with room air before delivery to the room may permit a reduction in fan capacity and the use of smaller ducts. In some cases a by-pass may be desirable in order to maintain the primary air volume and to provide additional control. This system can provide a degree of zoning that is usually difficult with conventional design since the air delivered by each unit can be controlled individually.

Selection of induction convectors should be made with due regard to noise level. The inductive capacity of the device increases with the jet velocity, but high jet velocities may result in objectionable noise.

INDUCTION CONVECTORS—HIGH PRESSURE TYPE

Another type of induction convector Fig. 5 employs nozzles which produce a high velocity air jet without objectionable noise. The term, high pressure, is to some extent inaccurate, since the air pressure at the nozzles, while several times that used with a low pressure induction convector, is still less than the total resistance pressure of a conventional central system. The high velocity jet of primary air induces a flow of air from the room through coils located in the secondary air stream and

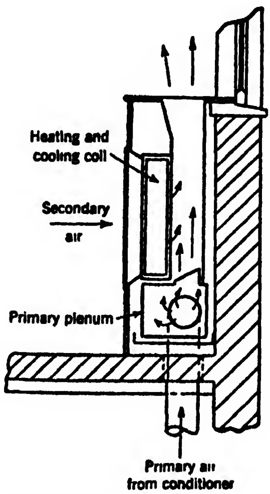


Fig. 5. Induction Unit (High Pressure Type)

supplied with chilled water in summer and with hot water in winter. The chilled water removes a large portion of the sensible heat in summer and the hot water supplies the sensible heat loss in winter. The primary air is delivered at a sufficiently low dew-point to compensate for the latent heat gain in summer. In winter the primary air is supplied at a sufficiently high dew-point to take care of latent heat losses. Control of temperature is obtained by throttling the water quantity supplied to the secondary coil. The required flow of primary air is greatly reduced due to the fact that a portion of the sensible heat load is carried by the secondary air stream. Since the primary quantity is small, very high velocities can be maintained in the supply ducts without requiring fan power in excess of that for a conventional system. Therefore, the supply ducts or pipes can be very small and can be run in chases, or furred in at columns along with the water pipes. The primary air is treated in the usual manner to reach the required dew-point and a surface or spray dehumidifier or a dehydrator may be used. The primary air quantity is sufficient for ventilation purposes and frequently consists entirely of outdoor air. The water piping for the coils can be so valved that hot water will be supplied to one zone that may require heating while cold water may be supplied at the same time to a zone that requires cooling.

This system usually is limited in application to hotels, apartments, office buildings and other multi-room installations having a large perimeter with relation to the floor area. The units usually are installed beneath the windows, replacing direct radiation or thermally-circulating enclosed convectors. Where the spaces to be conditioned extend a considerable distance from the outer wall into the interior of the building, a separate system or zone for the conditioning of the interior portions may be required.

EVAPORATIVE COOLING

In climates where, on the hottest days, the outdoor wet-bulb depression is relatively great, it may be possible to dispense with refrigeration or other cooling sources by use of the evaporative cooling effect. A well designed air washer using recirculating sprays will reduce the entering dry-bulb temperature to within a degree or two of the entering wet-bulb condition. Thus, it may be possible that, with air entering at 100 F dry-bulb, 60 F wet-bulb, a leaving condition of 62 F dry-bulb, nearly saturated, can be obtained. Under some conditions of latent and sensible heat load, evaporative cooling may be adequate.

At times when the outdoor wet-bulb temperature is not quite low enough to permit the use of straight evaporative cooling it is possible to use precooling convectors with refrigeration, well water or a cooling tower as the basic source of sensible heat removal to reduce the wet-bulb temperature of the air before it enters the spray chamber. Where internal heat loads are high, this scheme may be more economical than one using return air. Where the required supply air dew-point is too low to permit straight evaporative cooling and where the sensible heat load is not too great, intentional partial saturation may be employed. That is, the low dew-point of the outdoor air is utilized by permitting some of this air to pass through the humidifying sprays untreated, or to by-pass the humidifier. All of these remarks with regard to evaporative cooling are based on the assumption that all of the supply air will be taken from outside. Provision should be made in most cases for the return of some air from the conditioned spaces for control purposes as well as for economy of fuel in winter.

PRECOOLING

Where sufficiently cold water from wells or streams is available, a saving in refrigeration may be obtained by the use, in location ahead of the dehumidifier, of precooling coils through which the cold water is circulated. The resultant cooling of the air decreases the load to be carried by the dehumidifier and refrigeration plant. In normal practice the water, after passing through the precooling coils may be further utilized in the refrigeration plant condenser. The economic advantages of this scheme are apparent and it is frequently used.

SENSIBLE COOLING WITH UNWETTED COILS

Under favorable atmospheric conditions where a large wet-bulb depression exists and the dew-point of the outdoor air is sufficiently low at all times, acceptable cooling may be obtained by removing only the sensible heat from the outdoor air delivered to the rooms. Under this condition of a great wet-bulb depression, a temperature-reducing coil may be located in the air stream and supplied with water from a cooling tower. When humidity control is desired, sprays to saturate or partially saturate the air

may be used down-stream from the unwetted coil. Saturation or partial saturation after the coil will reduce further the dry-bulb temperature and the air quantity required. This system has very definite application in hot dry climates.

RUN-AROUND SYSTEM

An interesting method of control is found in the use of combined reheating and precooling usually termed the run-around system. Typically three coils are placed in series in the air stream. The primary one receives liquid that has been cooled in the third coil. The center coil is maintained at a temperature colder than the dew-point of the air. The primary coil thus pre-cools the air and the third coil reheats the saturated air from the center coil. The third coil is heated by the relatively warm water coming to it from the primary coil. The run-around scheme has the advantage of permitting a higher supply air dew-point temperature than would be possible otherwise. This is due to the fact that continual reheating is available, which is not a large penalty on the refrigeration plant since it provides precooling at the same time. This reheating at peak load creates an artificial sensible heat gain which increases the ratio of sensible heat to total heat and for a given room temperature results in a higher apparatus dew-point. Thus, while the volume of supply air is increased, the low-side temperature level of the refrigeration plant is raised and this may effect savings in initial and operating costs. The run-around system has the disadvantage of providing a decreasing amount of heat for reheating as the demand for reheating increases.

SELECTION OF TYPE OF SYSTEM

If the perimeter of the building is large with regard to the area, and if there are many rooms, induction convectors of either the low or high pressure type may be employed. Occasionally a dual system, one duct carrying air at a warmer temperature than the other, may be considered.

Low buildings with large floor areas may be divided into sections or zones with separate central air supply systems to facilitate temperature control. In the case of large department stores it may be possible to provide a single conditioner with a fan delivering the conditioned air to local mixing fans which supply the various departments or spaces. This application is limited by the practicability of running the large conditioned air ducts to the various recirculating fans. Each vertical section of the building also may be supplied by a separate fan delivering conditioned air to local mixing fans. In many cases the most economical and satisfactory scheme may be to employ a hot water or steam reheater in each branch duct to each room. Where vertical sectionalizing is not indicated, the building may be divided into horizontal groups, each handled by a central system and adequately zoned. In some large buildings, apparatus rooms for the systems may be located in the basement and in the attic and on intermediate floors.

In high buildings the necessity for horizontal sectionalizing may be suggested by the size of air supply and return risers and by the extent to which they encroach upon usable space. Each story should be cut off by doors from other stories as otherwise the cool air tends to collect in the lower story and the warm air is forced to the upper story.

Balconies and large lobbies in theatres and similar high rooms frequently justify the use of separate zoning fans, to counteract the tendency of the heavier, cooler air to collect at the lower levels of these spaces.

Fans operate at full capacity continuously in many systems and therefore should be selected for good efficiencies. In winter when higher temperature differentials are used, it is sometimes practicable to deliver smaller air quantities than when cooling.

In climates where winter temperatures fall below freezing, the tempering coils should be of the steam-distributing type; or if they are heated by a liquid, this liquid should contain some anti-freezing substance such as ethylene glycol. If hot water is employed in cold climates, the temperature control of the air should be obtained through use of face and by-pass dampers rather than by throttling the valves, to prevent damage due to freezing.

If zone reheaters placed in supply duct branches are employed, they should be of such type as to be heated over the entire surface so that no temperature-stratification can occur in the delivered air. Steam distributing-tube coils or mechanically circulated water coils are serviceable in such cases and throttling valves may be used.

Refrigeration equipment must be carefully selected to satisfy the particular requirements of each installation. For some small plants the evaporator may be placed in the air stream, when type of refrigerant and nature of occupancy permit. In many cases, chilled water coils are required by considerations of safety. Where low temperature and relative humidity are necessary, brine, often of calcium chloride, may be indicated.

Condensing requirements must have economic analysis. Wells, public water service, cooling towers and evaporative condensers present possibilities for consideration. Condenser water may have a secondary use for roof sprays in hot weather, and is usually suitable for lawn sprinkling. Most health department rules in cities prohibit any connection from refrigerant condensers that might permit the water to be used for drinking or lavatory purposes.

Practically without exception, air cleaners should be provided for both outside and recirculated air.

Control of temperature and of relative humidity by automatic means is vital, if comfort and economical operation of air conditioning equipment are to be attained.

The insulation of ductwork is not merely a matter of economics but sometimes is a necessity from the standpoint of limiting the temperature change of the air between the conditioning apparatus and the point of final delivery. Such temperature change of the air should be taken into account when apportioning the air and sizing the ducts. When computing heating or cooling loads, due allowance must be made for the effect of any hot or cold ducts or pipes contained in the space under consideration and insulation must be incorporated where necessary or justified. Consideration must also be given to the possibilities of condensation of moisture on either the inside or outside surfaces of pipes, ducts, housings, fan encasements, etc., and insulation should be applied to prevent corrosion and water damage and to conserve refrigeration.

The location of the apparatus room often is determined by building construction or available space. The closer the apparatus room is to the conditioned space, the less expensive are the ducts. If the equipment is noisy, it should be located at some distance from the occupied spaces or be provided with adequate sound and vibration treatment. The scattering of wet apparatus throughout a building is to be avoided unless suitable precautions are taken. It must be remembered that encroachment on

spaces that are otherwise usable can be charged against the system as an operating cost.

In general, the apparatus should be arranged to have straight line air flow. Each change in direction is the cause of air resistance and, in addition, elbows and offsets may cause eddy currents resulting in stratification. The usual order of equipment location, beginning at the outside air intake, is: weather hood of louvers, outside air dampers, return air connection, filters, tempering coils, cooling coils or sprays, by-pass connection with or without reheaters, reheaters, fan and distributing ducts.

Screens at the intake prevent the entry of large foreign matter, birds, etc. A hood or louver at the outside air intake prevents the entry of rain and snow. Since in most climates there are many days during which use of 100 per cent outside air unheated or uncooled may be economical, the areas of all air-passing and treating apparatus should be large enough for such a volume, and the exhaust or spent air equipment should be capable of discharging out of doors, all of the air admitted.

The by-pass connection normally connects the return air duct system with the apparatus casing between the conditioner and the supply fan. Usually the by-pass opening is sized to handle about 50 per cent of the fan capacity where a variable by-pass is used, though extreme load variations may require a larger size. It is at times good design to locate a reheating coil in the by-pass connection to permit using some by-pass air when heating is required. Since the relatively high resistance of the cooling coil or spray is to be balanced by the heating coil and by-pass connection, enough heating surface can be provided to raise the temperature of the by-pass air to the point where the mixture of by-passed air and conditioned air will have the required temperature. When a variable by-pass is used, a damper working in opposition to the by-pass damper should be placed across the face of the dehumidifier, for unless the resistances of the two are carefully balanced at all operating points, the proper mixtures of air will not be obtained. Outside air that has not been dehumidified should not be by-passed around a cooling coil or spray dehumidifier if accurate control of the delivered relative humidity is desired. Where the by-pass is made a part of the dehumidifier or conditioner and is located on the top or side of it, the return air connection should be arranged so that stratification of return air is insured, baffles being provided to accomplish this purpose if necessary. Where return air and by-pass air connections are taken off a return duct system, it may be necessary to install a back-draft damper between the return air connection and the by-pass connection. When the by-pass damper is at maximum opening it may be much easier for outside air to pass through the return damper, into the return duct connection and through the by-pass than for return air to pass through the by-pass connection into the fan. Air tends to take the path of least resistance and, if the dehumidifier resistance is high, and if the return duct resistances are low, this situation is apt to occur. A recirculating air fan instead of a back-draft damper may be required for this case if the failure of return air to reach the dehumidifier or conditioner is a serious matter under reduced load conditions.

LOCATION OF APPARATUS

In general, the outside air intake, preheaters, and return air connections precede the conditioner, while the by-pass, reheaters and fan follow it. In the case of a blow-through system, where the fan is located ahead of the conditioner, the leakage of air at the conditioner is outward instead of inward and may be accompanied by water leakage.

The location of the complete apparatus assembly including the dehumidifier will be dependent on the type of building, spaces available, structural characteristics, etc. The type of conditioner used may limit the location under certain conditions. Where cooling coils employing chilled water or brine as the cooling agent are used, there are few limitations with regard to location other than those of pumping power, working pressures, piping costs, etc. Where spray dehumidifiers are used, very definite limitations present themselves, and these may require certain extraneous equipment to make the system workable. If several spray type dehumidifiers are located on different levels, thus involving different water pressures, a surge or storage tank to which the return water from each dehumidifier can be taken is required. Should the water level in the pan of the dehumidifiers be low in relation to that of the surge tank, return water pumps will be required, and these pumps will have to be operated until the water supply lines are drained in order to prevent flooding of the lower dehumidifiers. Where spray dehumidifiers are on the same level, equalizing lines between the pans may be required if a storage tank is not provided. It is exceedingly important that water-tight drained floors shall be provided under all overhead cooling systems, since water condensed out of the air generally will be present and may damage the interior finish of the rooms below the apparatus.

All of the various pieces of equipment from the outdoor air intake through the fan usually are connected together by sheet metal casings. Frequently the building structure or specially constructed walls or partitions may be used to form a portion of the casing. In any case the casing or connection must be sufficiently sturdy for the required duty. Sheet metal work must be well braced not only to prevent vibration under pulsations in air flow but also to withstand the abuse of normal usage. Casings should be braced wherever access doors are installed and all large panels should be adequately reinforced by structural steel.

Accessibility for Service

Each apparatus layout is to be made with accessibility in mind. Where cooling convectors are used, space for removing and repairing or replacing them should be provided. Adequate space is to be provided for the servicing and replacement of eliminators. Whether these accompany sprays or wetted coils, filters must be so located that the proper cleaning, replacement or routine servicing can be accomplished without difficulty. Free access to the bearings of all moving machinery is a necessity. Provision should be made for the complete removal and replacement of any parts of the apparatus that are subject to wear, deterioration or damage, whether it may be filter, fanwheel, motor, pump, impeller or heat transfer surface.

DESIGN PROCEDURE

The customary design procedure is outlined herewith. For simplification the procedure is set up on the basis of a year-round system. For systems designed only for winter or summer, the unrelated parts may be omitted.

1. **Selection of design conditions (inside and outside).**
   - a. **Summer.**
   - b. **Winter.**

2. Determination of outside air requirements.
3. Determination of cooling load.
   a. Room sensible heat gain.
   b. Room latent heat gain.
   c. Room total heat gain.
   d. Grand total heat gain.
4. Determination of heating load.
   a. Room sensible heat loss.
   b. Room moisture loss.
   c. Humidification requirement.
   d. Total heating requirement.
5. Determination of apparatus dewpoint and dehumidified or humidified air quantity.
   a. Summer (full load and part load).
   b. Winter.
6. Supply air temperature difference and quantity.
   a. Summer.
   b. Winter.
7. Equipment selection.
8. Equipment layout.

The foregoing steps are merely typical. Many applications will require at least a preliminary investigation of some of the latter steps before proceeding with the earlier steps. A permanent record of all design assumptions and computations should be made and preserved for comparison with the performance of the installation.

CHAPTER 44

OWNING AND OPERATING COSTS

Fixed Charges: Amortization, Interest, Taxes, Insurance, Rent; Maintenance Costs, Service Costs: Operating Refrigeration Equipment, Condenser Water, Heating

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THE purpose of this chapter is to discuss the owning and operating costs of heating, ventilating, air conditioning and refrigeration systems for buildings from an economic standpoint so that owners or prospective purchasers may compare the operating economics of one system with another and evaluate properly the over-all costs of the systems instead of considering only the first cost.

There are cases in which it may be desirable to study the possibilities of installing a system for the purpose of obtaining a substantial increase in income or a better return on the investment due to: increased patronage in theaters, stores, or hospitals; increased occupancy in office buildings; improved efficiency of employees in offices or factories; or improvement in a manufactured product or a decrease in its cost of production.

Owning and Operating Costs may be grouped under three headings: (1) *Fixed Charges*, (2) *Maintenance Costs*, and (3) *Service Costs*.

FIXED CHARGES

Fixed Charges, which are the costs of owning the system, include: (1) Amortization, (2) Interest, (3) Taxes, (4) Insurance, and (5) Rent.

Amortization

Amortization cost will depend on: (1) the total first cost, and (2) the amortization period.

The *total first cost* of an installation is the actual dollar outlay or capital expenditure required to buy and install the air conditioning, or heating and ventilating system ready for operation. It can be divided into two parts: (*a*) The first cost of the air conditioning or heating and ventilating system itself, and (*b*), other first costs incurred because of the installation of the air conditioning or heating and ventilating system.

The *first cost* of air conditioning or heating and ventilating systems includes the following:

1. Heat producing equipment including boilers, burners, etc.
2. Heat distributing equipment including direct radiation, piping, etc.
3. Air handling equipment including fans, air heaters, air conditioners, filters, controls, etc.
4. Air distributing equipment including ducts, outlets, grilles, etc.
5. Refrigerating equipment including piping, pumps, etc.
6. Water conservation devices including towers, evaporative condensers, etc.

The best procedure for establishing the first cost of any system is to select the various parts after thorough engineering study, and then to estimate the installed costs of same. When such detailed work is not war-

ranted or when only rough comparisons are desired between several types, approximate unit costs are of value as time savers.

Approximate installed prices of the various parts are shown in Tables 1, 2, and 3 and, as indicated, vary with the size or capacity of the equipment. The apparatus and items included in each group are stated in the footnotes beneath the tables. By use of these three tables it is possible to obtain a reasonable approximation of the cost of heating or air conditioning a given space or building. In order to use the approximate values indicated in Tables 1, 2 and 3, a rough estimate of the load is required. If time does not permit a determination of the load and if extremely rough figures will suffice, Table 4 may be used as an indication of the price of various types of air conditioning applications.

Other first costs, incurred because of the installation of the air conditioning or heating and ventilating system, include costs of electrical work, plumbing, miscellaneous piping, building alterations, cutting, patching, remodeling, or redecorating after installation, consulting engineer's fees, licenses, permits, etc. These vary so widely that no approximations are possible and each case must be considered alone.

The length of the amortization period to be used depends upon: the type and remaining life of the building or space for which the system is to be used; the type of equipment to be employed as a part of the system; the character of the business; and the lease or ownership conditions. For small shops in rented quarters on short term leases, a period of 5 years or

Table 1. Typical Installed Costs of Heat Producing and Heat Distributing Systems

Btu per Hour (Millions)	Sq Ft EDR (Steam) Thousands	Boiler Horse Power	Cost in Dollars—per Million Btu per Hour[a,b]			
			1	2	3	4
			Boilers Water Tube	Boilers Fire-Tube	Hot Water System Forced Circulation	Direct Radiation System Steam
0.8	3.3	24		2100	8500	7850
1.2	5.0	36	2300	1750	8200	7600
1.6	6.6	48	2000	1570	8000	7350
2.0	8.3	60	1850	1460	7800	7200
3.0	12.5	90	1600	1230	7600	7000
4.0	16.6	119	1400	1100	7500	7000
5.0	20.8	149	1250	980	7500	7000
6.0	25.0	179	1160	920	7500	7000
7.0	29.2	209	1100	880	7500	7000
8.0	33.3	239	1050	850	7500	7000
10.0	41.6	299	1000		7500	7000
12.0	50.0	358	970		7500	7000
14.0	58.3	418	930		7500	7000
16.0	66.6	478	900		7500	7000
18.0	75.0	538	860		7500	7000

[a] Columns 1 and 2 include hand fired boiler erected, with shaking grates and standard trimmings, Column 1 includes brickwork and rotating soot blowers. Foundations and piping are not included.
Columns 3 and 4 include direct radiation, piping, valves, specialties, and insulation. Boilers, condensate and circulating pumps, boiler connections, and all building construction or alterations are not included.

[b] The approximate figures given above may vary as much as 30 per cent due to job conditions, labor rates, and locality. These figures represent the selling prices of the individual items indicated above based on costs encountered in the year 1940. It is suggested that correction be made according to the particular locality for increases in labor and material costs since that year.

TABLE 2. AVERAGE INSTALLED COSTS OF FORCED AIR HEATING AND AIR CONDITIONING APPARATUS

CFM Supply Air (Thousands)	Cost per CFM—Dollars[d]			
	1	2	3	4
	Air Conditioning Equipment[a]	Heating and Ventilating Equipment[b]	Ducts Office Bldgs.[c]	Ducts Specialty Stores[c]
5	0.300	0.210		0.250
10	0.270	0.180		0.249
15	0.250	0.155		0.247
20	0.235	0.140		0.245
30	0.220	0.120		0.240
40	0.210	0.110	0.340	0.235
60	0.210	0.110	0.330	0.230
80	0.210	0.110	0.325	
100	0.210	0.110	0.320	
120	0.210	0.110	0.310	
140	0.210	0.110	0.305	
160	0.210	0.110	0.295	
180	0.210	0.110	0.290	
200	0.210	0.110	0.280	

[a] *Air conditioning equipment* includes fans and drives, filters, heaters, spray dehumidifiers or cooling coils, automatic controls, apparatus casings and insulation, and recirculating pumps.
[b] *Heating and ventilating equipment* includes fans and drives, filters, heaters, automatic controls and apparatus casings
[c] Includes ducts, grilles, outlets. insulation where required and specialties.
[d] The approximate figure given above may vary as much as 35 per cent due to job conditions, labor rates and locality. Figures are based on conventional systems. Building alterations, piping, plumbing and wiring are not included. These figures represent the selling prices of the individual items indicated above based on costs encountered in the year 1940. It is suggested that correction be made according to the particular locality for increases in labor and material costs since that year.

TABLE 3. AVERAGE INSTALLED COSTS OF REFRIGERATION SYSTEMS AND WATER SAVING DEVICES

Tons Refrigeration	Cost—Dollars per Ton[e]					
	1	2	3	4	5	6
	Reciprocating (Water Cooling)[a]	Centrifugal (Water Cooling)[a]	Reciprocating Direct Expansion[b]	Evaporative Condenser[c]	Cooling Tower Steel[d]	Cooling Tower Wood[d]
25	147		106	47		
50	132		86	40		
75	125		77	35		
100	120		72	32	47	40
150	115	137	68	30	45	38
200	113	124			44	36
250	110	115			42	35
300		107			40	33
400		97			38	31
500		93			36	29

[a] Columns 1 and 2 include compressors, evaporators, and water-cooled condensers; auxiliaries; electric motor, starter and drive; refrigeration piping, refrigerant and insulation of cooler and suction lines. Turbine driven centrifugal equipment may cost about 4 to 6 per cent more than motor driven, if condensing turbine and steam condenser, with supplementary equipment, are used.
[b] Column 3 includes same items as Column 2 except evaporator and auxiliaries are omitted.
[c] Column 4—These values are additive to Columns 1 and 3.
[d] Columns 5 and 6 include towers erected and a reasonable allowance for condenser water piping and pumps.
[e] The approximate figures given above may vary as much as 25 per cent due to job conditions, labor rates and locality. Building alterations, supply water and plumbing connections, wiring, chilled water piping and pumps are not included in these figures and may vary widely. Chilled water piping and pump costs may vary between $10 and $50 per ton of refrigeration effect. These figures represent the selling prices of the individual items indicated above based on costs encountered in the year 1940. It is suggested that correction be made according to the particular locality for increases in labor and material costs since that year.

TABLE 4. APPROXIMATE COSTS OF ALL-YEAR AIR CONDITIONING[a,b]

Application	Selling Price per Unit Indicated											
	Sq Ft Net Area			Person			Ton Refrigeration			CFM Supply Air		
	Low	Avg	High	Low	Avg	High	Low	Avg	High	Low	Avg	High
Banks	$1.25	$2.00	$3.50	$80	$140	$320	$350	$480	$700	$0.95	$1.40	$2.75
Department Stores: Basement	0.60	1.00	1.15	25	40	100	250	325	400	0.60	0.80	1.00
Department Stores: Main Floor	0.60	1.05	1.25	25	40	80	250	325	400	0.60	0.75	0.90
Department Stores: Upper Floors	0.60	0.90	1.10	30	45	100	300	350	400	0.60	0.80	1.00
Hotels—Guest Rooms				$240[c]	$300[c]	$420[c]						
Office Buildings	$0.85	$1.25	$2.00	110	135	180	$300	$375	$500	$0.75	$1.05	$1.50
Offices—Small Suites	1.00	1.75	3.25	110	180	350	300	450	800	0.75	1.25	2.50
Restaurants	1.40	2.20	3.50	15	18	25	215	280	400			
Specialty Shops: Barber Shops and Beauty Parlors	$1.50	$2.50	$4.00	$50	$100	$150	$325	$400	$500	$0.70	$1.00	$1.50
Specialty Shops: Dress Shops	1.25	1.50	2.00	45	55	75	230	275	350			
Specialty Shops: Drug Stores	1.25	1.50	2.00	45	55	75	240	275	325			
Specialty Shops: Five-and-Ten	0.90	1.20	2.80	22	33	55	220	270	425	0.80	0.95	1.50
Specialty Shops: Shoe Stores	0.95	1.20	1.75	35	40	50	230	275	350			
Theaters				$9[d]	$15[d]	$23[d]	$215	$300	$450	$0.65	$1.00	$1.25

[a]Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne and W. A. Grant (*Pitman Publishing Corp.* 1940, p. 64).

[b]There is a wide variation in unit prices such as shown above, due to differences in load concentration and in complexity of physical layout. Prices cover air-conditioning work only, for year-round use (except the *low* column, which in general includes summer cooling only). Net area (square feet) includes corridors, but excludes elevators and service spaces. *Excluded* are the cost of plumbing and steam connections, electrical wiring, painting and redecorating, building construction and alterations, and boilers or burners. Low figures in general are simply direct expansion systems using city water for condensing. High figures are usually chilled water systems with cooling towers or evaporative condensers, and may be considered high quality systems as well.

[c]Price per room.

[d]Price per seat.

less may be proper, whereas for larger installations in buildings that are owned outright a period of 10 or 20 years or more may be used.

Depreciation due to deterioration and obsolescence must also be considered in arriving at the amortization period. Deterioration and maintenance generally go hand in hand. If a long depreciation period is to be

TABLE 5. APPROXIMATE LIFE OF EQUIPMENT
(Including Obsolescence and Deterioration)[a]

	LIFE IN YEARS
1. HEAT PRODUCING EQUIPMENT	
(a) Boilers	15
(b) Stokers and burners	10
2. HEAT DISTRIBUTING EQUIPMENT	
(a) Piping—copper	20
(b) Piping—iron	10
(c) Radiation—concealed	12
(d) Radiation—direct	10
(e) Valves and specialties	5
3. AIR HANDLING EQUIPMENT	
(a) Filters—automatic	8
(b) Heating and cooling coils	10
(c) Spray humidifiers and dehumidifiers	10
(d) Fans	10
(e) Air conditioning units	10
(f) Motors	15
(g) Electrical starting equipment	8
(h) Pneumatic control systems	10
(i) Electric control systems	8
4. AIR DISTRIBUTING EQUIPMENT	
(a) Ductwork	20
(b) Outlets, grilles	20
(c) Duct insulation	10
(d) Painting	
5. REFRIGERATING EQUIPMENT	
(a) Centrifugal refrigerating machines	15
(b) Reciprocating refrigerating machines	10
(c) Motors and starters (see Item 3 above)	
(d) Piping—copper	20
(e) Piping—steel	15
(f) Pumps	15
6. WATER SAVING DEVICES	
(a) Evaporative condensers	10
(b) Cooling towers	10
(c) Wells	Varies Widely

[a] Modern Air Conditioning, Heating and Ventilating, W. H. Carrier, R. E. Cherne and W. A. Grant (*Pitman Publishing Corp.* 1940, p. 66).

used, then the item for maintenance, repair and replacement of wearing parts must be greater than for a short depreciation period.

Obsolescence depends mainly on time required for the equipment to become out-moded. Air conditioning, particularly, would probably suffer more from obsolescence in small plants than in large establishments. In addition the obsolescence of the building or property in which the equipment is installed may have a similar effect upon the equipment.

An approximation of the useful life of various items of equipment and parts of systems is shown in Table 5. It should be noted that if an appropriate maintenance item is not established, the rate of equipment deterioration may be substantially increased.

Interest

The interest chargeable may not represent the existing money rates. It may include an item to cover the diversion of capital or other items

Table 6. Owning and Operating Cost

First Cost		Annual Service Cost	
Cost of mechanical system		Electric Power Costs	
Other costs		Fans	
First Cost (FC)—Total		Pumps—Chilled water	
Annual Fixed Charges		Pumps—Condenser water	
Amortization—Depreciation period Y years		Pumps—Well water	
Interest rate $I\%$		Cooling tower fans	
Amortization and Depreciation $\frac{FC}{Y} =$		Cooling tower pumps	
Interest: $\frac{Y+1}{2Y} \times I =$		Refrigeration machines	
Taxes		Miscellaneous or other	
Insurance		Gas	
Rent		Coal	
Annual Fixed Charges: (Total)		Oil—for boilers or Diesel engines	
Annual Maintenance Costs		Steam	
Lubricating oil and grease		For direct heating	
Painting for corrosion protection or other purposes		For Ventilation—preheaters	
Replacement of worn parts		For Ventilation—reheaters	
Refrigerant		For Turbine driven equipment	
Wages of engineer or operator		For Engine driven equipment	
Annual Maintenance Cost—TOTAL		Sewers	
		Charges for discharging well water into public drainage systems	
		Annual Service Costs—TOTAL	
		Summary	
		Annual Fixed Charges	
		Annual Service Costs	
		Annual Maintenance Costs	
		Annual Owning and Operating Costs—TOTAL	

depending on existing tax laws which may make it necessary to charge interest due to diversion of capital as a cost item. It should be noted that interest may be based on an unamortized balance. As an example, a 15 year amortization period with a 4 per cent interest rate will approximate a 2.1 per cent average annual interest rate.

Taxes

The taxes chargeable will be the proportion of property tax caused by the increased valuation of the property due to air conditioning.

Insurance

The rate for insurance may vary considerably depending on the type of structure in which the equipment is located and upon other governing factors. A rate of about $0.60 per $1,000 may be considered as being representative for normal installations.

Rent

If the equipment under consideration is to be located in rented or leased quarters it may be necessary to include an item for space rental.

An orderly arrangement of the various components of owning and operating costs, which will also serve as a check list to forestall inadvertent omissions, is illustrated in Table 6. The formulas for computing amortization and interest are given in the table. Interest should be computed on the undepreciated portion of investment only.

MAINTENANCE COSTS

Maintenance costs include replacement parts and the labor required for making repairs, replacing parts, cleaning, painting, etc. It should be noted

TABLE 7. APPROXIMATE MAINTENANCE COST FOR LARGE AIR CONDITIONING INSTALLATIONS, USING HIGH QUALITY EQUIPMENT

	DOLLARS PER TON PER YEAR
Repairs for refrigeration machinery	0.60
Refrigerant	0.25
Oil and grease	0.07
Painting (Water boxes and dehumidifiers)	0.25
Filters, clean and reoil 4 times per year	0.85
Controls, outside service	0.15
Cleaning air conditioners	1.15

that major overhauling or complete replacement may restore the capital value of certain items of equipment, and in such cases the costs incurred may not necessarily be charged as maintenance costs. Generally, routine labor requirements will be the function of an operating engineer or staff and the responsibility of this group may extend beyond the equipment being discussed here; hence, it is important to include only an equitable share of the time of this group. Extraordinary repairs involving special machinery will usually be covered by contract with equipment service divisions and should be accounted for on that basis.

Many of the items included in maintenance costs are highly variable and depend on the type and quality of the purchased equipment. For large air conditioning installations, using high quality equipment, some approximate costs per ton are given in Table 7.

In addition to the costs listed in Table 7, consideration should possibly be given to other items such as: water treatment for boilers; other boiler and heating plant cleaning and repairs; repair and replacement of heating plant valves, traps, and vents; water treatment for cooling tower or chilled spray water; drive belts, possible damage due to freezing weather; cleaning of air ducts; and repairs to insulation.

Table 8. Equivalent Full Load Operating Hours of Refrigeration Equipment Used for Summer Cooling May 15 to Oct. 15[a]

Application	Hr Open for Business	Atlanta	Boston	Chicago	Detroit	Los Angeles	New Orleans	New York	Phila-delphia	Okla-homa City	St. Louis	Wash-ington D.C.
Barber shops	1280	1010	650	720	720	680	1080	830	860	1020	890	940
Department Stores	940	840	560	610	610	580	890	700	720	840	750	780
Drug Stores	2100	1630	950	1060	1060	980	1790	1280	1330	1650	1420	1530
Funeral Parlors	600	440	300	330	330	310	470	370	380	440	400	410
Offices	940	870	560	620	620	580	900	710	740	880	770	810
Restaurant (Short Hour)	1290	970	535	620	620	570	1060	760	800	980	830	880
Restaurant (Long Hour)	2100	1510	820	930	930	850	1690	1170	1210	1530	1300	1400
Specialty Shops (5 & 10)	1090	800	530	590	590	560	860	670	690	810	720	750
Theaters—Continuous	1500	1010	700	750	750	720	1080	850	870	1020	910	950
Theaters—Neighborhood	900	640	420	450	450	430	650	500	520	650	550	580

[a]Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne and W. A. Grant (*Pitman Publishing Corp.* 1940, p. 73).

SERVICE COSTS

Service costs include the costs for power, water, steam, coal, oil, etc., consumed to operate the system.

From the selected equipment and type of installation, it is possible to segregate the relatively constant power loads and the total brake horsepower. Annual power cost can then be figured from the following formula:

$$\text{annual power cost} = \frac{0.746\,(\text{bhp})\,H\,R}{\eta} \tag{1}$$

where

bhp = Brake horsepower.
H = Annual operating hours.
R = Power rate, dollars per/kwhr.
η = Motor efficiency (decimal).

In using Equation 1 it must be pointed out that the electric rate, R must reflect the proper combination of energy and demand rates. These vary widely between the utility companies, and sometimes the rate structure is such that it is largely the demand charge which determines the proper value of R to use in Equation 1.

Refrigerating Equipment Operating Cost

In an air conditioning system, the refrigerating equipment is usually the largest power-consuming item to be considered. Also, the prediction of operating cost is more complex, since the power required for summer cooling is affected by many factors which are of a variable nature. Among these are solar radiation, temperature difference between outside and inside, sensible and latent heat brought in with outside air, sensible and latent heat from people, heat released by electric lights, and, in some cases motor-driven equipment, cooking devices, and other equipment used in the conditioned spaces. Some of these factors vary with the weather while others are substantially independent of it. The relative proportion of each factor varies widely even among installations of the same application, depending on building layout and location, the size and quality of the establishment, the personal idiosyncracies of the owner, and other items.

In a strict sense, it is necessary to evaluate the effect of all such factors in order to predict the operating cost of a refrigeration plant. In most cases this procedure will prove to be too tedious, or it may not be possible because the exact breakdown of load data is unknown. The use of a simplified semi-rational formula, Equation 2, which takes into account for each application the number of hours open for business and the geographical location, will provide a value of H which may be used in Equation 3 to determine the *Season Power Cost.*

$$H_e = m\,(b + cf) \tag{2}$$

where

H_e = Equivalent full load operating hours of refrigeration equipment used for summer cooling during period May 15 to October 15.
m = Total hours during period May 15 to October 15 that the establishment is open for business.
b = Fraction of maximum load from internal heat under average operating conditions.

c = Fraction of maximum load which is due to external sources at maximum design conditions.

f = Ratio of the number of hours for a particular city, when the outside wet-bulb exceeds 65 F, during the period June 1 to October 1 to the total number of hours during that same period. Total hours are assumed as 8 hr per day period for barber shops, department stores, funeral parlors, offices, short hour restaurants, and specialty shops and 12 hr per day period for drug stores, long hour restaurants, and theaters.

Table 8 was calculated from Equation 2. It should be pointed out that certain southern cities may have seasons longer than the 5-month period indicated in Table 8. If it is desired to consider a longer season of operation, the ratio of full load operating hours to hours open for business is smaller; in other words, the *refrigeration load factor* is lower. This is true

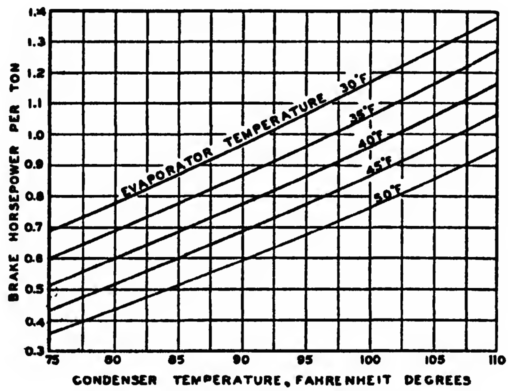

FIG. 1. TYPICAL BRAKE HORSE POWER REQUIREMENTS FOR REFRIGERATION[a]

[a] Values given are representative of "F-12" reciprocating machines of about 25 tons capacity in air conditioning applications. Requirements of smaller machines are usually higher, and for larger machines may be lower. Values shown are for liquid refrigerant at condenser temperature (no subcooling). Subcooling of the liquid may decrease these values approximately 0.3 per cent to 0.5 per cent for each Fahrenheit degree the liquid temperature is lowered.

because the extra increment of days added will be at relatively light load, since the table already includes the more severe part of the season.

The season electrical power cost for refrigerating equipment is then given by the following equation:

$$\text{Season power cost} = \frac{0.746\,(\text{bhp}_t)\,T\,H_e\,R}{\eta} \tag{3}$$

where

bhp_t = Brake horsepower per ton (see Fig. 1) for *average* load during period. (Due allowance should be made for poorer compressor efficiency at light load.)

T = Tons of refrigeration at maximum design load.

H_e = Equivalent full load refrigeration operating hours (Table 8).

R = Power cost, dollars per kwhr, including demand and energy charges.

Motor efficiency at average load (decimal).

Table 9 will be useful in estimating the design load in the absence of detailed load estimates.

In considering refrigeration power consumption, it should be noted that the use of weather records for a specific year may lead to large inaccuracies in estimating operating costs, since there may be wide variations from year to year, and therefore, average yearly weather records should be used rather than those for any individual year.

If the refrigeration compressor is steam turbine driven, the same general method can be followed, taking into account average water rate per brake horse power-hour and the cost of steam.

Condenser Water

Condenser water cost estimates can also be based on equivalent full load operating hours of the refrigeration equipment. The varying temperature of the water at its source, as well as the temperature of the discarded water, must however be taken into account. In general, when water is purchased, control is provided to hold the leaving water temperature (or condensing temperature) constant; and in such case the entering water temperature becomes the major variable and the gallons per minute per ton can readily be calculated for any water temperature rise.

TABLE 9. REPRESENTATIVE TONS PER 100 SQ FT FOR VARIOUS APPLICATIONS

	Low	Avg	High
Department Store (Main Floor)	0.50	0.58	0.67
Department Store (Upper Floors)	0.29	0.46	0.54
Dress Shop	0.29	0.50	0.71
Drug Store	0.33	0.54	0.83
Lunch Room	0.83	1.08	1.33
Office Bldg	0.21	0.25	0.38
Restaurant	0.58	0.75	1.00
Shoe Shop	0.29	0.42	0.63
Theater (Tons per Seat)	0.064	0.078	0.093

The following equation for cost of condenser water is useful:

$$B = 0.060\, a\, T\, H_o\, C \tag{4}$$

where

- B = Cost of water for refrigeration during period, dollars.
- a = Average gallons per minute (ton).
- T = Tons of refrigeration at maximum design load.
- H_o = equivalent full load refrigeration operating hours (Table 8).
- C = Water cost, dollars per 1000 gal.

The average gallons per minute per ton must take into account the variable water temperature. When well water is used as a source, and entering and leaving temperatures are considered constant, the average gallons per minute per ton obviously is equal to the design gallons per minute per ton. However, when the source is river or lake water, its maximum seasonal temperature will generally be reached at the same time that the refrigeration load factor is highest. The average gallons per minute per ton should be calculated from known or estimated water temperatures because they vary through the season. Maximum water main temperatures

are given in Chapter 37 but should always be verified locally. In lieu of this tedious work, the average gallons per minute per ton may be taken as 80 per cent of design gallons per minute per ton with reasonable accuracy, for the condition of variable temperature of entering water obtained from rivers and lakes.

When cooling towers or evaporative condensers are used, the windage and evaporation losses are usually between 3 per cent and 5 per cent of the water circulated.

Heating

The method of estimating fuel consumption to balance the building heat loss is given in Chapter 20. It is important to include the fuel required to heat ventilation air as used in ventilating and air conditioning systems. In estimating fuel consumption for ventilation air, the tendency of the conventional control systems to use less than the estimated quantity of outside air in cold weather should be considered in its effect in lowering fuel consumption. In addition, the heat required to accomplish winter humidifying must not be neglected, when this feature is included in the equipment.

BIBLIOGRAPHY

Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne, and W. A. Grant—Chapter IV (*Pitman Publishing Corp.*, 1940).

Predicting Operating Hours of Refrigeration Equipment Used in Air Conditioning, by W. A. Grant (*Refrigerating Engineering*, July, 1941).

Cost of Operation of Refrigeration Used for Air Conditioning, by R. E. Cherne (*Refrigerating Engineering*, December, 1943).

CHAPTER 45

INDUSTRIAL AIR CONDITIONING

General Requirements; Typical Applications in Storage and Processing: Air Quality Control, Control of Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical and Biochemical Reactions, Control of Rate of Crystallization, Elimination of Static Electricity; Calculations

INDUSTRIAL air conditioning is as much concerned with the atmospheric conditions required for maintaining the health, safety and efficiency of workers as with those required for the manufacture, processing, and preservation of material, equipment and commodities.

GENERAL REQUIREMENTS

Specialists in the field of industrial hygiene should be consulted in case of doubt concerning the presence of airborne industrial hazards to health. Chapter 10, Air Contaminants, Chapter 12, Physiological Principles, and Chapter 13, Air Conditioning in the Prevention and Treatment of Disease, will be of help in determining the atmospheric conditions which should be maintained around the worker. Comfortable conditions are desirable because they are likely to increase the efficiency of workers.

Table 1 lists the temperatures and relative humidities required for storage of certain commodities and for manufacturing and processing of others. The desirable relative humidity may range from a low of 5 per cent in certain industrial applications, such as insulation winding processes, up to a condition approaching saturation, as in processes relating to textile, tobacco and baking industries. Relative humidities of 50 per cent or less are on the dry side and are conducive to low regains in hygroscopic materials, drying out, increased brittleness of fibrous materials, increased static electricity, and tendencies toward increased dust liberation from the product. Relative humidities higher than 50 per cent are considered to be on the damp side. These conditions are conducive to high regain, promote softness and pliability in materials, decrease static electricity and tend to reduce the generation of product dust.

The most favorable temperature will vary according to the specific material and particular process. In some cases the temperatures listed in Table 1 have no direct influence upon the product itself but do affect the efficiency of employees, which in turn affect workmanship, uniformity and the cost of production. Frequently a compromise between the known optimum condition for processing and that required for worker comfort is unavoidable.

In many processes, the optimum air conditions are variable according to the stage and progress of the processing cycle from the raw material to the finished product. Some materials, such as cotton textiles, begin with a low relative humidity in the carding and picking rooms, and after passing through the various intermediate steps with a gradual increase of relative humidity, are subjected to relative humidities of from 75 to 85 per cent in the final stage of weaving. Other processes are encountered that require the reverse of this procedure, starting with a high relative humidity and finishing with a low relative humidity, as in the manufacture of glue and gelatinous materials, and making various types of gelatine capsules.

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING

INDUSTRY	PROCESS	TEMPERATURE FAHRENHEIT DEGREES	RELATIVE HUMIDITY PER CENT
AUTOMOBILE	Assembly line	65 to 80	40 to 55
	Precision parts—honing—machining	75 to 80	40 to 55
BAKING	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Dough retarding	32 to 40	76 to 85
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	90 to 95	80 to 90
	Storage of flour	65 to 75	55 to 65
	Storage of yeast	32 to 45	60 to 75
BIOLOGICAL PRODUCTS	Vaccines	below 32	
	Antitoxins	38 to 42	
	Blood bank	38 to 42	60 to 65
	Penicillin (dehydrated)	36 to 40	
BREWING	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
CERAMIC	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60 to 80	35 to 65
CHEMICAL	General storage	60 to 80	35 to 50
CONFECTIONERY	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
DISTILLERY	General manufacture	60 to 75	45 to 65
	Storage of grains	60	30 to 45
DRUG	Deliquescent powder	75	35
	Effervescent granulations	80	40
	Liver extracts (powdered)	70	20 to 30
	Storage of powders and tablets	70 to 80	30 to 35
	Tablet compressing	70 to 80	40
	Packaging	80	40
ELECTRICAL	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
FOOD	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats (frozen)	0 to 5	85
	Storage of meats (above freezing)	36	85
	Storage of sugar	80	35
FUR	Drying of furs	110	
	Storage of furs	28 to 40	50 to 65

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Concluded)

Industry	Process	Temperature Fahrenheit Degrees	Relative Humidity Per Cent
INCUBATORS	Chicken	99 to 102	55 to 75
INSTRUMENTS	Repair and calibration	68	50 to 55
LABORATORY	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
LEATHER	Drying of hides	90	
	Mulling	95 to 100	95
LIBRARY	Book storage	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCHES	Manufacturing	72 to 74	50
	Storage of matches	60	
MUNITIONS	Fuse loading	70	55
PAINT	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
PAPER	Binding, cutting, drying, folding, gluing	60 to 80	40 to 60
	Storage of paper	75 to 80	40 to 60
	Testing Laboratory	60 to 80	55 to 65
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	75 to 80	50 to 60
	Storage of rollers	70 to 90	50 to 55
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
	Cementing	80	25 to 30
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50 to 55
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	50 to 70
	weaving	68 to 75	85
	Rayon— spinning	70	85
	throwing	70	60
	weaving	75 to 88	60 to 75
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
	Testing Laboratory	70	65
TOBACCO	Cigar and cigarette making	70 to 75	55 to 75
	Softening	90	85
	Stemming or stripping	75 to 85	70

Safeguarding the health, safety and efficiency of workers requires control of dusts, fumes, smokes, mists, fogs, vapors, and gases, and control of the effective temperature, which includes temperature, humidity and motion of air about the worker.

General ventilation may be relied upon in some cases to control air contaminants. Chapter 9 gives information on natural ventilation. If mechanical ventilation is to be used, Chapter 32, Fans; Chapter 40, Air Distribution; Chapter 41, Air Duct Design; Chapter 33, Air Cleaning Devices; and Chapter 37, Spray Apparatus, furnish information on a broad range of industrial design conditions. If ventilation is to be provided for isolated areas using booths or enclosures, or if local exhaust ventilation is to be employed using hoods, exhaust slots or flexible tubes, fundamental design information will be found in Chapter 46, Industrial Exhaust Systems.

In controlling effective temperature, or in providing appropriate air conditions for storage and for processing, it may be necessary to humidify or dehumidify, to heat or cool, to clean and to transport the air. These subjects are treated in other chapters in the Guide. When exhaust ventilation is used, care should be taken to properly admit a supply of air into the building to replace that removed; otherwise the exhaust system may fail to perform its intended function. Chance infiltration should not be depended upon for this supply. Optimum results are most probable if the industrial process or project and its air conditioning system are designed together.

TYPICAL APPLICATIONS IN STORAGE AND PROCESSING

Air conditioning in storage and processing may have one or more of the following objectives: (1) air quality control, (2) control of moisture regain, (3) control of rate of chemical reactions, (4) control of rate of biochemical reactions, (5) control of rate of crystallization, and (6) elimination of static electricity.

Air Quality Control

The control of humidity has proved to be useful in blast furnace operations; better uniformity of product and increased rate of production are the result. Very low humidity, with or without control of air flow and temperature, is essential in preserving stored machinery and instruments against corrosion. Control of temperature is necessary in high precision manufacturing. If, in addition to temperature, humidity is controlled and air cleanliness maintained, good conditions for gaging laboratories, for optical goods production, and for such processes as photographic film and pharmaceutical manufacturing may be provided. In libraries and museums, control of humidity and temperature is important to the preservation of the materials stored there; air cleanliness is of almost equal importance in minimizing damage during use or when cleaning the materials. In drafting rooms, control of temperature and humidity to eliminate sweating of workers reduces damage to drawings and tracings, improves accuracy of work that is to be used in direct transfer processes, and improves worker efficiency.

Moisture Content And Regain

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of pro-

duction and upon the weight, strength, appearance and general quality of the product. The moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, comes to equilibrium with the moisture of the surrounding air. This is the fundamental basis for the control of certain physical qualities of the material during manufacture. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are fixed in the trade.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final specified moisture content suitable for the goods as shipped. *Moisture content* refers to free moisture (as in a sponge) and to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a bone-dry weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture.

A basis for calculating the regain of textiles is obtained by drying, under standard conditions, a sample from the lot; and the dry weight thus obtained is used in the calculations to determine the regain.

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary. The rate of absorption or drying varies with the nature of the material, its thickness and density.

During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded. These preliminary processes are carried out best in a relative humidity of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Rayons, like cotton, require different humidities and temperatures at different steps in manufacture, but on account of great loss of strength with the higher regains, should be finished in a relative humidity of 55 to 70 per cent.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent									Authority
			10	20	30	40	50	60	70	80	90	
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fiber	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tires	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Food-stuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulfuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

When hygroscopic materials absorb moisture from the surrounding air they deliver to the air sensible heat equivalent to the latent heat released by the moisture to the material. This may account for a small part of the total heat load of the conditioned space.

Conditioning and Drying

In general, the exposure of materials to desirable humidities for treatment may be coincidental with the manufacture or processing of the

materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material. When the final moisture content is lower than the initial one, the term *drying* is applied (See Chapter 47). If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco, for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content. Frequently conditioning or drying is made a continuous process in which the material is conveyed through an elongated compartment by suitable means and subjected to controlled atmospheric conditions.

Control of Rate of Chemical Reactions

A typical example of control of the rate of chemical reactions occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that during this process close control of both temperature and relative humidity should be maintained. The temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and obtains a solution of known strength throughout the mercerizing period.

Another well-known example in this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding effect on the rate of oxidization at the surface and allow the internal gases to escape freely as the chemical oxidizers *cure* the varnish from within. This produces a surface free from bubbles and a film homogeneous throughout. Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

Control of Rate of Biochemical Reactions

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 65 per cent is maintained to hold the surface of the dough open to allow the carbon dioxide gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

The curing of fruits, such as bananas and lemons, also comes under this classification. Bananas require a cycle of temperatures and relative humidities for ripening. The starches in the pulp of the fruit must be changed and the skin cured and colored, after which the fruit is cooled to maintain as low a rate of metabolism as possible. Ideal storage conditions range between 56 and 60 F with about 75 per cent relative humidity and ventilation at the rate of three or four air changes per hour.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high rela-

tive humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, control of moisture regain, and of chemical and biochemical reactions, is involved and only through close atmospheric control can the best quality of leaf be developed.

Control of Rate of Crystallization

The rate of cooling of a saturated solution determines the size of the crystals formed. Both dry- and wet-bulb temperatures are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right dry- and wet-bulb temperatures. The wet-bulb depression determines the rate of moisture pick-up or drying by the air and may be termed the *drying head*. Of equal importance is the uniformity at which the wet-bulb is maintained. If this is allowed to vary, poor results will follow due to checking and cracking of the unfinished coating.

Elimination of Static Electricity

The presence of static electricity is very detrimental to the satisfactory and economical processing of many light materials, such as textile fibers, paper, etc. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is minimized by increasing the relative humidity if the material being processed is not damaged thereby.

In attempting to eliminate static electricity, it must be borne in mind that for successful elimination the air that actually comes in contact with the material in the machine must be at a relative humidity of 50 per cent or more. As some machines consume a great deal of power which is converted directly into heat, the temperature in the machine may be considerably higher than the temperature adjacent to the machine where the relative humidity is normally measured. In such cases, the relative humidity in the machine will be appreciably lower than that elsewhere in the room, and it may be necessary to maintain a room relative humidity of 65 per cent, or even more, before the desired results can be obtained inside the equipment.

CALCULATIONS

The methods for determining the heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 14 and 15. The several other chapters previously cited call attention to additional phases of calculation which may be needed. Because of the large number of motors and heat producing units usual in an industrial application, it is particularly important that operating allowances for the latent and sensible heat loads be definitely ascertained and used in the calculations to determine the total design load.

BIBLIOGRAPHY

HEALTH, SAFETY AND EFFICIENCY

Lectures, The Inservice Training Course in Environmental Controls for Industrial Processes, University of Michigan School of Public Health, 1946: Environmental Controls in Industrial Health, by R. R. Sayers; Principles of Industrial Process Ventilation, by W. N. Witheridge; The Selection and Maintenance of Equipment for Process Control, by R. P. Warren; Environmental Control of Industrial Processes as Affected by Foundry Lay-out, by John Linabury; Foundry Ventilation, by J. M. Kane; Control of Welding Hazards, by W. C. L. Hemeon; Environmental Control of the Metal Cleaning Processes, by F. A. Patty; Painting, by T. F. Mooney; Health Hazards in the Electroplating Industry, by William Blum; Industrial Housekeeping and Sanitation, by John Soet.

Air Sanitation and Industrial Ventilation, by W. N. Witheridge, Detroit, 1945.

An Investigation of the Bacterial Contamination of the Air of Textile Mills with Special Reference to the Influence of Artificial Humidification, by W. F. Wells and E. C. Riley (*The Journal of Industrial Hygiene and Toxicology*, Vol. 19, No. 10, December, 1937).

Industrial Cooling as a Production Aid, by J. Partington, Jr. (*Heating and Ventilating*, April, 1944, p. 47).

Industrial Exhaust Ventilation in Industrial Hygiene, by A. D. Brandt (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 331).

Control of Industrial Atmospheres, by W. N. Witheridge (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 227).

Mine Ventilation and its Relation to Health and Safety, by D. Harrington (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 243).

Ventilation Requirements for Industrial Solvents, by W. C. L. Hemeon (*Heating and Ventilating*, December, 1945, p. 95, January, p. 69, March, p. 82, and April, p. 79, 1946).

Guides for Industrial Ventilation, by Allen D. Brandt (*Heating and Ventilating*, March, 1946, p. 67).

Industrial Dust Explosions, by Hylton R. Brown (*Heating and Ventilating*, March, 1946, p. 73).

Supply Air in Plant Ventilation, by J. B. Skinner and William M. Pierce (*Heating and Ventilating*, May, 1946, p. 57).

What Air Conditions Make for Comfort in Industry?, by H. A. Mosher (*Heating, Piping and Air Conditioning*, April, 1946, p. 106).

Air Conditioning Requirements for Workers in Factories, by H. A. Mosher (*Heating, Piping and Air Conditioning*, August, 1946, p. 82).

Hot Weather Limits Hard Work. (Brief summary of investigation done for Army Quartermaster Corps, by Department of Physiology, Medical School, University of Indiana. *Heating and Ventilating*, August, 1947, p. 110).

Methods Used in Determining the Health Hazards Arising from the Inhalation of Various Chemicals, by Francis F. Heyroth (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1947, p. 109, and discussions, *Heating, Piping and Air Conditioning*, April, 1947, p. 113).

Air Recirculation from Exhaust Systems, by John M. Kane (*Heating and Ventilating*, April, 1947, p. 75).

Should Air Be Recirculated from Industrial Exhaust Systems? by Allen D. Brandt (*Heating, Piping and Air Conditioning*, August, 1947, p. 69).

Dehumidification Methods and Applications, by John Everetts, Jr. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1946, p. 121).

Rating Dynamic Dehumidification Equipment, by E. R. Queer and E. R. McLaughlin (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1947, p. 103).

PROCESSING

Refrigerating Data Book, Refrigeration Applications Volume, Second Edition, 1946, Section VIII, Industrial Air Conditioning, Chapters 67 to 76, and Section III, Refrigeration in Food Manufacture, Chapters 25 and 26, *American Society of Refrigerating Engineers.*

Air Conditioning for Sausage Manufacturing Plants, by M. G. Harbula (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 343).

Air Conditioning in the Bakery, by W. L. Fleisher (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 141).

Proper Air Conditions for the Manufacturing of Confections, by A. E. Stacey, Jr. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1937, p. 640).

Air Condition the Bakery Throughout, by W. W. Reece (*Heating, Piping and Air Conditioning*, August, 1936, p. 149).

Bureau of Standards Studies Determine Press Room Conditioning Requirements, by C. G. Weber (*Heating, Piping and Air Conditioning*, March, 1936, p. 137).

Silica Gel Air Conditioning System Serves Rotogravure Printing Plant, by H. E. Ryerson (*Heating, Piping and Air Conditioning*, August, 1937, p. 497).

Moisture Control by Liquid Absorption Offers a Useful Air Conditioning Tool, by F. M. Johnson (*Heating, Piping and Air Conditioning*, December, 1938, p. 782).

Humidity in the Pressroom (*Heating and Ventilating*, May, 1932, p. 35).

Dehumidifying with Gas in a Printing Plant (*Heating and Ventilating*, May, 1935, p. 31).

Air Conditioning in the Paper Industry (*Heating and Ventilating*, October, 1935, p. 23).

Air Conditioning the Newspaper Plant, by R. T. Williams (*Heating and Ventilating*, September, 1937, p. 63).

Air Conditioning Requirements of Multicolor Offset Printing, by C. G. Weber (*Refrigerating Engineering*, December, 1936, p. 6).

Reactions of Lithographic Papers to Variations in Humidity and Temperature, by C. G. Weber and L. W. Snyder (*U.S. Bureau of Standards Journal of Research*, January, 1934).

The Treatment of Offset Papers for Optimum Register, by C. G. Weber and M. N. V. Geib (*U. S. Bureau of Standards Journal of Research*, February, 1936).

Effect of Air Conditioning Upon Munitions, by J. I. Lyle (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 383).

Air Conditioning of Blackout Plants, by W. A. Grant (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 459).

The Engineering Control of Some Solvent Hazards in War Industries, by S. C. Rothman (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 319).

Air Conditioning as Applied in Theatres and Film Laboratories, by D. C. Lindsay (*Transactions Society of Motion Picture Engineers*, April, 1927, Vol. XI, No. 30 p. 335–365).

Air Conditioning for Textile Plants Making and Using Synthetic Yarns, by L. L. Lewis (*Rayon Textile Monthly*, July, August, September, 1930).

Air Conditioning for Clothing Research Laboratory (*Heating and Ventilating*, July, 1943, p. 69).

Refrigeration Insures Quality of Clothing for the Army, by A. J. Mallinckrodt (*Heating and Ventilating*, March, 1944, p. 79).

Refrigeration for Textile Mill Air Conditioning, by P. L. Davidson (*Heating and Ventilating*, May, 1947, p. 57).

Relation of Air Conditions to Tobacco Curing, by J. Johnson and W. B. Ogden (*Wisconsin Agricultural Research Bureau* 110: 1–48, 1931).

Air Processing Needs of Plywood Plane Parts, by F. O. Jordan (*Heating and Ventilating*, April, 1944, p. 67).

Heating, Air Conditioning and Insulation for Penicillin Production, by J. C. Siegesmund (*Heating, Piping and Air Conditioning*, August, 1944, p. 475).

Controlled Air Supply for Supercharger, Carburetor and Engine Testing, by C. S. Leopold (*Refrigerating Engineering*, August, 1943, p. 85).

Close Machine Tolerances Possible Through Temperature Control, by R. P. Dewey (*Heating and Ventilating*, April, 1944, p. 60).

Operating and Maintenance Charges for Air Conditioning Machine Shops, by H. B. Lammers (*Heating, Piping and Air Conditioning*, February, 1946, p. 93).

Ventilation of Long Tunnels During Construction, by W. B. Harris (*Heating and Ventilating*, July, p. 63 and August, p. 95, 1947).

Foundry Ventilation, by Jim Black and Lester Avery (*Heating, Piping and Air Conditioning*, March, p. 71, April, p. 78, and May, p. 89, 1947).

Smoke and Dust Control, Heating and Ventilating in the Sorel Plant (*Industrial Heating*, February, p. 236. and March, p. 422, 1947).

CHAPTER 46

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, Hood Design Principles, Capture Velocity and Hood Suction, Duct System Design, Resistance of System, Efficiency of System, Air Flow Producing Equipment, Protection Against Corrosion and Abrasion

IN many industrial plants some type of exhaust system designed to collect and remove dusts, fumes, mists, vapors and gases is essential in order to promote efficiency, economy, and safety of operation. Definitions of these various contaminants are included in Chapter 10, Air Contaminants.

The theory of air flow in an exhaust system in the following paragraphs *A* to *D* will be found in a publication[1] of the *American Foundrymen's Association*.

A. When the air flow producing equipment of an exhaust system is properly operated it will produce a negative pressure (below atmospheric) in the exhaust side of the system sufficient to overcome all resistance and to sustain the desired air velocity; and, further, will overcome all resistances on the discharge or positive pressure side of the system so that the air drawn through exhaust inlets will be discharged against atmospheric pressure.

B. An exhaust system is entirely dependent on a *sufficient volume* of air flowing into the exhaust inlets to catch the matter to be exhausted before such matter has an opportunity to diffuse into the general atmosphere of the work place or room.

C. The velocity of the air flowing into an exhaust inlet is usually of secondary importance and becomes an essential factor only when a certain velocity is required to overcome some force acting on the matter to be caught. The velocity within an exhaust system is only important to the extent that it shall be sufficient to convey the entrained matter and prevent it from settling or dropping out in the piping system. Velocity in terms of velocity pressure is most essential in designing a system because it is the basis upon which all calculations are made. In testing and checking a system the velocity as determined from the velocity pressure reading obtained by means of a Pitot tube is the only true indication of the exact air flow in a pipe or system.

D. The total pressure within an exhaust system is only of importance in determining the power required to operate the system. Total pressure tests do not indicate the proper functioning of an exhaust system as related to the volume and velocity of the air flowing into an exhaust inlet.

General design information is included in this chapter which is intended to relate primarily to industrial exhaust systems.

CLASSIFICATION OF SYSTEMS

In general there are two basic layouts of exhaust systems, the central and the multiple unit system. In the central system a fan is located near the center of operations with a piping system radiating to the various machines to be served. In the multiple unit system, which is sometimes employed where the machines to be served are widely scattered, or where the operations are apt to be independent or intermittent, small individual exhaust fans are located at the center of the machine groups or at each machine. The unit arrangement has the advantage of flexibility.

Exhaust systems may be classified with respect to the nature of the material to be handled by them as (a) those handling dusts and certain fumes and mists of large particle size, and (b) those handling vapors, gases and certain fumes and mists of small particle size. Design details differ in systems serving dust producing operations and those exhausting the

more vapor-like matter even though the same basic theories govern both classes.

Dust or gas may be captured by enclosure or by open hoods with positive inward air movement. With some classes of machinery it is not feasible to hood the machines closely and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. Examples of these classes include such machines or operations as pickling tanks, melting furnaces, arc welding and monument finishing operations.

Open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. In no instance should the operator be located between the source of dispersion and the exhaust hood or enclosure. When the hood must be placed at some distance above the machine it should be large enough to cover a large area as dispersion (considering dust) is usually quite rapid.

Some consideration should be given to the natural movement of the contaminant. In many cases there are convection currents and other atmospheric disturbances in the work room. These disturbances diminish the tendency of dust and fumes to settle from the room air.

In some classes of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, and the removal of some dust from the machine or enclosure may be merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air flow through all cracks and openings, thus preventing escape of the dust. While the exhaust system is required to handle only the air which enters through the crevices and openings in the enclosure, in many installations leakages are very high and great care is required to reduce such leakages to a minimum.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space.

HOOD DESIGN PRINCIPLES[2, 3, 4, 5, 6, 7]

The first and most important steps in the design of a local exhaust system are to determine the number and shape of hoods or enclosures and the size of the branch connections. No general rules, however, can be given since hood and duct designs are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available.

In general, the most important requirements[8] of an efficient local exhaust system are:

1. Hoods, ducts, fans, motors and collectors should be of adequate size and type.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should be placed so as not to interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When flammable contaminants are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall (Refer to Pamphlet No. 91, *National Board of Fire Underwriters*[9]).

6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity. Motor and starting equipment should conform to Article 500—Hazardous Locations—of the National Electrical Code[10].

7. The exhaust system should be readily accessible for inspection and maintenance.

CAPTURE VELOCITY AND HOOD SUCTION

The removal of dust or contaminant by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it into the exhaust system. The air velocity necessary to accomplish this depends upon the physical properties of the material to be controlled and the direction and speed with which it is dispersed. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood must be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible. This will reduce the size of equipment, the power required by the system, and also the heating load requirements in winter.

TABLE 1. MINIMUM AIR VELOCITIES REQUIRED AT POINT OF ORIGIN TO CAPTURE CONTAMINANT EFFECTIVELY

CONDITION OF GENERATION OF CONTAMINANT	MINIMUM CAPTURE VELOCITY, FPM	PROCESS
Released without noticeable movement	50–100	Evaporation of vapors, exhaust from pickling, washing, degreasing, plating, welding, etc.
Released with low velocity	100–200	Paint spraying in booth; inspection, sorting, weighing, packaging, low speed conveyor transfer points, rotating mixtures, barrel filling.
Active generation	200–500	Foundry shakeout, high speed conveyor transfer points, crushers, screens.
Released with great force	500–2000	Grinding, tumbling mills, abrasive cleaning.

Capture Velocities

Data for the selection of *capture velocities* of many operations are not available, but it is safe to assume that for most dusty operations velocities should not be less than 200 fpm at the point of origin. Recommended minimum capture velocities for various processes are given in Table 1.

The method for determining approximately the quantity of air that must be exhausted to produce these capture velocities at the point of origin is given in Equation 1:

$$Q = V(10X^2 + A) \tag{1}$$

where

Q = Quantity of air exhausted, cubic feet per minute.

V = Air velocity in feet per minute at X distance in feet from the hood and on the centerline of the hood.

X = Distance in feet along the hood centerline from the face of the hood to the point where the air velocity is V feet per minute.

A = Area in square feet of the hood opening.

Table 2. Branch Pipe Size for Woodworking Machine Hoods

Based on a Pipe Velocity of 4000 fpm.

Type of Machine	Size, In.		No. of Branches	Minimum Diameter, In.		
	Min.	Max.		Bottom Branch	Top Branch	Others
Self feed table saw			2	5	4	
Other single saws		18	1		4	
	18		1		5	
Saws with Dado Head			1		5	
Band saws		2	2	4	4	
	2	3	2	5	4	
	3	6	2	5	5	
Disc sanders		18	1	4		
	18	28	1	5		
	26	32	2	4	4	
	32	38	2	5	4	
	38	48	3	5	4	4
Triple drum sanders		30	1	7		
	30	36	1	8		
	36	42	1	9		
	42	48	1	10		
Single drum sanders: (area in sq in.)		350	1	4[a]		
	350	700		5		
	700	1400		6		
	1400	2800		7		
Horizontal belt sanders		9	2	5	4	
	9	14	2	6	4	
Vertical belt sanders		6	1	4		
	6	9	1	5		
	9	14	1	6		
Jointers		8	1	4		
	8	20	1	5		
Single planers		20	1	5		
	20	26	1	6		
	26	36	1	7		
Tenoner			2	5	5	

[a]Not over 10 in. diameter.

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dust and articles dropped by the operator. Such provisions are of

assistance in keeping the ducts clear. The air quantity required to capture dust which is thrown or projected in a direction away from the hood at considerable velocity, may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Air Flow from Static Readings

State Codes for local exhaust systems at certain operations list minimum static suction requirements which may range from $1\frac{1}{2}$ in. to 5 in. water column. Frequently, in grinding, buffing and polishing operations a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. Where the hood coefficient is known the volume of air flow through any hood may be determined from the equation:

$$Q = 4005\, f A \sqrt{h_s} \qquad (2)$$

where

Q = quantity of air exhausted, cubic feet per minute.
A = area of connecting duct, square feet.
h_s = static suction measured at approximately 3 diameters from throat of hood, inches of water.
f = orifice or restriction coefficient which varies from 0.6 to 0.95 depending on the shape of the hood.[5]

An average value of f is 0.8, although for a well-shaped opening a value of 0.85 to 0.9 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_s}} \qquad (3)$$

where h_v is the velocity head in the connecting duct.

The *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 4 which shows that the velocity at any point along the axis varies approximately inversely as the square of the distance. This formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted externally to the hood opening.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, Equation 4 may be used to determine the air velocity at any point along the axis[11]:

$$V = \frac{0.1\, Q}{x^2 + 0.1\, A} \qquad (4)$$

where

V = velocity at point, feet per minute.
Q = quantity of air exhausted, cubic feet per minute.
x = distance along axis, feet.
A = area of opening, square feet.

Design Based on Total Air Flow

Where the foregoing factors are not known, the usual method of designing an exhaust system is to base the air flow through the system on rates of flow (through each hood) which have been found by experience to provide adequate control. For woodworking systems the sizes of branch connections in common use are given in Table 2.

Similar data for grinding and buffing wheels are given in Table 3.

Velocity Contours

It is possible by use of a specially constructed Pitot tube[12] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be

Table 3. Branch Pipe Sizes for Grinding and Buffing Hoods
Based on a Pipe Velocity of 4500 fpm.

Type of Wheel	Wheel Size Diameter, In.		Maximum		Branch Pipe Minimum Diameter, In.
	Min.	Max.	Width In.	Area Sq In.	
Grinding	....	9	1	30	3
	9	18	3	175	4
	18	24	4	300	5
	24	30	5	500	6
	30	36	6	700	7
Disc Grinding	....	20	....	300	4
	20	30	....	....	5
Buffing, Polishing and Scratch Brushing	....	8	2	50	3½
	8	16	3	150	4
	16	24	4	300	5
	24	30	6	600	6

expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[13].

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio, provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

Low Velocity Systems

On multiple installations of the same operation it is often possible to institute a great saving in power cost by designing an exhaust system using low velocities in the main ducts. Such a system for use in grinding and shaping porcelain has been described[14]. In these operations, the separate machines are grouped around a central plenum chamber and exhausted by means of a low pressure fan connected to the plenum. In

one such case a power saving of over 90 per cent was obtained. A similar design technique[15] has been described for use in ventilating plating tanks.

Canopy Hoods

Canopy hoods are being replaced by other types of hoods, such as slotted hoods at tank operations. Where canopy hoods are used, they should

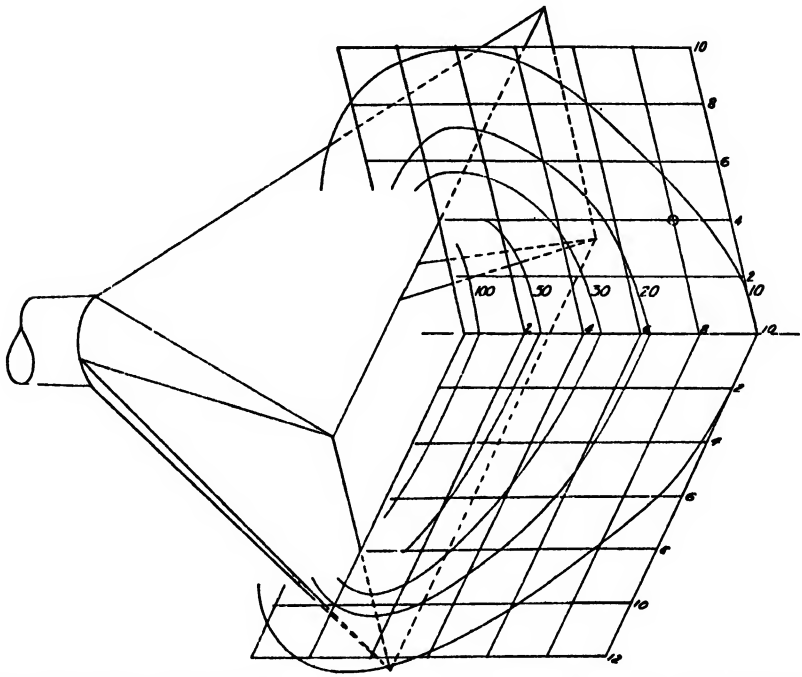

FIG. 1. VELOCITY CONTOURS FOR RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

extend 6 in. laterally from the tank for every 12 in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off. The recommended minimum capture velocity is 100 fpm.

The quantity of air which must be exhausted to obtain any given capture velocity is expressed by the following equation:

$$Q = 1.4\,PDV \tag{5}$$

where

Q = quantity of air exhausted by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = capture velocity, feet per minute.

Lateral Exhaust Systems

Lateral exhaust as developed for chromium plating[16] is preferred to canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts located at the top and extending fully along one or more sides of the tanks. The slot width is usually based on a slot velocity of 2000 fpm, but should not be less than 1 in. wide. The hood should not be required to draw the air laterally for a distance of more than 24 in. and the level of the solution should be kept 6 to 8 in. below the top of the tank. If width of tank is over 24 in. a double lateral exhaust should be used with slots on both sides.

It has also been determined that a similar control may be used for tanks wider than 3 ft when the same velocity (2000 fpm) is maintained through a slot which is increased $\frac{1}{4}$ in. for every foot of width greater than 3 ft. When these slots must be extended more than 6 ft in length, some method of spreading the flow is necessary to provide even air flow distribution through the entire slot length. This can be accomplished by tapering the slot, which incidentally will add to the resistance of the system. A more economical approach is to place properly spaced vanes in the side ducts, or to branch the side ducts[17].

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile solvent vapors produced by spraying operations may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept well below 100 parts per million in the breathing zone of the worker. Vapors from many spraying operations are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located at the end of the booth opposite the opening. In front of this duct should be placed baffle plates which will cause a uniform air velocity distribution across the frontal area. The air volume should be sufficient to maintain a velocity of not less than 150 fpm over the open area of the booth.

Spray booths may be either the dry or wet type. The latter is the more modern design, provided with a water-wash section for the removal of the solid over-spray contaminants and for the absorption of water-soluble thinners or solvents.

The most modern innovation is the electrostatic spraying and detearing unit. Objects to be sprayed are passed through a high tension electrostatic field, which not only produces a more evenly sprayed surface, but materially reduces excessive over-spray. The detearing unit removes tear-drops of sprayed material from the edges or ends of air-drying sprayed objects as these objects pass through a high tension electrostatic field.

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should not exceed 100 fpm when the window is fully open.

Kitchen Hoods

The length and width of kitchen hoods should be such as to extend beyond the extreme projection of the ranges, broilers, etc., over which they are installed. The minimum projection or overlap should be 12 in. Where space conditions permit, range hoods should be about 2 ft high so as to provide a reservoir to confine momentary bursts of smoke and steam until the exhaust system can evacuate the hood. Range hoods should be located as low as possible to increase their effectiveness.

In general the amount of air to be exhausted from restaurant range hoods is at the rate of 100 cfm per square foot of face area. In some cases, where the application is principally frying and where it is not practicable to install a hood 2 ft high, it is recommended that the face velocity be increased from 100 to 150 fpm, depending on peak load conditions in the kitchen. Exhaust connections to range hoods should always be made at the top and back of hoods, and should be spaced preferably not more than 6 ft apart and be rectangular in shape with the long side parallel to the back of the hood. Exhaust openings into range hoods should be designed to maintain a velocity of 1500 to 1800 fpm.

An approved fire damper with fusible link should be (and is required by code in many states) installed in the main exhaust duct or branch adjacent to the range hood. Should there be more than one hood connected to a common duct, then the branch duct to each hood should be provided with a fire damper. Access doors should be provided at the fire damper for purpose of inspection, cleaning, or for renewal of fusible link. All exhaust piping to range hoods, commonly called grease ducts, should be provided with tight fitting cleanout doors of adequate size to permit easy removal of grease. Some engineers use filters to advantage in hoods which are subject to grease conditions.

Hoods over steam tables should be of construction similar to range hoods. It is good practice to design such hoods with a face velocity of 60 to 70 fpm. Hoods over dishwashing machines are usually relatively small and generally 1500 to 2000 cfm per hood is allowed, which is equivalent to a velocity of approximately 100 fpm per square foot of face area. Range hoods in diet kitchens are constructed the same as restaurant range hoods but with less exhaust air per square foot of face area, depending upon the nature of the food cooked.

Hoods are not often used in private residences unless they are quite large and the consideration of expense is not important. For such residences the hoods should be designed on the same basis as diet kitchens. Most all residence kitchens can be effectively and economically ventilated by the installation of a built-in kitchen ventilator, which should be located in an outside wall and in close proximity to the kitchen range. It has been found that the capacity of the built-in kitchen ventilator should be at least 350 cfm regardless of the size of kitchen. This can be justified on the

Table 4. Approximate Conveying Velocities

Material Conveyed	Design Velocity FPM
Vapors, gases, fumes, very fine dust	2,000
Fine dry dusts	3,000
Average industrial dusts	3,500
Coarse particles	3,500–4,500
Large particles, heavy loads, moist materials	4,500 and over

basis that the smaller the kitchen the more concentrated the heat will be thus requiring a more rapid rate of air change. Standard size built-in kitchen ventilators are generally available in three sizes, namely, 350, 500 and 800 cfm. The proper size to use will depend on design conditions and available wall space.

DUCT SYSTEM DESIGN

In designing a duct system it is necessary to recognize a few fundamental principles (see also Chapter 41). Knowing the quantity of air required, the size of the duct may be computed from Equation 6:

$$A = \frac{Q}{V} \tag{6}$$

where

A = cross-section area of duct, square feet.
Q = air quantity to be exhausted by the duct, cubic feet per minute.
V = velocity of air, feet per minute.

Air Velocities in Ducts

Where it is necessary to transport the particulate material collected in an exhaust system, minimum carrying velocities must be maintained in the ducts preceding the collector. It has been found that good results are obtained when design air velocities in horizontal runs are not less than 2000 fpm or not greater than 5000 fpm. When the dust being carried is organic and other than wood flour, or similar material, a velocity of 2500 fpm is adequate. Approximate required conveying velocities are given in Table 4.

For duct systems wherein the air has no dust or solid load, a lower velocity is desirable, which may range from 1500 to 2500 fpm. In view of the fact that the horsepower required by a system depends directly on the resistance and the resistance is a function of the velocity, economical design requires velocities of this magnitude.

The equal friction method is generally used for designing a duct system as this insures equal resistance to air flow in all branches throughout the system (see Chapter 41). Long main ducts do not generally provide the most economical layout. Where it is necessary to ventilate a large number of machines, or machines which are widely separated, it is desirable to locate the fan at approximately the center of the system. With this arrangement it is possible to choose a fan which will deliver the required air quantity against a lower resistance pressure, and this will generally result in a horsepower saving.

When a system carrying dust is designed with an oversize main duct to allow for future extension, the air velocity may be found to be too low to carry the dust, and serious plugging may occur. In this case it is desirable to install an orifice in the end of the pipe to allow for the lower air quantity.

Construction

The interior of all ducts should be smooth and free from obstructions at joints and soldered air-tight. Other sealing mediums are permissible where soldering is impracticable.

Ducts should be constructed of galvanized sheet metal except when the presence of corrosive fumes or gases, temperatures above 400 F, or other factors would make galvanized material impracticable. For the usual ex-

TABLE 5. GAGES OF METALS FOR EXHAUST SYSTEMS[a]

Diameter of Round Pipe or Greatest Dimension of Rectangular Pipe, Inches	Thickness of Duct Material U. S. Gage Number	
	For Highly Abrasive Matter	For Other Matter
Up to 8 inclusive	20	22
Over 8 to 18 inclusive	18	20
Over 18 to 30 inclusive	16	18
Over 30	14	16

[a]Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems (*American Foundrymen's Association*, p. 53).

haust systems the metal thicknesses shown on Table 5 are recommended. Elbows and angles should be a minimum of two gages heavier than straight lengths of equal diameter. Hoods should be a minimum of two gages heavier than straight sections of a connecting branch.

Longitudinal joints of ducts should be lapped and riveted or spot-welded on 3-in. centers maximum. Girth joints or ducts should be made with lap in direction of air flow, with 1 in. lap for duct diameters through 19 in. and $1\frac{1}{4}$ in. lap for diameters over 19 in. Elbows and angles should have an inside or throat radius of two pipe diameters whenever possible. Large radii are recommended for heavy concentrations of highly abrasive dusts. Elbows 6 in. or less in diameter should be constructed of at least 5 sections and, if over 6 in. in diameter, of 7 sections, with angles pieced proportionally. Hoods should be free of sharp edges or burrs and reinforced to provide necessary stiffness. Transitions in mains and sub-mains should be tapered with a taper 5 in. long for each 1 in. change in diameter whenever possible. All branches should enter the main at the large end of the transition at an angle not to exceed 45 deg or preferably 30 deg. Branches should be connected only to the top or sides of mains, with no two branches entering diametrically opposite to each other. Dead end caps should be provided within 6 in. from last branch of all mains and sub-mains. Cleanouts should be provided every 10 ft and near each elbow, angle, or duct junction in horizontal sections. Ducts should be supported sufficiently to place no loads on connected equipment and to carry weight of a system plugged with material. The maximum distance between supports should be 12 ft for 8 in. or smaller ducts and 20 ft for larger ducts. Six inches minimum clearance should be provided between ducts and the ceiling, wall or floor. Blast gates for adjustment of the system should be placed near the connection of a branch to the main and means of locking gates after the adjustments have been made should be included. Rectangular ducts should be used only when clearances prevent the use of round construction. Rectangular ducts should be as nearly square as possible. The weight of metal and the lap, and other construction details, should be the equal of round duct construction having a diameter equal to the longest side. All pipes passing through roofs should be equipped with collars so arranged as to prevent water from leaking into the building.

The main trunks and branch pipes should be as short and straight as possible.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely. The passing of pipes through fire-walls should be avoided wherever possible, and floor sweep connections should be so arranged that foreign material cannot be easily introduced into them.

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always good practice and is frequently done at the expense of a reduced air velocity, it is often done where future expansion of the exhaust system is contemplated.

Duct Resistance

The resistance to flow in round galvanized duct riveted and soldered at the joints may be obtained from Fig. 2, Chapter 41. The pressure drop through elbows depends upon the radius of the bend. For elbows whose

Table 6. Loss Through 90-Deg Elbows

Ratio of Elbow Center Line Radius to Pipe Diameter or Depth	Approximate Loss in Per Cent of Velocity Head
1	80
1½	31
2	22
2½	19

centerline radii vary from 100 to 250 per cent of pipe diameter, the loss may be estimated from Table 6.

RESISTANCE OF SYSTEM

The resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the duct system.

The loss through the hoods is usually assumed to be equal to one-half the suction at the hoods. Where possible the resistance of the particular collector to be used should be obtained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. The velocities should be found in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6 in this chapter and Fig. 2, Chapter 41. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[18].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level[19]. Commonly accepted

values of threshold limits for usual atmospheric contaminants will be found in Tables 3, 4 and 5, Chapter 10.

AIR FLOW PRODUCING EQUIPMENT

In any type of exhaust system some form of air flow producing equipment is required to create the pressure necessary to cause the air to flow through the system to the discharge stack. The principal types of air moving equipment are centrifugal exhaust fans, disc or propeller fans, axial flow fans and venturi ejectors.

TABLE 7. CORROSION RESISTING MATERIALS FOR EXHAUST SYSTEMS[a]

<table>
<tr><th rowspan="2">MATERIAL</th><th colspan="16">ACID[b]</th></tr>
<tr><th colspan="2">ACETIC</th><th colspan="2">CHROMIC</th><th colspan="2">HYDRO-CHLORIC</th><th colspan="2">HYDRO-FLUORIC</th><th colspan="2">NITRIC</th><th colspan="2">PHOS-PHORIC</th><th colspan="2">SUL-PHUROUS</th><th colspan="2">SUL-PHURIC</th></tr>
<tr><th>METALS</th><th>Dil</th><th>Conc.</th><th>Dil.</th><th>Conc.</th><th>Dil.</th><th>Conc.</th><th>Dil.</th><th>Conc.</th><th>Dil.</th><th>Conc.</th><th>Dil</th><th>Conc.</th><th>Dil.</th><th>Conc</th><th>Dil</th><th>Conc.</th></tr>
<tr><td>Aluminum</td><td colspan="2">Good</td><td colspan="2">Fair</td><td colspan="2">Poor</td><td colspan="2">No Data</td><td>Poor</td><td>Good</td><td colspan="2">Poor</td><td colspan="2">Poor</td><td colspan="2">Poor</td></tr>
<tr><td>Magnesium and Alloys</td><td colspan="2">No Data</td><td>Good</td><td>Poor</td><td colspan="2">No Data</td><td>Poor</td><td>Good</td><td colspan="2">No Data</td><td colspan="2">No Data</td><td colspan="2">No Data</td><td colspan="2">No Data</td></tr>
<tr><td>Lead and Lead-Coated</td><td colspan="2">Poor</td><td colspan="2">Good</td><td colspan="2">Poor</td><td colspan="2">Poor</td><td colspan="2">Poor</td><td colspan="2">Poor</td><td colspan="2">Good</td><td>Good</td><td>Poor</td></tr>
<tr><td>Moly Alloy (60 Ni—20Mo —20 Fe)</td><td colspan="2">Good</td><td colspan="2">No Data</td><td colspan="2">Fair</td><td colspan="2">No Data</td><td colspan="2">Poor</td><td colspan="2">Poor</td><td colspan="2">No Data</td><td colspan="2">Good</td></tr>
<tr><td>Monel Metal</td><td colspan="2">Fair</td><td colspan="2">Poor</td><td>Fair</td><td>Poor</td><td colspan="2">Good[d]</td><td>Fair</td><td>Poor</td><td colspan="2">Fair</td><td colspan="2">Poor</td><td>Good[e]</td><td>Poor</td></tr>
<tr><td>Bronze</td><td colspan="2">Poor</td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2">Good</td><td colspan="2"></td></tr>
<tr><td>Silicon Iron</td><td>Fair</td><td>Good</td><td colspan="2">No Data</td><td colspan="2">Fair</td><td colspan="2">Poor</td><td colspan="2">Good</td><td colspan="2">Good</td><td colspan="2">No Data</td><td colspan="2">Good</td></tr>
<tr><td>Stainless Steel[c] (18 Cr-8 Ni)</td><td colspan="2">Good</td><td colspan="2">Good</td><td colspan="2">Poor</td><td colspan="2">No Data</td><td colspan="2">Good</td><td colspan="2">Poor</td><td colspan="2">Good</td><td>Poor</td><td>Good[e]</td></tr>
<tr><td>Enameled Steel</td><td colspan="2">No Data</td><td colspan="2">No Data</td><td colspan="2">Good</td><td colspan="2">Poor</td><td colspan="2">Good</td><td colspan="2">Poor</td><td colspan="2">No Data</td><td colspan="2">Good</td></tr>
<tr><td>MISCELLANEOUS</td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td>Asbestos Comp.</td><td colspan="2"></td><td colspan="2"></td><td colspan="8">Good except against strong acids and alkalies</td><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td>Wood</td><td colspan="14">Some woods are decomposed or softened faster than others.</td><td colspan="2"></td></tr>
<tr><td>Rubber</td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td><td colspan="2">Poor</td><td colspan="2"></td><td colspan="2"></td><td colspan="2">Poor</td></tr>
<tr><td>Plastics</td><td colspan="16">In general plastics resist weak acids and are decomposed by concentrated acid.</td></tr>
</table>

[a] Standard Practice Sheet No 115 (Division of Industrial Hygiene, New York State Labor Department).
[b] Acid mists in air are more corrosive than as liquid in storage tank. Galvanized iron not resistant to acid.
[c] Stainless steel of (24 *Cr*—10 *Ni*) fairly resistant at low temperature for HCl and H_3PO_4.
[d] Under most conditions.
[e] At room temperatures.

PROTECTION AGAINST CORROSION AND ABRASION

Manufacturers generally provide special fans for the handling of various industrial wastes. When corrosive or abrasive materials are conveyed, the fan blades and interior of the fan housing should be protected from wear. This may be accomplished by placing the collector on the suction side of the fan. Excellent protection against many corrosive acids may be obtained by lining the interior surfaces of the ducts and fans, including wheels, with rubber.

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to chem-

ical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 7. Hoods and ducts, when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases as chlorine and hydrochloric acid.

REFERENCES

[1] Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems, Page 21, Industrial Hygiene Codes Committee, *American Foundrymen's Association.*

[2] How to Design Exhaust Hoods, by J. M. DallaValle (*Heating and Ventilating*, Series of 12 articles March, 1943 to February, 1944).

[3] Industrial Exhaust Ventilation in Industrial Hygiene, by Allen D. Brandt (A.S.H.V.E. Transactions, Vol. 50, 1944, p. 331).

[4] Fan Engineering, 1948, Buffalo Forge Co.

[5] Nature of Air Flow at Suction Openings, by A. D. Brandt, R. J. Steffy and R. G. Huebscher (A.S.H.V.E. Transactions, Vol. 53, 1947).

[6] Energy Losses at Suction Hoods, by A. D. Brandt and R. J. Steffy (A.S.H.V.E. Transactions, Vol. 52, 1946, p. 205).

[7] Industrial Health Engineering, by Allen D. Brandt (John Wiley & Sons, Inc., New York).

[8] For more detailed requirements refer to Fundamentals Relating to the Design and Operation of Exhaust Systems, Z9-1936 (*American Standards Association*). Industrial Code Bulletin Nos. 10 and 12 (New York State Labor Department). Principles of Exhaust Hood Design, by J. M. DallaValle (*U. S. Public Health Service*, 1939).

[9] Standards of the N.B.F.U. for Installation of Blower and Exhaust Systems for Dust Stock and Vapor Removal or Conveying, *National Board of Fire Underwriters.*

[10] National Electrical Code 1947, N.B.F.U. Pamphlet No. 70, *National Board of Fire Underwriters.*

[11] The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October, 1933).

[12] Studies n the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (*A.S.M.E. Transactions*, Vol. 54, 1932).

[13] Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 387).

[14] Low Velocity Exhaust Systems, by Theodore Hatch (*Heating and Ventilating*, October, 1940, p. 27).

[15] Tank Ventilating Power Costs Cut by Low Velocity Systems, by William B. Harris (*Heating and Ventilating*, July, 1942, p. 42).

[16] Health Hazards in Chromium Plating, by J. J. Bloomfield and William Blum (*U. S. Public Health Report*, Vol. 43, No. 26, September 7, 1928).

[17] New Data for Practical Design of Ventilation for Electroplating, by W. P. Battista, Theodore Hatch and Leonard Greenburg (*Heating, Piping and Air Conditioning*, February, 1941, p. 81). Ventilation of Plating Tanks, by Allen D. Brandt (*Heating, Piping and Air Conditioning*, July, 1941, p. 434).

[18] Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 353).

[19] Keeping Dust Under Control, by John M. Kane (presented to the 31st National Safety Congress, October 28, 1942, and reprinted in part in *National Safety News*, January, 1943). The Determination and Control of Industrial Dust, by J. J. Bloomfield and J. M. DallaValle (*Public Health Bulletin 217*, 1935), Engineering Control of Air Contamination of the Working Environment, by A. D. Brandt (In Manual of Industrial Hygiene, *U. S. Public Health Service*, 1943, p. 198–266).

CHAPTER 47

DRYING SYSTEMS

Mechanism of Drying, Omissions in Drying Cycle, Factors Influencing Drying Rates, Drying Methods and Equipment, Radiant Drying, Conduction Drying, Convection Drying, Dryer Calculations, Typical Solution of Dryer Problem

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THE term drying, in a broad sense, encompasses the removal of water, and occasionally other liquids, from gases, liquids, or solids. However, the common usage of the word confines the meaning principally to the removal of water or solvent from solids. Dehumidification is the term that is commonly assigned to the drying of gases. This is usually accomplished by condensation or adsorption by various drying agents and is treated in Chapter 38. Distillation and more particularly fractional distillation is associated with the drying of liquids.

It is usually more economical to employ, whenever possible, mechanical means of separating as much water as is practicable from the solid materials before undertaking drying or dehydration steps. These mechanical methods such as filtration, screening, pressing, centrifuging, or settling usually require much less power and frequently less capital outlay, thereby making the operation cheaper in terms of cost per pound of water removed.

MECHANISM OF DRYING

The removal of moisture from solids is largely dependent on the difference in vapor pressure between the moisture in the pores of the material and the surrounding air, and the rate of diffusion of internal moisture to the surface of the material. As long as the vapor pressure of the moisture within the material is greater than the vapor pressure of the surrounding air, moisture will travel outward to the surface of the material and be evaporated. While the magnitude of the vapor pressure difference is the force or potential that causes the drying, it is not representative of the actual results in terms of time or removed moisture.

Water existing on the surface or held in the voids of a solid is known as *free* or *unbound* moisture and its vapor pressure will be equal to that of water at the same temperature. *Bound, hygroscopic* or *chemically combined* moisture on the other hand is intimately associated with the physical nature of the material and its vapor pressure during drying is less than that of free water at the same temperature.

The course of drying of any substance under constant conditions of temperature, humidity and air velocity and distribution follows a definite pattern which can be represented by graphs such as Fig. 1 and Fig. 2.[1]

Fig. 1 may be considered a typical *drying time* curve for solid material. Fig. 2 is the *drying rate* curve derived from Fig. 1, the ordinate in this case being the drying rate which can be expressed as pounds of water evaporated per hour per square foot or pounds of water evaporated per hour per pound of dry stock.

In Fig. 2 the section CB is attributed to a warming up period during which the surface of the solid is reaching a uniform temperature. This temperature is maintained constant during the period of a *constant drying*

rate shown by BA. During the constant rate period water is being evaporated from the surface of the material at a rate comparable to that of a free water surface and moisture is being supplied by diffusion or capillarity to the surface at a rate equal to or greater than the evaporative rate. During this period the material is generally considered to assume the wet bulb temperature of the air, although this is not strictly true as, in many cases, radiant heat is absorbed or some heat is supplied by conduction.

AD represents the first stage of the falling rate period and when this portion of the curve is linear it is assumed that the surface, although still wet, is gradually drying out. The moisture content corresponding to point A is termed the *critical* moisture content and it is reached when the moisture reaches the surface at a rate less than the potential evaporative rate. The portion of the rate curve corresponding to DE represents the period of drying where the outer surface has become dry and subsurface

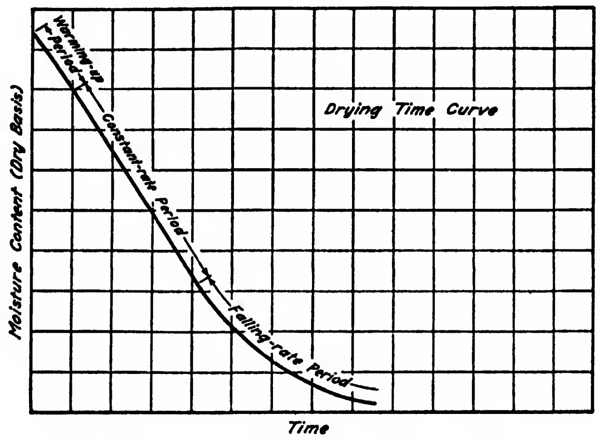


FIG. 1. MOISTURE CONTENT VS. DRYING TIME[1].

evaporation is taking place. During the period A to E, the temperature of the material gradually increases and tends to approach the dry bulb temperature of the air.

Finally, all drying stops at the *equilibrium moisture content* (point E on the curve) where the vapor pressure of water in the air is equal to the vapor pressure of water in the material. The equilibrium moisture content varies with the humidity of the air and the hygroscopic properties of the material as explained in Chapter 45, Industrial Air Conditioning.

The drying rate during the constant rate period is subject to mathematical analysis and can be predicted with some degree of accuracy but calculations involving the falling rate period are seldom satisfactory because of the number of variables involved. The flow of water within the material may be caused by a combination of many factors some of which are capillarity, pressure due to shrinkage, true diffusion of liquid moisture, the force of gravity and a sequence of vaporizations and condensations within the solid. Because of this complexity of internal moisture flow and the tremendous number of solid materials with different internal structures, no satisfactory theory has yet been developed for predicting the rate of this flow and hence the rate of drying. As a result, the fundamentals of a practical approach to drying have been based on the

external drying conditions; i.e. the temperature, humidity and velocity of the drying air, and the physical conditions of the material being dried. Prediction of drying rates may be based on past experience but are preferably obtained from experiments employing the conditions which are expected in commercial large-scale practice.[1]

Omissions in Drying Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the

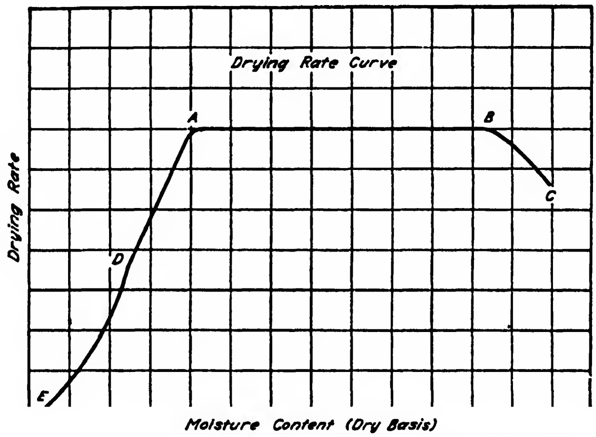


FIG. 2. RATE OF DRYING VS. MOISTURE CONTENT[1]

intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the falling rate period being confined in practice to unsaturated surface drying.

MOISTURE CONTENT

The moisture content of solid materials can be expressed in terms of per cent on *wet* or *dry* basis. The per cent moisture on a wet basis is the ratio of the weight of moisture in the sample to the total weight of the sample whereas on a dry basis it is the ratio of the weight of moisture in the sample to the weight of solids in the sample. A vegetable, for example, having 80 per cent moisture on a wet basis would have 80 grams of moisture and 20 grams of bone dry solids in a 100 gram sample. On a dry basis, the moisture content would be 4.0 (400 per cent).

In drying calculations the dry basis is usually preferred because small variations in initial moisture content expressed on a wet basis can lead to serious errors. For example, if a quantity of 100 lb of material is to

be produced having a final moisture content of 5 per cent wet basis and having initial moisture content of 85 per cent, the weight of solids involved is 95 lb and the initial amount of water is 538 lb. The water to be removed is therefore 533 lb. If the initial moisture content should be increased to 87 per cent, then the initial amount of water is 636 pounds and 631 pounds must be removed. Hence a 2 per cent variation in initial moisture content results in almost 20 per cent increase in required drying capacity. The use of the dry basis, however, is not misleading.

FACTORS INFLUENCING DRYING RATES

Temperature

The driving force for evaporation is directly proportional to the difference between the absolute humidity of the air being used and the absolute hu-

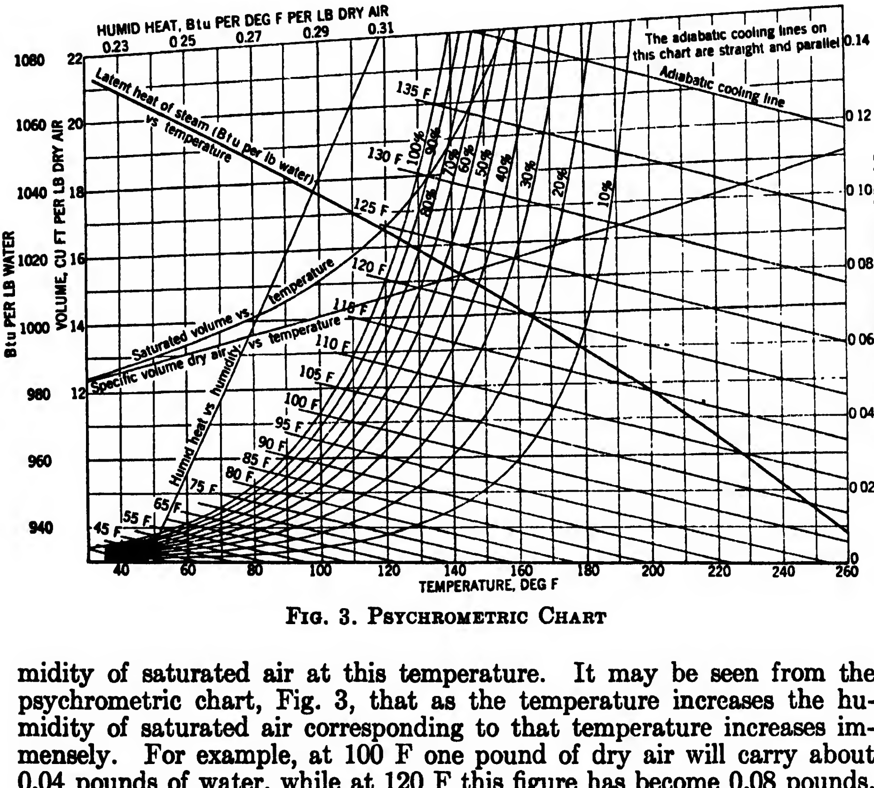


FIG. 3. PSYCHROMETRIC CHART

midity of saturated air at this temperature. It may be seen from the psychrometric chart, Fig. 3, that as the temperature increases the humidity of saturated air corresponding to that temperature increases immensely. For example, at 100 F one pound of dry air will carry about 0.04 pounds of water, while at 120 F this figure has become 0.08 pounds. Thus during the constant rate and uniform falling rate periods an increase in temperature will increase the drying rate. During the varying falling rate period, i.e., sub-surface drying where diffusion is the controlling factor, an increase in temperature will lower the viscosity permitting a more rapid drying. Temperature and thickness of the material are the only external variables which affect the drying rate during the variable falling rate period. The drying rate varies inversely with the square of the thickness.

In general, then, it is preferable to use as high a temperature as possible for any drying operation because of the faster rate of drying and the greater capacity of the air to remove water. During the constant rate

period of drying, the temperature of the material approximates the wet-bulb temperature of the air. Therefore, high dry-bulb temperatures may be employed.

The nature of the material and its characteristics often determine the temperatures that can be employed with safety. For example, chemically combined water may be lost, cells may be steam exploded, charring may take place or exothermic reactions initiated if the product temperature exceeds a critical value.

To illustrate the effect of temperature refer to Fig. 4, a typical vegetable drying curve.[2] Fig. 5 shows the change in drying time as affected by dry-bulb temperature change–other variables remaining constant.

Humidity

As previously pointed out, absolute humidity is the driving force for evaporation, therefore, a low humidity increases the drying rates. However, certain materials if dried too rapidly may case harden, i.e., form a nearly impermeable skin which greatly reduces the drying rate. Also, certain undesirable physical changes may take place such as the cracking and warping of lumber. In such instances, it may even be necessary to increase artificially the humidity to assure satisfactory drying by the recirculation of moist air or the use of water or steam jets.

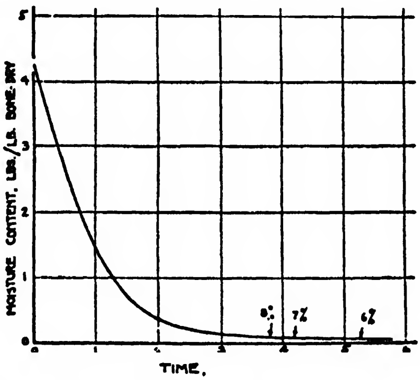


Fig. 4. Typical Vegetable Drying Curve

Russet potato, blanched $\frac{5}{16}$-in. strips, wood-slat tray, load 1.5 lb. per square foot, cross-circulation, air velocity 500 fpm, air temperature 150 F, wet-bulb temperature 90 F.

In the case of substances that must be dried at relatively low temperatures (120–140 F) it is often necessary to dehumidify the air to secure satisfactory drying rates. In such instances, a closed system using the adsorptive methods described in Chapter 38, is frequently applicable.

Fig. 6 shows the effect of change in humidity on the rate of drying curve for the same material considered in Figs. 4 and 5.

Air Circulation

Air velocity and distribution are important factors only in the first two periods of drying when surface evaporation is taking place. During the varying falling rate period of gaseous diffusion enough ventilation must

be provided to prevent stagnation. The velocity of the air passing over the surface to be dried determines the rate at which the moisture-laden air is swept away. The more rapidly this moist air can be removed the more rapidly can evaporation occur. It has been demonstrated that this drying rate is a function of the 0.8 power of the velocity. Air directed perpendicularly to the drying surface exhibits the greatest efficiency in dispersing the *dead air* film. The limiting factors in air velocity are the power requirements to remove the air and the danger of blowing away the lighter particles of material.

Fig. 7 indicates the effect of changes in air velocity for the same material previously considered.

Miscellaneous Items

Many other factors affect the rate of drying curve such as size and shape of pieces, thickness of layer, type of tray or other support, mode of

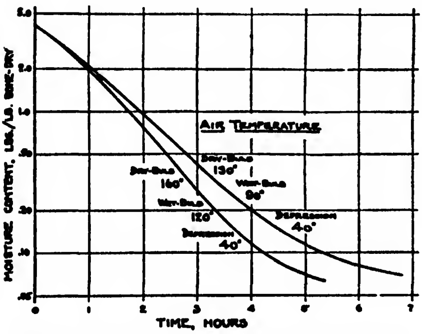


FIG. 5. DRYING CURVES AT DIFFERENT TEMPERATURE LEVELS

Russet potato, blanched $\frac{3}{16}$-in. strips wood-slat tray, load 1.5 lb per square foot, cross-circulation, air velocity 500 fpm.

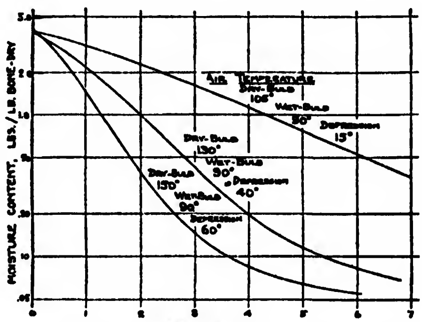


FIG. 6. DRYING CURVES AT DIFFERENT WET-BULB DEPRESSIONS

Russet potato, blanched $\frac{3}{16}$ in. strips, wood-slat tray, load 1.5 lb per square foot, cross-circulation, air velocity 500 fpm.

exposure to air stream, and relative amounts of heat received by radiation, conduction and convection. Fig. 8, for example, illustrates the effect of changes in layer thickness.

DRYING METHODS AND EQUIPMENT

Drying systems are sometimes classified according to the method of heat transfer that is employed since the entire problem of drying resolves itself into individual problems of heat transfer and the thermodynamics of air and water vapor. The methods of heat transfer are radiation, conduction and convection. Many types of dryers have been built on these principles for different purposes.

Drying systems can also be classified according to the method of product handling—thus batch operation, semi-continuous and continuous.

Radiant Drying

Sun drying, the oldest form known to man, is still practiced where the material is amenable to such treatment, where the necessary time can be allowed, and where there is little danger of rain or atmospheric pollution.

In artificial systems radiating surfaces, heated by steam, electricity or other means, afford a good method of heat distribution and control. Ra-

diant heating sets up convection currents, and in low-temperature dryers only about one-third to one-half of the total heat for evaporation is actually supplied to the material by radiation. At high temperatures the radiation output increases rapidly, according to the *fourth-power law*. The total radiation may be computed by the equations and tables given in Chapter 5. In general, fins and irregular surfaces do not increase radiation, hence the area to be used in calculations is the area of a smooth-surface envelope enclosing the radiating elements.

A certain amount of air circulation is required through a radiant dryer, in order to carry off the vapor.

Radiant heat from infra-red lamps has been accepted by certain industries as practicable for their specific problems. An example of successful application is found in the drying of lacquers.

Electric heating by induction and dielectric means, as described in Chapter 30, is applicable to the drying of certain materials.

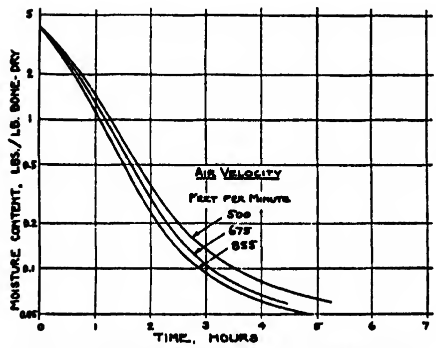


FIG. 7. DRYING CURVES AT DIFFERENT AIR VELOCITIES

Russet potato, blanched $\frac{3}{16}$-in. strips, wood-slat tray, load 1.5 lb per square foot, cross-circulation, air temperature 150 F, wet-bulb temperature 90 F.

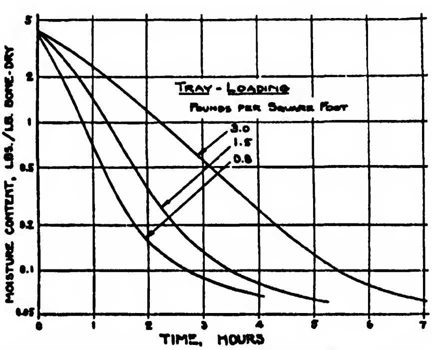


FIG. 8. DRYING CURVES AT DIFFERENT DENSITIES OF LOADING ON TRAYS

Russet potato, blanched $\frac{3}{16}$-in. strips, wood-slat tray, cross-circulation, air velocity 500 fpm, air temperature 150 F, wet-bulb temperature 90 F.

Conduction Drying

Drying rolls or drums, Fig. 9[4], flat surfaces, open kettles and immersion heaters are examples of the direct-contact method. Intimate contact of the material with the heating surface is important, and in some cases agitation is desirable to increase the uniformity of heating or to prevent overheating.

Greatest resistance to heat transfer occurs on the air side of the material being dried. The rate of heat transfer from the surface of the heated material to the air, and hence the rate of drying, may be increased by: (a) forced convection or air circulation and (b) vacuum operation to lower the boiling point of the liquid being evaporated.

A rather interesting method of conduction drying was put into practical use during the war for the drying of blood plasma and has since been expanded to other fields such as the preservation of bacteria and other micro-organisms. This has come to be known as freeze drying or drying by sublimation. The material to be dried is first frozen and then placed in a high vacuum chamber connected to extremely low temperature con-

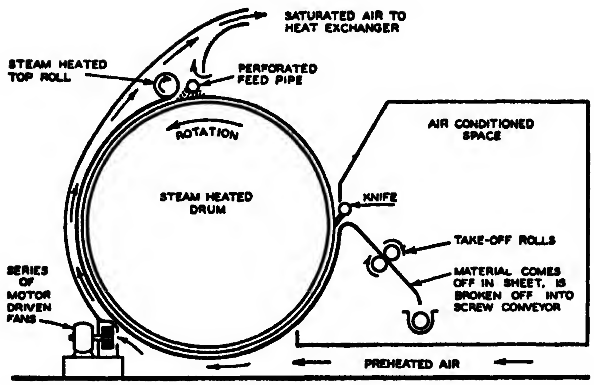


FIG. 9. DRUM DRYER[4]

densers. The water is removed by vaporizing from the solid directly to the gas without ever becoming liquid.

Convection Drying

A limited amount of convection drying takes place in almost any dryer such as those described in the preceding paragraphs. However, to be classified as a convection dryer the principal source of heat is the heated air or other gases circulated in the dryer. There are a number of mechanical means of accomplishing this circulation of air or gases, each of which has some particular virtue. Brief descriptions of some important types of convection dryers follow:

Rotary Dryers. These dryers are cylindrical drums usually having internal flights, as shown in Fig. 10, which cascade the material being dried through the air stream. The driers may be heated directly or indirectly and the air circulation may be parallel or counter flow. A variation is the rotating louvre type dryer, which introduces the air beneath the flights thus securing very intimate contact.

Cabinet and Compartment Dryers. These are generally considered batch dryers wherein each charge is dried to completion before removal.[4] A wide range includes types from the heated loft with only natural convection, and usually poor and non-uniform drying, to the self contained units with forced draft and properly designed baffles which give positive results. It is also possible to evacuate some of the systems for low temperature drying of delicate or hygroscopic materials. These dryers are usually loaded with material spread in trays to increase the exposed surface.

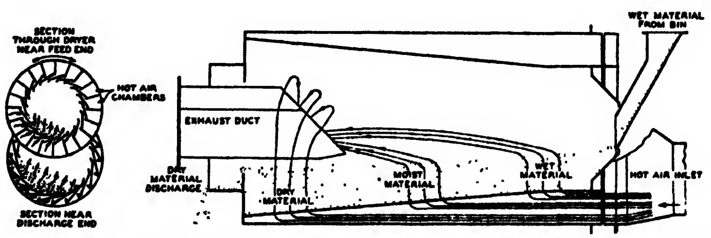


FIG. 10. CROSS SECTION AND LONGITUDINAL SECTION THROUGH ROTARY DRYER[4]

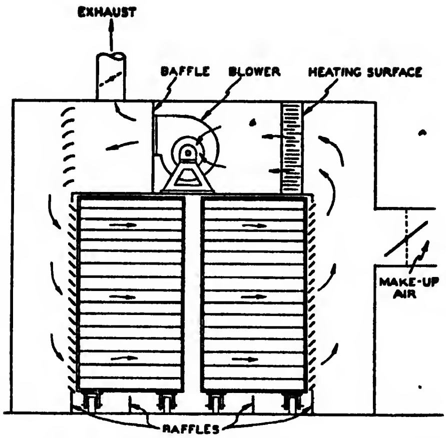


FIG. 11. COMPARTMENT DRYER, SHOWING TRUCKS, WITH AIR CIRCULATION[4]

The trays are loaded directly into the dryer or may be stacked on trucks which are wheeled in. (See Fig. 11).

Tunnel Dryers. Tunnel dryers are a modification of the compartment dryer and as a rule are continuous or semi-continuous in operation. Heated air or combustion gas is usually circulated by means of fans although a few natural draft units are still in use. The material is handled on trays or racks on trucks and moves through the dryer either intermittently or continuously. The air flow may be parallel, counter flow or a combination of the two, obtained by center exhaust. Further, the air flow may be across the surface of the trays or up or down through the bed or in any combination of directions. By reheating the air in this type of dryer or recirculating it a high degree of saturation is achieved before exhausting the air. This reduces the waste of sensible heat.

A variation on this type dryer is the strictly continuous type having one or more mesh belts which travel through the dryer carrying the product, such as Fig. 12. Innumerable combinations of temperature, humidity and air direction and velocity are possible. The labor requirement is low on such a dryer as it can be loaded and unloaded mechanically. There is the disadvantage of hot air leaks at the entrance and exit although these can be minimized by means of baffles or inclined ends where the material enters and leaves from the bottom.

Spray Dryers. In recent years the spray dryer has become important for the drying of liquids in many fields, especially in the food industry.

The liquid is atomized by means of pressure nozzles, air jets or centrifugal bowls into the air stream of a tower or chamber. Inlet air tempera-

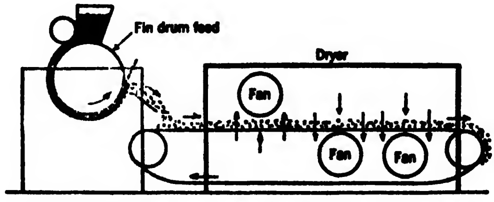


FIG. 12. SECTION OF CONTINUOUS DRYER, BLOW-THROUGH TYPE

tures may run from 250–300 F up as high as 1200 F. Drying is almost instantaneous because of the minute particle size. The dry powder is separated from the air by cyclone separators which are sometimes followed by cloth bags or scrubbing towers.

Because of the high inlet temperatures and the relatively large volume of air required the efficiency of the spray dryer is not too good and consequently is seldom used for dilute solutions (less than 30 per cent solids).

Figure 13 shows a typical arrangement for a spray drying system.[4]

A common and important feature of all spray processing is the direct conversion of the spray liquid to a granular product suitable for packaging without grinding or other intermediate handling. Another aspect is the unusually high rate of drying attained. In a well designed system 15 to 30 seconds is a fair time for the passage of the sprayed partical through the drying zone; the partical temperature need not rise materially above the wet bulb temperature of the liquid solution. This makes the process particularly adapted to the drying of heat sensitive material, some of its most important applications being the drying of milk, eggs, potato flour, soap and blood.[3]

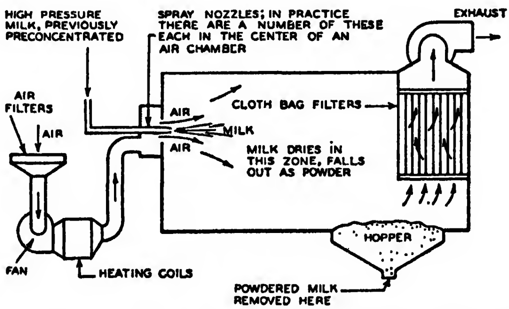


Fig. 13. Spray Dryer of the Pressure-Spray Rotary Type[4]

DRYER CALCULATIONS

The calculations for drying during the constant-rate period are different from those applying to the falling-rate period, and in contrast are subject to relatively simple mathematical analysis.

The rate of drying by air passing over a wet surface is directly proportional to the vapor pressure difference, and also proportional to the 0.8 power of the air velocity. For practical calculations, the wet surface is assumed to attain the wet-bulb temperature of the air passing over it, and evaporation takes place at constant rate under equilibrium conditions. The equation may then be expressed in three forms:

$$R = C\,A\,V^{0.8}\,(\Delta P) \tag{1}$$

$$R = C'\,A\,V^{0.8}\,(\Delta H) \tag{2}$$

$$R = C''\,A\,V^{0.8}\,(\Delta t) \tag{3}$$

where

R = rate of drying during constant-rate period, pounds of moisture per hour.

A = area of bed or material in contact with air, square feet.

V = air velocity over material, feet per minute.

ΔP = difference between vapor pressure at wet-bulb (surface) temperature and at dew-point of air.

ΔH = difference between humidity ratio of saturated air, at the surface temperature, and the actual humidity ratio of the air stream, pounds of water per pound of dry air.

Δt = difference between dry-bulb and wet-bulb temperatures of air, *i.e.*, the wet-bulb depression.

C, C', C'' = proportionality constants (for numerical values consult references).

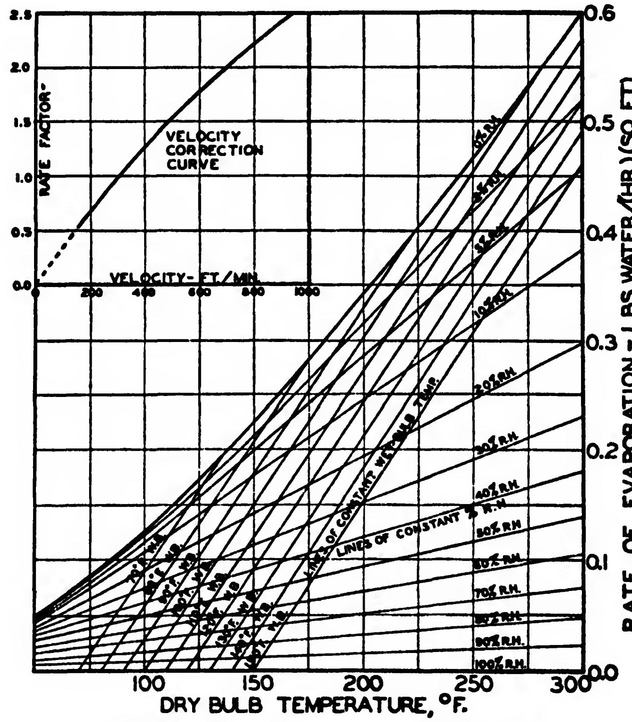


Fig. 14. Rate of Evaporation Chart[6]

These equations are useful mainly for computing the effects of changes in operating conditions, such as changes in air velocity, air temperature, humidity and surface area. The equations assume that the material is in equilibrium at the wet-bulb temperature of the air. If equilibrium has not been reached, or if heat is being added to the charge by radiation or conduction, such conditions must be taken into account. For large tray dryers or continuous surfaces, the logarithmic mean difference should be substituted for the simple difference in ΔP, ΔH and Δt.

Figure 14 permits a ready estimate to be made of the relative rates of drying for various air temperatures and humidities. The chart is based on the difference between the dry-bulb and wet-bulb temperatures of the entering stream of air and on air velocity of 300 fpm. It may be assumed satisfactory for any material in the constant-rate drying period.

A correction curve to correct the air velocity in any given problem is incorporated in Fig. 14. This curve is based on the variation of drying rate with the 0.8 power of the velocity.

The use of Fig. 14 in practical drying problems is as follows: Since the drying conditions of temperature and relative humidity are fixed, the corresponding absolute drying rate is read from Fig. 14. This value is then multiplied by the correction factor corresponding to the air velocity employed. The rate so obtained, however, does not include any effects of radiation or of conduction through unwetted surfaces. These effects tend to increase the rate of evaporation so that the chart is conservative.[5]

While it has been demonstrated empirically that the rate of drying during the falling-rate period is approximately proportional to the free water content of the material, actual calculations of drying during the falling-rate period are not satisfactory because of the number of variables involved. There are always present too many unknowns or experimentally determined coefficients, constants and limits. Consequently, whenever unbound moisture is to be removed, it is best to determine drying time by means of pilot tests as indicated by Fig. 4.

Although Fig. 14 is applicable only to the constant-rate period of drying, it has been found that some materials exhibiting a distinct falling-rate period are influenced by velocity, humidity and temperature in approximately the same degree as is the case for the constant-rate period (see Figs. 4 to 7). For these materials a knowledge of the drying time under any set of conditions can be used to predict the drying time under other conditions by analogy to the constant rate period.

The following nomenclature will be used in the discussion of design calculations:

H = humidity ratio of air, pounds of water vapor per pound of dry air

G = pounds of dry air supplied to the dryer per unit of time.

S = pounds of stock dried per unit of time in a continuous dryer.

S' = pounds of stock charged per batch to a discontinuous dryer.

θ = time.

Q = total heat supplied to the dryer.

t = air temperature.

t' = stock temperature.

t'' = average stock temperature over short time interval, in a batch dryer.

t_w = wet-bulb temperature.

s' = specific heat of the stock.

B = total radiation and conduction losses per unit time.

w = pounds of water per pound of dry stock.

r = heat of evaporation of water.

s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F deg.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters, and (2) where it leaves the dryer.

Air dryers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the dryer as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the*

drying equipment itself. In any continuously operating dryer of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\ (H_2 - H_1) = S(w_1 - w_2) \tag{4}$$

In discontinuous dryers, *e.g.*, compartment dryers, the drying operation is given by the equation:

$$G(H_2 - H_1) = S'\frac{dw}{d\theta} \tag{5}$$

In the continuous dryer, the heat consumption per unit time is:

$$\frac{Q}{\theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)(H_2 - H_1) + S(t'_2 - t'_1)(s' + w_1) + B \tag{6}$$

Equation 6 assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average values of t, t' and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S'(t''_2 - t''_1)\ (s' - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically, the error introduced by using a small number of long periods and employing average values of the variables over each is not serious. The evaluation of Equation 5 may be approximated in a similar manner.

The first term of the right hand member of Equation 6 represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air into contact with the stock with sufficient intimacy so that the air leaving the dryer is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air, due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air are imperative.

The second term of the right member of Equation 6 represents the latent heat of evaporation of the water plus the heat to raise this water to the temperature of evaporation. The third term of the equation represents heat to raise the temperature of the stock plus the water which remains unevaporated in the stock.

The changes taking place in the air during the drying process can be

illustrated on the skeleton psychrometric chart Fig. 15. The case illustrated is typical of tunnel and rotary dryers where heat is applied to the air at one point only. After the first adjustment stage during which both the material and the dryer reach the working temperature, the only heat losses from the dryer are those of radiation and conduction from the housing and these are practically negligible for an insulated dryer. Hence, the drying process can be considered to be adiabatic.

If 100 per cent outside air is used, the air can be considered to enter at point A, Fig. 15 (the prevailing outside air condition) and be heated to point B (the maximum permissible temperature t_m or the temperature determined by previous test). As the air evaporates moisture, it cools along the constant wet-bulb line BD to point C. The difference between the moisture content of air at B and at C represents the moisture *pick up* of the air. The maximum possible *pick up* from B to D is never achieved in practical dryers; the actual *pick up* being anywhere from 10 to 75 per cent of the maximum.

In order to conserve heat and to control the wet-bulb temperature at

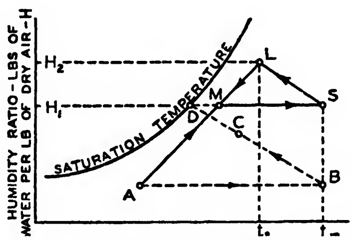


FIG. 15. CHANGES IN AIR DURING DRYING PROCESS

which the drying takes place, recirculation is used. The process is shown on Fig. 15. The outside air at A is mixed with recirculated air until the moisture level is raised to the desired point. The mixture is represented at point M, the heaters heat the mixture to the desired dry-bulb temperature t_m at point S. The moisture is picked up from S to L. Point L is the condition at which air is exhausted.

Actual dryer operation is somewhat more complicated because even if radiation and conduction losses are neglected, the wet-bulb temperature of the air remains constant only as long as surface evaporation of water is taking place. When sub-surface evaporation is occurring, some heat from the air is used to heat the material and hence there is a drop in wet-bulb temperature. Fig. 16 illustrates the drying process in a tunnel dryer in which the air is flowing parallel to the product.[1]

In design calculations using Equation 6, the following steps outline the procedure:

1. The unit drying rate, pounds of water per hour, is determined from the experimental drying time curve and the amount of product to be dried per hour. The drying time may also be approximated from previous experience.

2. The experimental data or experience also determine the drying condition, i.e., point L Fig. 15. This fixes H_2. Where experimental data are lacking L may be approximated from Regain Tables (see chapter Industrial Air Conditioning) since the relationship between the vapor pressure in the product and in the air at equilibrum for the desired final moisture content must prevail in the dryer. The dryer temperature must be not greater than the maximum permissible product temperature.

3. The rate of air circulation G must be determined and also the supply air condition S. In the design of some dryers such as rotary or tunnel types, it is customary to determine S first and then to calculate the air rate G. In other types of dryers, such as tray dryers or through circulation dryers where a fixed air velocity is maintained, G is calculated first and then point S is found. Because of the many variables involved, it is generally not possible to select S except on the basis of past experience or on the basis of experimental drying tests.[4]

4. The prevailing outside air conditions establish point A and hence the line A L. The per cent of recirculated air can then be calculated.

5. The physical arrangement of the dryer must then be selected to handle the desired quantity of product and at the same time circulate the calculated air quantity at the desired velocity.

6. Equation (6) can then be used to calculate the heat requirements.

TYPICAL SOLUTION OF DRYING PROBLEM

Since there are so many types of dryers which may be used, and so many special conditions surrounding each particular problem, it is usually recommended that those having experience with the dryer to be used be consulted. The following example, however, will serve as a guide for typical dryer calculations.[6]

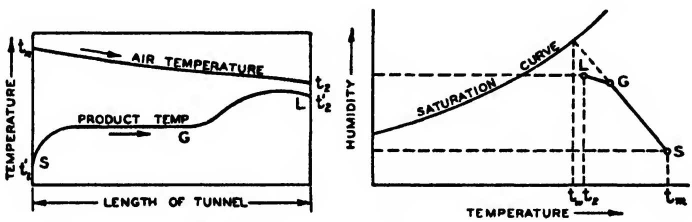


FIG. 16. TEMPERATURE AND MOISTURE CONDITIONS IN A TUNNEL DRYER
Parallel Flow Air and Product

Example 1. Assume 900 lb per hour of ceramic powder is to be produced. The powder has a specific heat of 0.22 and density of 98 lb per cu ft, wet. Initial moisture content is 19 per cent on a wet basis; final moisture content is to be one-half of one per cent on a wet basis.

A continuous belt dryer is a logical choice and previous experience indicates that rubber belts will withstand temperatures up to 200 F which is also about the highest desirable product temperature. Experience also indicates that a drying time of 45 min is possible at about 160 F dry-bulb and 100 F wet-bulb.

Step 1: Let x = pounds moisture at final condition.

Then,
$$\frac{x}{900} = 0.005$$

or,
$$x = 4.5 \text{ lb moisture}$$

and therefore the solid will amount to 895.5 lb.

Likewise the weight of the initial moisture x can be found

from
$$\frac{x}{895.5 + x} = 0.19$$

or
$$x = 210 \text{ lb.}$$

The weight of moisture to be removed is 205.5 lb per hour and wet material entering dryer is 1105.5 lb per hr.

Step 2: Previous tests indicate that a ½ in. layer of powder gives satisfactory results, and that a desirable air velocity is 50 fpm applied at a right angle to the belt. Based on 45 min (¾ hr) drying time, the dryer holding capacity will have to be

High Temperature Drying, by Burt S. Harrison (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 7).

The Temperature of Evaporation, by W. H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 25).

Address on Dehydration, by H. C. Gore (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 323).

Drying as an Air Conditioning Problem, by A. W. Lissauer (A.S.H.V.E. TRANSACTIONS, Vol. 27, 1921, p. 251).

A Chronological Survey of Drying and Driers, by J. E. Bolling (A.S.H.V.E. JOURNAL, Vol. 27, October, 1921, p. 715).

Thermodynamic Properties of Moist Air, by John A. Goff and S. Gratch (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 125).

Adiabatic Drying of Hygroscopic Solids, by A. M. McCready and W. L. McCale (*Transactions, American Institute Chemical Engineers*, 1933).

Limitations of Diffusion Equations in Drying, by O. A. Hougen, H. J. McCauley and W. R. Marshall (*Transactions, American Institute of Chemical Engineers*, 1940).

Factors Influencing the Performance of Rotary Dryers, by C. F. Prutton and C. O. Miller (*Transactions, American Institute of Chemical Engineers*, Part 1, February, 1942; Part 2, August, 1942).

Drying of Solids by Through Circulation, by W. R. Marshall and O. A. Hougen (*Transactions, American Institute of Chemical Engineers*, 1942).

Factors that Influence Drier Performance, by A. Weisselberg (*Chemical and Metallurgical Engineering*, August, 1932).

Typical Dryer Calculations, by O. A. Hougen (*Chemical and Metallurgical Engineering*, January and March, 1940).

Symposium on Drying, Articles by W. K. Lewis, W. H. Carrier, A. E. Stacey, Jr. and R. S. Fleming, R. G. Metz, G. B. Ridley, C. O. Lavett, D. J. Van Marle (*Industrial and Engineering Chemistry*, May, 1921, pp. 427–460).

Principles of Drying Lumber and Humidity Diagram, by H. D. Tiemann (*Forest Service Bulletin* 104, 1912).

The Drying of Solids, by T. K. Sherwood (*Bulletin Massachusetts Institute of Technology*, Nos. 237, 247 and 258).

Radiant Energy Drying with Heat Lamps, by T. P. Brown, *Metal Industry*, Vol. 37, Dec. 1939, p. 607).

Drying by Sublimation, by E. W. Flosdorf (*Food Industry*, Vol. 17, Jan. 1945, p. 607).

Evaporative Drying System, by F. H. Slade (*Food Manufacturing*, Vol. 18, March, 1943, p. 70).

High Frequency Methods in Gluing and Drying Wood, I. R. Berkness (*Wood Products*, Vol. 45, 1940, p. 12).

Development of the Unit Operations of Chemical Engineering: Drying, by T. K. Sherwood (*Chemical & Metallurgical Engineering*, Vol. 42, p. 214).

The Spray Dryer—Its Possibilities in Industry, by D. W. Biocheno (*Food Manufacturing*, Vol. 19, June, 1944, p. 195).

Mechanism and Rate of Drying by Near-infra-red Radiation, by L. E. Stout, K. J. Caplan and W. G. Baird (*Transactions American Institute of Chemical Engineers*, Vol. 41, 1945, p. 283).

Electronic Dehydration of Foods, by V. W. Sherman (*Electronics*, Vol. 17, 1944, p. 94).

Air Conditioning and Engineering, American Blower Co., 1935.

Drying in Industrial Plants, by J. O. Ross.

Elements of Chemical Engineering, by Badger and McCabe (McGraw Hill Co., 1931).

Fan Engineering, Buffalo Forge Co.

Die Trockentechnik, by M. Hirsch (Julius Springer, Berlin, 1932).

Drying (*Kent's Mechanical Engineers Handbook*).

Drying, by W. H. Carrier (*Marks' Mechanical Engineers Handbook*).

Drying (*Perry's Chemical Engineers Handbook*).

Kiln Drying of Lumber, by A. Koehler and R. Thelen, New York, 1926.

Kiln Drying of Lumber, by H. D. Tiemann (Lippincott, 1920).

Modern Drying Machinery, by H. B. Grenshaw, London, 1926.

Principles of Chemical Engineering, by Walker, Lewis, McAdams (Chapters on Evaporation, Humidity and Drying, McGraw Hill Co.).

CHAPTER 48

TRANSPORTATION AIR CONDITIONING

Railway Passenger Car Air Conditioning, Passenger Bus Air Conditioning, Streetcar and Trolley Coach Heating and Ventilating, Automobile Air Conditioning, Airplane Air Conditioning

THE principles of air conditioning applying to stores, restaurants, hospitals, theaters, and homes are applicable to railway passenger cars, passenger buses, automobiles, streetcars, trolley coaches, airplanes and ships. However, equipment used for mobile applications differs from that used for stationary purposes in that it must meet additional requirements. Equipment must be compact, accessible for quick inspection and servicing, light-weight and unaffected by vibration and impact. Freedom from vibration which could be transmitted to supporting vehicle and thus to passengers is essential. (For Air Conditioning Ships see Chapter 49.)

RAILWAY PASSENGER CAR AIR CONDITIONING

The railway passenger car represents a very difficult air conditioning problem. Space is strictly limited so that all equipment and ducts must be reduced to minimum size. Electric power supply and water supply also are limited. All equipment must withstand severe vibration and shock and must be very reliable in performance since major servicing points are frequently far apart.

During heating season it is necessary to heat conventional cars with steam from locomotive at pressures that may vary from 250 psi to only 5 or 10 psi on last car in long trains. Passengers in window seats must sit only a few inches from cold outside walls and windows, and must also be very close to standing radiation installed along sides of cars. Sudden changes in load may be caused by changes in sun, wind, or train movement. Even in coldest weather, outside doors must be opened frequently.

During cooling season, the problem is further complicated by high concentrated internal load represented by the passengers. Also, air distribution problems are increased by low ceilings and short air throws.

Heating

The heating of passenger cars is accomplished by using a split system consisting of an overhead air circulating system with heating and cooling coils and standing radiation (floor heat) along car sides. The floor heaters, which usually consist of finned tubing, may be made more efficient by using covers designed to increase gravity air circulation and to direct the warm air from finned heating surface along cold outside walls and car windows. In some new cars, wall convector panels are used and extend the full length of the car with air intakes along the floor and outlets at window-sill height and at window head height in dead-light panels. The heated wall panel protects passengers from cold outside walls and the chimney effect of slender panel duct increases air flow and improves heating surface efficiency.

Steam may be used directly in finned tubing or steam may be used to heat a liquid, (usually a mixture of water and diethylene glycol), which is

circulated through the finned tubing by means of circulators. If steam is used directly, it is difficult to distribute the heat uniformly along the length of the car. In the case of steam it is customary to divide long finned tubes into separate sections which are fed independently. This is not true zoning since it is not based on principles for zoning. (See Chapter 43.)

The finned tubing at floor must have sufficient capacity to offset effects of cold walls and windows during normal operation, and to heat the entire car to a minimum temperature of 60 F during standby when the overhead system is not operating. The maximum capacity required (determined by standby requirements) varies with car construction and design temperatures, but is approximately 90,000 Btu per hour. This requires a heating capacity in finned tube of approximately 650 Btu per linear foot.

The overhead air heating coil must have sufficient capacity to heat outside air brought into car for ventilation and to supply approximately 20 per cent of internal heat loss of the car so as to permit supply of floor heat at all times at an output that will not be objectionable to passengers sitting near it. The usual capacity of overhead heating coil is approximately 100,000 Btu based on 2400 cfm of circulated air with 600 cfm of this being outside air for ventilation. All Btu figures are approximations of actual heating requirements and do not include heat losses in trainline (or leakage) or losses in undercar piping. The heat losses in undercar piping can become a major portion of train heating boiler load on cars having many undercar loops and steam regulating devices.

Refrigeration

For cooling and dehumidification during summer, refrigeration may be obtained from ice bunkers, steam jet systems, mechanical compressors driven directly from car axle, by electric motors or by gas engines. Refrigeration required varies with load conditions but 7½ tons per car is one capacity frequently used. Evaporative type condensers are sometimes used in combination with the usual air condenser on either steam jet or mechanical refrigeration.

When an electric motor (approx. 10 hp) is used to drive the air conditioning compressor the electric power source is a problem. If power source is an axle-driven generator, there is an appreciable increase in the drag on locomotive, and power available for refrigeration, when train is stopped at a station, is limited to storage batteries. One solution to this problem is to use a d-c generator driven by a gas engine. Another is to use a Diesel-driven alternator in a special head-end car to furnish power to entire train. Recently there have been installations of a Diesel-driven alternator mounted on an individual passenger car to supply power requirements of car. The attractiveness of this type of installation can be increased by utilizing spare alternator capacity in winter for electric heating. If this capacity is supplemented by exhaust heat from the Diesel engine, there is sufficient capacity to heat entire car and provide hot water for wash rooms when outside temperatures are above approximately 30 F. This feature is important on trains using Diesel locomotives since it eliminates need for firing the steam heating boiler in locomotive during a portion of the year.

Air Distribution and Cleaning

Railway cars present critical problems in air distribution because air space per passenger is small (60 to 190 cu ft), and sun load is great. An average passenger car contains approximately 5000 cu ft of air and

may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but other constituents can be successfully handled only by proper ventilation and air cleansing. In an average car from 2000 to 2500 cfm are circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from outside. The amount of outside air desirable depends upon type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of total air circulated.

Careful attention must be exercised in specifying rate of outside air taken in so as to fit type of service adequately and yet not to supply more ventilation than is necessary. Conditioning this outside air is a major factor in determining size of both summer and winter equipment.

For normal conditions, 10 cfm of outside air per passenger are provided. When smoking is permitted, at least 15 cfm should be admitted. In some dining cars and deluxe sleeping cars, outside air rates as high as 20 to 30 cfm per occupant are used. A ceiling duct lengthwise along center of the car is usually used to distribute the air to interior by fans or blowers. A perforated ceiling supplied from an overhead duct, or delivery grilles and plaques designed to give considerable entrainment and mixing, are used to deliver air to car space.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near ceiling can be broken up by directing incoming air along ceiling at a velocity somewhat higher than that used for the rest of car. The air is exhausted through washroom or lavatory. For compartments, provision is made in door or partition for removal of used air. Lower berths are provided with a low velocity air outlet.

Recirculating air grilles are usually of straight flow type. Outside air intakes are usually located in vestibule, on side of car, or on roof, depending upon location of cooling coils. On many recently air-conditioned cars, there are no dampers or shutters at outside air intakes; the percentage of outside air is controlled by adjusting flow through the recirculating grille.

A considerable number of coach cars are now being equipped with return air ducts fitted in structure of baggage racks. Part of air circulated is returned to blower unit through these ducts and part through car body. This arrangement reduces quantity and velocity of air returning through car body and removes smoke fumes at the source. This, and any other design features aimed at taking recirculated air at floor and adjacent to both end doors (rather than drawing all recirculated air to one end of car) also reduces infiltration of cold air in ankle height strata, when doors are opened during heating season.

All air circulated by blower is filtered before passing over cooling and heating coils. In some cars outside and recirculated air are filtered separately before mixing, while in others air from the two sources is mixed before passing through a common filter. Filters in use are combinations of metal, wool, cloth, spun glass, hemp, paper, hair and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, re-treated, and re-used, while other types are discarded when dirty. Applications are also being made of electric precipitation for air cleaning. In this system coarser particles are removed from air by mechanical separation, finer materials by electrostatic action.

Activated carbon units sometimes are used in addition to the regular

filters for absorbing odors and other impurities, thus reducing the amount of outside air necessary for ventilation.

Temperature and Humidity Control

Controls in a passenger car should be as automatic as possible. The regular train crew cannot be relied on to make adjustments for comfort of passengers. For this reason latest systems of temperature control have only an off-on switch to be operated by train crew. When the system is in operation, heating or cooling is provided automatically as required.

When heating, it is important that floor-heat finned tubing be controlled at stable temperatures. Wide fluctuation in its temperature is highly objectionable because of location close to passengers. Stable operation may be secured by controlling floor heat on basis of outside conditions (See Chapter 43 on zoning), and using an overhead air circulating system to maintain final car temperatures.

Because of window condensation and other problems, no attempt is usually made to raise relative humidity in a railroad car in winter time.

When cooling, the steam jet refrigeration system is controlled in an on-off manner. Some means are ordinarily provided for operating mechanical compressor systems at partial capacity. In this case split evaporators are used, and evaporator surface and compressor are reduced together under light load conditions.

Under very light cooling loads, relative humidity in a car tends to increase because of long *off* periods of refrigeration equipment. This can be prevented by starting refrigeration equipment on low capacity at an established outdoor air temperature, operating it continuously and using a heating coil in overhead system to re-heat sufficiently to maintain car temperatures. Under higher load conditions the heating coil becomes inoperative and compressor and evaporator capacity are increased as needed.

Room type sleeper cars introduce a further problem of providing individual adjustment of room temperatures. Sometimes this individual control is secured by adjusting air volume, but such adjustment is unsatisfactory for overall comfort, and may affect air supply to other rooms. Another method is to use the heaters at the floor to control room temperature but this tends to cause unstable and improper floor heat temperatures which may be objectionable to passengers. A simpler and basically more satisfactory system is to use a small booster heater in individual overhead air supply ducts to each room under manual control of occupant. In this case, floor heat and basic overhead systems are automatically adjusted for varying load conditions just as in a simple coach type car. A fixed amount of heat determined by occupant can be added by the room booster heater to maintain desired individual room temperature. Temperature lag is reduced in case of room boosters over use of gravity floor heating surfaces. Any adjustment of booster will give occupant immediate change in space conditions. The use of floor heat surfaces for room control may also mean low or excessive surface temperatures close to passenger, with resulting discomfort.

STREETCAR AND TROLLEY COACH HEATING AND VENTILATING

Streetcars and trolley coaches present a special problem in maintenance of satisfactory comfort conditions due to frequent opening of doors and highly fluctuating passenger load. Space limitations for ducts and a desire

to keep outlet grilles well above floor to facilitate car cleaning, add further problems to distribution system. Maintaining comfort conditions at driver's station cannot be overlooked since his term of occupancy is considerably longer than that of any passenger and because he is usually dressed more lightly than passengers. A separate source of heat is usually provided for operator and is under his control.

Heating and Ventilating

Recently-built streetcars and trolley coaches use heat from air blown over main accelerating resistors and track switch resistors to heat passenger space. In the modern streamlined streetcars, designated P.C.C., approximately 2400 cfm are drawn from car and blown over these resistors to dissipate their heat. In trolley coaches, an amount of 800 cfm is customary. The heated air is then delivered to passenger space or diverted to outside by means of dampers as required. If available heat from this source is insufficient, auxiliary electric heaters in supply ducts may be cut in. The air distribution system is 100 per cent recirculating when the maximum heating requirement is being met. At conditions other than maximum heating demand, part of air drawn from car is exhausted to atmosphere. Outside air enters car body, under these conditions, by infiltration at all cracks and through doors when opened at stops.

The most recently-built streetcars have added ventilating fans in roof structure to introduce outside air through ceiling diffusing grilles. By governing volume of fresh air introduced through these roof fans in coordination with heated air distributing system, it is possible to maintain slight pressurization of passenger space and avoid inrush of air when doors are opened to load passengers. The roof fans also provide an effective means of maintaining lower inside temperature during summer operation. Ventilation tests on P.C.C. streetcars indicate that with 90 F outside temperature and above, 12,000 cfm is required to provide sufficient air change to keep inside temperature within a few degrees of outside air temperature and to provide enough air movement over passengers for comfort. The best results have been obtained by operating ventilating fans with windows closed. New cars provided with adequate ventilating capacity have been built with fixed sash.

Control

Temperature control consisting of equipment especially designed to withstand vibration present on transportation equipment is used. Automatically operated dampers are used to control flow of heated air to passenger space or to direct heated air to atmosphere. An automatically operated rheostat or multi-point switch is used to vary speed of ventilating fans. Recent control system applications employ one thermostat to operate both heating dampers and ventilating fans in a modulating or graduated manner with compensating thermostat in heat supply duct to correct for wide fluctuations in air temperature available from resistors. Ventilating fans are usually stopped or operated at lowest speed during heating cycle, and then speed is gradually increased as car temperature rises above heating cycle control point.

PASSENGER BUS AIR CONDITIONING

The passenger bus designed for urban transportation operation presents a greater problem to designer of heating system than does the inter-urban bus. More frequent stops, and rapidly changing passenger load create this problem on urban vehicles. Provision of heat for driver independent

of passenger heating system is a further problem. The inter-urban bus, however, is usually a deluxe vehicle and may require a comfort cooling system. Space and weight limitations and vibration must be considered.

Heating

Recent designs of bus heating systems include improved air distribution. Heat in engine coolant liquid is used to warm air by means of suitable finned coils and air is distributed throughout the passenger space by ducts and outlets directed toward floor. Some designs include finned surface near the floor similar to application of heating surface in railway passenger cars. Forced air circulation over this finned floor heating surface has been provided to increase its effectiveness. Oil burning booster heaters have been applied to many Diesel-powered buses to raise temperature of engine coolant for maximum engine operating efficiency and to provide sufficient heat for passenger space.

Ventilation

Air for ventilation is usually brought into a bus at front and distributed throughout length of passenger space by a duct or ducts near ceiling. Except for a few designs employing 100 per cent outside air for heating, no heating of ventilating air has been provided. One recently designed distribution system for an inter-urban bus provides for a fixed minimum of outside air, and is arranged to increase percentage of outside air to 100 per cent when heating or cooling load diminishes. The distribution ducts and diversion damper arrangement of this system make available two supply ducts and one return duct for heating and for cooling, with a change-over to all three ducts to supply air during intermediate ventilating cycle. This system permits utilization of atmospheric cooling and ventilation to greatest degree when it can be most economically employed in interval between heating and cooling demand.

Conventional throw-away type filters or renewable filters are used in intake air ducts for many vehicles. Electrostatic filters have been successfully used in some installations. The need for elimination of dirt is great, but the problem is complicated by space limitations and limited power.

Refrigeration

Summer conditioning systems for inter-urban vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using refrigerants are used, and are powered by water-cooled gasoline engines of approximately 14 hp.

Complete systems add from 800 to 1300 lb to weight of coach. Sometimes an auxiliary generator driven by the refrigeration system engine is used and serves to help charge bus battery, thereby offsetting power drain imposed by the ventilating blower. Belted reciprocating compressors and direct-driven V-type and rotary compressors are used, with engine speeds up to about 1800 rpm. Air-cooled condensers for this service require about 5000 cfm of outdoor air, and this is provided by either centrifugal or propeller type fans belted or direct-driven by air conditioning engine. Preventing noise and vibration from affecting passengers is of vital importance. Installations must be so made that quick daily servicing of engine is possible. In all cases fuel is obtained from main bus tanks, and in some the main engine cooling system cools the air conditioning engine.

Control

Automatic temperature control is receiving more attention in design of new vehicles. Some municipalities and states have enacted laws requiring that buses operated on their streets and roads be so equipped. The simplest control systems for heating of urban buses consist of a single thermostat to start and stop the blower of the heater unit. Improved heating systems employ a thermostat to control liquid flow to heater cores by means of modulating valves in combination with a means of stopping heating blower when no heat is required. Heating and ventilating control is accomplished by controlling volume of fresh air over and above minimum required in accordance with temperature in passenger space. This is accomplished through automatic modulating dampers in outside air intake or by varying speed of ventilating air blowers.

In a large proportion of inter-city buses equipped with mechanical refrigeration, a single thermostat is used to start and stop the cooling operation. This may be accomplished by automatically starting an engine-driven compressor on cooling demand or by engaging a clutch to drive compressor. Modulated or graduated control of engine-driven compressors may be accomplished by automatic regulation of engine throttle controlled from a thermostat in the passenger space. Complete control systems are available to coordinate operation of heating, ventilating and cooling equipment from a single thermostat with automatic change-over from heating to ventilating to cooling.

AUTOMOBILE SUMMER AIR CONDITIONING

Recently summer cooling has been applied to automobiles. The average present day automobile with little insulation, large, single glazed window areas, and high infiltration and exfiltration losses, requires about 15,000 Btu per hour of cooling capacity. One system utilizes a reciprocating compressor belted from the main engine fan shaft thus operating at varying speeds up to 3000 rpm. The resulting refrigeration capacity varies from about 6000 Btu per hour at idling speed to 24,000 Btu per hour at maximum car speed.

A dry air condenser is placed in front of the engine radiator, and the liquid and suction refrigerant lines run back under the car floor to the evaporator which is located in back of the rear seat. Conditioned air is delivered into the car just above the shelf near the back of the rear seat. A return grille is provided under the rear seat, and the recirculated air is filtered. Outdoor air is provided by infiltration. Power for the air circulating blowers is obtained from the car storage battery. Equipment of this nature increases the car weight approximately 200 lb.

AIRCRAFT AIR CONDITIONING

In the space of a few years, heating, cooling and ventilating of airplanes has progressed from comparatively simple systems to highly complex multi-purpose designs. The attendant control problem has become correspondingly complex. On older, non-pressurized planes, the heating system consisted either of a steam boiler and radiator or a single stage or double stage heat exchanger. On both types, cabin temperature was adjusted by positioning face and bypass dampers. While these were sometimes moved by an automatic modulating control, in the majority of cases they were positioned by one of ship's crew, with results which, while not satisfactory, were passable. As these planes cruised at less than 200 mph and normally operated at low altitudes, changes in outside air temperatures

were generally gradual enough so that manual readjustment of controls could maintain reasonably comfortable cabin conditions. Nearly all of these heating systems were marginal in respect to heat available and the main problem was lack of heat, rather than inadequate control.

Non-Pressurized Cabins

With the advent of the combustion type heater, and use of larger and faster planes, use of manual controls became impracticable. The combustion type heaters reach full rating in less than a minute after being turned on, and as they are rated at 100,000 Btu per hr and up, and since several heaters are generally used, it would take full time of one crew member to keep cabin temperature regulated. As ships of this type are not pressurized, the heating system is still comparatively simple.

In one type two 100,000 Btu heaters are placed in parallel positions and ram air from an external scoop is passed through the heaters and discharged through a series of distributing outlets located in the cabin ceiling. The cabin air is discharged through grilles located in the bottom walls of the cabin. An auxiliary nose heater is used by the crew to obtain additional heat for the cockpit or for windshield defrosting.

The cabin is maintained at desired temperature by means of an automatic control which operates both heaters simultaneously. This control consists of two duct thermostats, one being mounted in the air inlet duct between the air scoop and the heaters so that it is affected by outside ambient temperatures, and the other being mounted in the heater outlet duct so that it is affected by the discharge air temperatures. A thermostat in the cabin is so located that a continuous stream of cabin air passes through it. This type of control has been found to respond to a 1 deg temperature change in less than a second. Its theory of operation follows.

As the outside temperature starts to drop, the outside air duct thermostat decreases in resistance, unbalancing an electronic bridge. This unbalance is amplified by vacuum tubes and causes a power tube to close a relay, turning on the combustion heaters. The resulting increase in temperature is sensed by the warm air duct thermostat which increases in resistance, thus re-balancing bridge and causing relay to open. If there were no loss by radiation or convection from the aircraft cabin, these two duct thermostats would be sufficient for adequate control. However, the cabin thermostat is given approximately 30 times more influence than the duct thermostats and so acts as master controller, and the duct thermostats prevent over-heating or under-heating and keep discharge air from alternating between extreme cold and extreme heat.

In a slightly more elaborate system, two combustion heaters supply a plenum chamber which is maintained at a constant temperature. Air from the plenum chamber is then mixed with outside air to maintain desired cabin temperature. All of the warm air is discharged into cabin through walls. The discharge grilles are located on floor under seats, and a modulating type controller varies proportions of heated and outside air necessary to maintain desired cabin temperature. The same type of control system as previously described is used, except that an amplifier operates a two-phase motor capable of positioning control dampers instead of operating a relay which merely opens and closes fuel valve. An auxiliary duct, running from plenum chamber, is used by pilot as a source of windshield defrosting air.

Pressurized Cabins

With the advent of new high speed pressurized transport planes and addition of cabin cooling in addition to heating, the control problem becomes more complex. On all of these airplanes, the heat of compression from cabin supercharger must be controlled, air cycle or expansion turbines must be turned on and also, heat exchanger or combustion heaters, which are used in system when cooling is required for additional heat, must be automatically controlled.

Assuming that one of these airplanes is operating in an extremely cold climate, the sequence of operation would be as follows:

The automatic controller for supercharged air inter-cooler would be in full closed position, so that none of heat of compression would be removed, and air would bypass expansion turbine and its compressor and secondary after-cooler. An additional automatic controller would be operating the combustion heater and supplying additional heat necessary to maintain desired cabin temperature. If a heat exchanger were used as a supplemental source of heat, a modulating control operating a damper on this exchanger would run towards full heat position.

As airplane enters a warm climate and heat requirements drop, the combustion heater would cease operation or heat exchanger would go to full cold position, and modulating control on supercharge compressor would move towards cold position. When outside ambient temperature rises so high that cooling is desired, cabin supercharger intercooler would be opened wide. If further cooling were required, the air would go into an air cycle turbine, which is modulated to deliver required amount of cold air to maintain a comfortable cabin temperature.

Pressure in the cabin is maintained by providing a controlled, constant rate of air flow into the cabin sufficient to maintain ventilation and adjusting the cabin pressure relief valve setting by means of a cabin pressure selector to maintain desired cabin pressure. Limits on maximum inside to outside pressure may be of the order of 4 or 5 psi and safety controls should be provided to prevent exceeding this. There is a maximum rate at which cabin can change to a newly selected value, this rate being in some cases also adjustable.

The requirements for controls of this nature are extremely rigid. It is commonplace for ships of this type to experience changes in outside ambient temperatures of as much as 100 deg in space of 5 minutes. For this reason, speed of sensing a change and rapidity of response in the control system is essential if satisfactory control is to be accomplished. The older type thermostats cannot be used in airplanes, due to mass of thermostat and to vibration experienced on all airplanes. All modern controls use some type of bridge system with temperature sensitive resistors as sensing elements. In some types of controls, bridge system feeds a sensitive balanced relay, which in turn runs a modulating motor or controls an on-off power relay. A recent sensitive and quickly responding type uses an electronic amplifier, which in turn controls a two-phase motor, or, through relays, controls a d-c motor, or merely closes and opens a power relay for on-off applications.

In addition to extreme speed and accuracy which is required of all aircraft temperature controls, they must be able to operate under great extremes of temperature, pressure and humidity, and also withstand continuous extreme vibration. Heaters should have, in addition to control from thermostats, suitable limit controls to prevent over-heating due to failure of air supply or for any reason. Also, there should be safety devices to shut off fuel in case of flame failure.

On latest *high speed jet airplanes*, the temperature control problem is still more severe than on latest transports; as in addition to accuracy re-

quired, control response must be phenomenally fast. For example, on some the air going to the cabin from jet engine compressor can change temperature at rate of 150 deg per second. This, coupled with the fact that on smaller size pursuit ships air is changed in cabin as much as four times per minute, makes instantaneous sensing of change and extremely rapid control movement essential. Also, in airplanes operating at Mach numbers in excess of 0.7, the control must react to large adiabatic temperature rises encountered. Some of these problems are so new that controls still have not been developed which will meet all of desired conditions. However, present studies being made by control manufacturers should result in developments of such controls in the near future.

BIBLIOGRAPHY

Railway Passenger Cars

Report on Performance and Cost of Operation of 1937 Internal Combustion Engine Mechanical Compression Equipment for Air Conditioning Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, May 1, 1937.

Report on Relative Performance of Air Filters, by Mechanical Division, *Association of American Railroads*, January 15, 1938.

Air Conditioning of Railroad Passenger Cars, by L. W. Wallace and G. G. Early, Jr. (*A.S.M.E. Transactions*, November, 1937).

Passenger Car Cooling Methods, by Kenneth Cartwright (*Refrigerating Engineering*, February, 1936, p. 83 and March, 1936, p. 158).

Diesel Drive for Passenger Air Conditioning, by J. R. Hornaday (*Refrigerating Engineering*, March, 1942, p. 139).

Railroad Air Conditioning, by Gordon T. Wilson (*Refrigerating Engineering*, May, 1943, p. 323).

Railway Air Conditioning, by M. R. Eastin, August–December 1942, *Railway Elec. Engr.*

Head-End Power for Railway Cars, by F. L. Sahlmann and E. M. Bill, May 1939, *Railway Elec. Engr.*

Head-End Power for Streamlined Passenger Trains, by J. D. Loftis, October 26, 1946, *A.S.M.E. Raleigh Section.*

Buses and Automobiles

Bus Heating, by E. T. Todd and F. O. Gadd (*Heating and Ventilating*, Dec. 1946, p. 83).

Air Conditioning of Automobiles and Buses, by L. W. Child (*Society of Automotive Engineers Journal*, June, 1938).

Bus Air Conditioning, by Jerry Hicke (*Heating, Piping and Air Conditioning*, October, 1938, p. 639).

Bus Air Conditioning, by A. J. Mallinckrodt and Lars Hanson (*Refrigerating Engineering*, June, 1939, p. 388).

Problems in Air Conditioning Automobiles, by F. J. Linsenmeyer (*Society of Automotive Engineers Journal*, July, 1939).

P.C.C. Car Ventilation, B-3697 Westinghouse Electric Corp., September 1946.

Airplanes

Comfort in High Altitude Flying, by D. W. Tomlinson (A.S.H.V.E. Transactions, Vol. 47, 1941, p. 57).

Heat Exchangers for Aircraft, by Arthur J. Hess (*Refrigerating Engineering*, September, 1944, p. 192).

Heating and Ventilating for Transport Airplanes, by B. M. Brod (A.S.H.V.E. Transactions, Vol. 52, 1946).

Comfortization of Aircraft, by Albert A. Arnhym, (Pitman Publishing Corp., New York, 1945).

Refrigeration for Air Conditioning Pressurized Transport Aircraft, by B. L. Messinger, (*Heating and Ventilating*, January, 1946, p. 63).

CHAPTER 49

MARINE HEATING, VENTILATION, AIR CONDITIONING

General Considerations, Ship Construction Features, Factors Affecting Design, Ventilation Requirement for Various Types of Space, Air Conditioning Space Treatment, Typical Air Conditioning Systems, Types of Refrigerating Systems, Dehumidification, Ship Insulation

THE importance of adequate shipboard heating, ventilation, and air conditioning arrangements cannot be overemphasized. Installations must be designed to keep the passengers comfortable and the operating personnel physically and mentally fit. The provision of satisfactory living and working conditions is one of the most economical means of keeping a high morale.

GENERAL CONSIDERATIONS

A ship must be self-sufficient and provide for all human needs. Facilities must be limited to the minimum space and weight practicable to conserve dead weight and to increase the pay load. The pay load may be expressed in terms of cargo-carrying ability, passenger-carrying capacity, fighting strength, or towing ability.

Conditions in living spaces must permit adequate rest and comfort. Passenger accommodations must be treated to provide service and living conditions similar to those afforded by the various classes of hotels. Many of the expedients used ashore for this purpose are not applicable afloat. For instance, all living quarters aboard ship cannot be located at a distance from the power plant, but must often have boundaries in common with heat producing spaces. The thermal conductivity of shipbuilding materials such as steel, copper, brass, etc., is many times the value for building materials used ashore, and this, together with concentrated arrangement of equipment, produces a difficult heat transfer and insulation problem. Furthermore, the use of portholes, windows, skylights, and similar openings is greatly restricted in marine applications because of the necessity for strength, water-tightness and protection from the sea in foul weather. During wartime, the utility of portholes, windows, and skylights is greatly restricted because they must be fitted with light-excluding devices.

Mechanical ventilation is absolutely necessary for most shipboard spaces to enable the crew to operate the vessel, to prevent the accumulation of objectionable combustible and toxic gases, and to preserve stores and cargo. Shipboard equipment must also be reliable inasmuch as specialized servicing facilities are not available at sea and failure during an emergency may jeopardize the vessel's safety. Adequate spare parts form an integral part of all equipment furnished. Experience has demonstrated that *simple* and *foolproof* heating, ventilating, and air conditioning arrangements are essential for satisfactory service. The only variation between ship and shore applications is that emphasis is placed on different aspects of the design.

SHIP CONSTRUCTION FEATURES

The seaworthiness of a vessel is increased by subdividing the main hull space into a number of watertight compartments, so arranged that the vessel will not sink even when the hull is pierced in one or more places, by collision, torpedoing, or other causes. Every effort is made to avoid penetration of watertight subdivisions and, where openings must be cut, special closures are fitted.

The ship's interior is fitted with decks and flats (partial decks). The deck heights (floor heights) in cargo spaces generally vary from 9 to 30 ft and in living quarters, from 7½ to 9 ft. Usually in main machinery spaces, the entire depth is subdivided only by a few flats, with gratings provided where necessary to service equipment.

There is very little unassigned space on a ship, and many studies must be made to ascertain the arrangement of equipment which will best suit the conditions. Cuts in structural members must be limited in number and size. Lack of headroom and floor space often causes unsatisfactory drainage, inaccessibility, and other undesirable conditions. Air conditioning equipment must be located where it will interfere least with basic arrangements of major equipment and construction, and with ship stability.

FACTORS AFFECTING DESIGN

A ship is a self-contained structure capable of operating in a wide range of climates. The ventilation of most ships is designed on the basis of an 88 F outside dry-bulb temperature. Heating facilities on merchant vessels are invariably selected for an outside temperature of 0 F dry-bulb, although ships encountering only moderately cold weather in normal service are designed for higher outside temperatures. Designers of naval vessels commonly use 10 F outside dry-bulb temperature for estimating heating loads.

Cooling loads for most comfort systems are based on 95 F dry-bulb and 80 F wet-bulb temperatures. Inside conditions equivalent to from 73 to 75 effective temperature are used. An 81 F dry-bulb temperature and 50 per cent relative humidity are generally used for the design of comfort systems, except those serving public spaces having relatively large latent loads. These are designed for 80 F dry-bulb and approximately 55 per cent relative humidity. Bracket fans are supplied in many instances. Systems on naval vessels are designed for higher effective temperatures, and outside design temperatures of 88 F dry-bulb and 80 F wet-bulb are used.

Preliminary design is simplified by the uniformity of the conditions which apply to most ships. Among such conditions are (1) the necessity for the vessel to supply its own power, (2) the availability of an unlimited supply of cooling water, (3) the all-year-round supply of low cost steam at suitable pressure, (4) the drastic restriction of shipbuilding materials because of strength requirements, corrosion resistance, and fire hazard, and (5) the limitation of available space and permissible weight. These limitations require precise layouts, with minimum design safety factor, and unusual attention to design details and to means for balancing systems.

The relation between supply and exhaust ventilation depends on the type of space, and the relation of the space to the weather. Positive means for supply and exhaust must be provided. Ventilation through doors, windows, hatches, etc., should not be considered in the basic design.

It is usually most economical to heat spaces served by mechanical supply systems with steam duct heaters, in order to save weight and space.

TABLE 1. TYPICAL HEATING AND VENTILATION PRACTICES FOR SHIPS

Space	Type of Ventilation		Type of Heating	Air Change Minutes	Remarks
	Sup.	Exh.			
Living Quarters	Mech.	Mech. or Nat.	H.B.[a] or D.R.[b]	3-6	
Toilets and Showers	Nat.	Mech.	D.R.	3-4	
Chart House	Mech.	Mech. or Nat.	H.B or D.R.	5-6	
Office and Similar Spaces	Mech.	Mech or Nat.	H.B or D.R.	4-6	May be exhausted through Galleys and Pantries
Mess Rooms	Mech.	Mech. or Nat.	H.B. or D.R.	4-6	
Dining Salons	Mech.	Mech. or Nat.	H.B. or D.R.	3-6	May be exhausted through Galleys and Pantries.
Galley and Pantry	Mech. & Nat.	Mech.	T.A.[c]	½-1	Tempered Air Supply
Deck Pantries (No Cooking Equipment)	Mech. & Nat	Mech.	H.B. or D.R.	3-6	
Stores Spaces (Isolated Locations)	Mech. or Nat	Nat. or Mech.	....	15-30	
Hospital Spaces	Mech	Mech	H.B. or D.R.	3-4	
Soiled Linens and Oilskin Lockers	Nat	Mech	....	6-10	
Shops	Mech.	Mech or Nat	H.B or D.R.	4-10	
Group Berthing Spaces	Mech.	Mech or Nat	H.B or D.R.	3-6	
CO_2 Bottle Space	Nat	Mech	....	10	
Battery Room	Nat	Mech	D.R.	2	
Laundry	Mech	Mech	H.B	1-4	Tempered Air Supply
Dry Cargo Holds	Mech.	Nat or Mech	....	20-30	
Stearing Gear Space	Mech or Nat	Mech or Nat	H.B. or D.R.	2-6	Unit Heater if desired.
Gyro Room	Mech.	Nat. or Mech	....	3-6	
Theaters and Halls	Mech.	Nat. or Mech.	H.B. or D.R.	4-6	
Passageways and Stairwells	Nat	Nat. or Mech.	D.R.	....	Exhausting Adjacent Quarters.
Engine and Boiler Rooms	Mech.	Mech or Nat	....	1-2	
Misc. Machinery	Mech. or Nat.	Mech. or Nat	....	2-4	
Resistor Spaces	Nat.	Mech.	....	10	

[a] H. B.—Hot Blast System.
[b] D. R.—Direct Radiation.
[c] T. A.—Tempered Air.

Spaces which do not have mechanical supply or do not require ventilation in cold weather are heated by steam convectors, or unit heaters where the load is large. Ventilation air, except that supplied to auxiliary and main machinery spaces, is usually preheated (50 to 70 deg). In addition, where blast heating is used, zone reheaters are provided. No recirculation is used on combined ventilation and heating systems, but manual speed reduction (25 to 50 per cent) is provided during the heating cycle. All heaters are

automatically controlled, and preheaters are designed and installed to minimize the possibility of freezing of condensate.

The usual combinations or variations of duct type and convection or radiant heaters are used, depending on basic design requirements, such as weight and space limitations, and economic justification for refinements.

VENTILATION FOR VARIOUS TYPES OF SPACE

In the paragraphs which follow the most important considerations are given for the treatment of the various spaces, but the resulting quantities of air should, in all cases, be checked with those shown in Table 1.

Machinery Spaces

The prime purpose of machinery space ventilation is to maintain a habitable temperature for the operating personnel. Of secondary importance is the necessity for maintaining temperature conditions satisfactory for the successful operation of machinery. It is more practicable to use *spot cooling* of personnel at working areas than to attempt to obtain uniform ambient temperature. The permissible temperature rise at working stations is usually about 15 deg, while the over-all temperature rise is usually between 30 and 50 deg.

These spaces must be exhausted adequately, preferably by mechanical means. Every attempt should be made to remove air at or close to heat sources. The capacity of the mechanical exhaust systems should take into consideration the expansion of the supply air, and should be sufficient to insure an indraft through access openings to the space.

Normally, heat is not required for machinery spaces except for those fitted with electrically operated equipment, which may remain inactive during periods while in port. Such spaces should be equipped so as to be heated to about 50 F during such periods.

Living Spaces

The minimum quantity of air required for any space which is fitted for sleeping or office work, including hospital space, should be that which will limit the temperature rise over the outside air conditions to not more than 10 deg (a rise of 7 deg is more satisfactory if practicable), or a minimum of 30 cfm per person, whichever is the greater. For spaces fitted for eating, recreation, or manual work the rise may be taken at 10 deg with not less than 20 cfm per person. The same requirement applies to mechanical exhaust, although natural exhaust may be used where only a short run of duct exists.

Heat should be furnished to maintain the following temperatures:

Staterooms, Berthing, Messing and Office Spaces	70 F
Working Spaces and Shops	60 F
Hospital Spaces	75 to 78 F

Toilets, Washrooms, Showers and Baths

Spaces for these purposes should be fitted with mechanical exhaust ventilation for odor and steam removal. Generally, the surrounding living or working spaces are exhausted through them. Air requirements on merchant vessels are commonly estimated on the basis of a 4 min rate of change. Where the amount of available air is limited by outside air requirements of air conditioning systems, rates of change as high as one change in 6 min

are used for private bathrooms. On naval vessels, the minimum total quantity of exhaust is that which changes the air within the compartment in 6 min, plus 25 cfm for each toilet fixture, plus 50 cfm for each shower head.

Heat is obtained by use of convector radiators which should maintain a temperature of 70 F.

Galleys, Bakeries, Laundries, and Food Handling Spaces

The problem of the ventilation of spaces fitted for cooking and food preparation is primarily one of heat and smoke or fume removal. Complete mechanical exhaust is always provided, and the mechanical supply is made equivalent to at least 50 per cent of the exhaust. Sufficient natural supply to provide an indraft is required. The exhaust quantities should be predicated on restricting the ambient temperature rise to about 15 deg above the outside summer design air conditions. The resulting quantity will change the air in these spaces in about $^3/_8$ to 1 min. All of the exhaust should be arranged to remove air from the space through hoods fitted over the heat producing equipment. The mechanical supply should blow air directly on the personnel, but away from the equipment, so as not to interfere with the flow of exhaust air to the hoods.

The problem of ventilating laundries is somewhat similar to that for galleys. Experience indicates that the quantity of mechanical supply ventilation should be sufficient to change the air in the spaces in 1 to 4 min. The total exhaust (including tumbler dryers) should be equivalent to at least one air change per minute and at least equal to 120 per cent of the supply so as to insure an indraft of air. Generally, no heat is required for these spaces in winter, but preheaters are provided to temper the supply to between 45 and 60 F.

Storerooms and Cargo Spaces

The ventilation provided for these spaces depends on the type of vessel, location of space, and nature of cargo.

On merchant vessels ventilation is required for all closed spaces. Even if ventilation is not necessary to preserve the stores or cargo, it is required to prevent the accumulation of toxic or combustible gases and putrid odors. One air change in 15 to 30 min is common practice, except where inflammable liquids or proximity to hot spaces requires additional ventilation. Mechanical supply and natural exhaust are usually used. However, where inflammable gases may exist, natural supply and mechanical exhaust are provided.

It is the practice on naval vessels to omit ventilation where it is not necessary to preserve the stores, except for those spaces which hold inflammable liquids. The latter are provided with mechanical exhaust systems and natural supply. As spaces below the waterline are frequently damp, those which are unventilated may require chemical desiccants for removing moisture.

Refrigerated stores and cargo spaces requiring constant temperature and humidity control require special treatment.

AIR CONDITIONING SPACE TREATMENT

The extent of the application of air conditioning to new American passenger ships is fairly well established. All passenger staterooms, except steerage and third class, are usually air conditioned. This includes state-

rooms for ship's personnel and offices within conditioned passenger areas. Third class staterooms are air conditioned on some ships, depending on the particular trade. Theaters, lounges, smoking rooms, beauty shops, barber shops and similar closed public spaces are usually air conditioned.

All messrooms, recreation rooms, captain's and chief engineer's staterooms and offices, crew's inboard rooms and those having fixed portlights are usually air conditioned on passenger vessels. Other living spaces are air conditioned on some ships and only ventilated and heated on others.

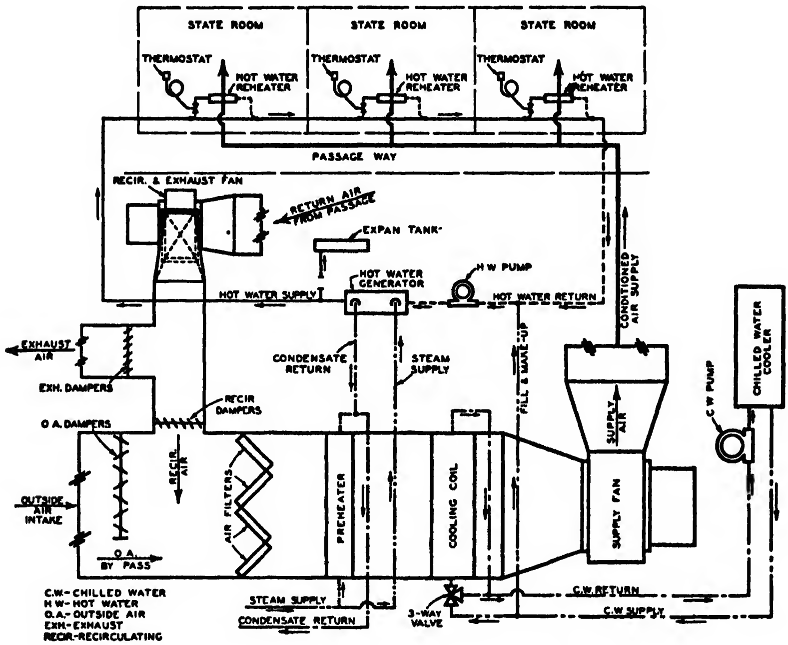


Fig. 1. Arrangement of Air Conditioning System with Central Supply Equipment and Individual Hot Water Room Reheaters

The treatment depends on the requirements of the operator and the proposed itinerary of the vessel.

TYPICAL COMFORT AIR CONDITIONING SYSTEMS

The various types of comfort conditioning systems used to date generally have followed conventional lines, except for those serving staterooms, offices, and similar small spaces. Large public spaces are fitted with individual systems which supply dehumidified and cooled air during the cooling cycle and warm air during the heating cycle. In many cases these rooms are fitted with large glass windows and doors, and require direct radiation to offset the downdraft which would occur in cold weather. Finned-tube radiation running the full length of the glass area is commonly used for this purpose.

Systems serving most public spaces are designed to provide all outside air as long as the refrigeration load is less than the capacity of the cooling equipment. Many central systems are simplified by using 100 per cent outside air all-year-round.

An important problem in ship air conditioning concerns the treatment of the small spaces such as passenger staterooms, offices, and crew quarters. Low headroom, congested quarters, double berths, and unsymmetric arrangements make each space a problem in air distribution and treatment.

The simplest system used for small spaces consists of a central filter bank, supply fan, preheater, and cooling and dehumidifying coil. Zone reheaters are provided to take care of variations in heating loads. Control of room temperature is achieved by manual dampers in the air supply to

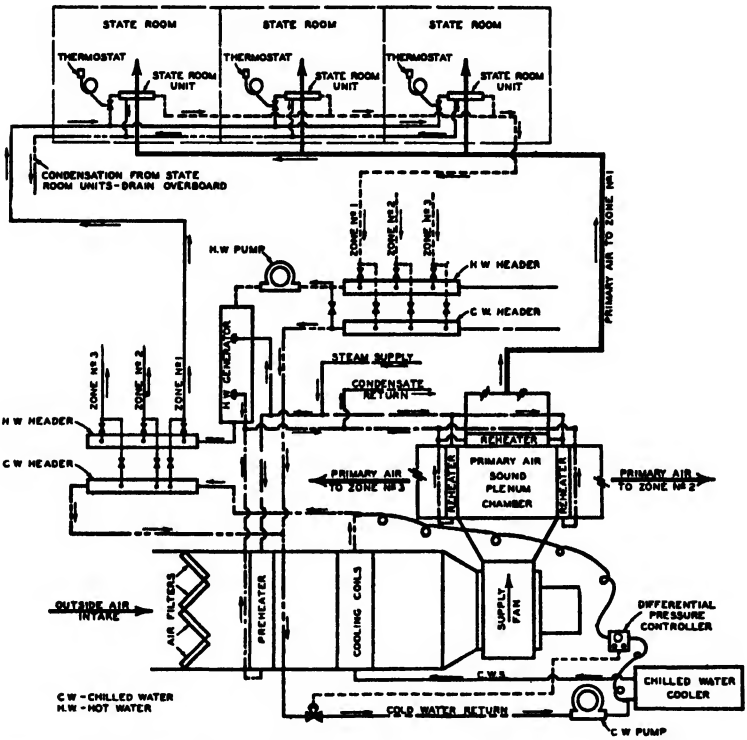


FIG. 2. ARRANGEMENT OF AIR CONDITIONING SYSTEM WITH CENTRAL SUPPLY EQUIPMENT AND INDIVIDUAL ROOM INDUCTION UNITS

the space. A recirculation-exhaust fan is provided, which operates in conjunction with automatic dampers to utilize the maximum quantity of outside air consistent with the capacity of the cooling coils. Manual speed reduction, without recirculation, is normally provided during the heating cycle. This system is used principally for third class, crew, and officers' accommodations. The average total air per person is about 45 cfm, and average outside air about 15 cfm.

A more popular system utilizes the same central supply equipment and recirculation-exhaust system as the one just described, except that the zone reheaters are omitted. Each space is provided with a small hot water reheater, automatically controlled by a room thermostat. Heat is provided through a forced circulation, single pipe, hot water system in most cases. Fig. 1 shows a diagrammatic arrangement of this system. It should be noted that reheat is used to eliminate overcooling of the indi-

vidual spaces during mild cooling conditions. This system is generally used for staterooms and small spaces devoted to first and second class passengers. The average total air per person is about 60 cfm and average outside air per person is about 18 cfm.

A third, less common, arrangement is a primary air system consisting of central filters, preheaters, cooling and dehumidifying coil, supply fan, and zone reheaters. In addition, each room is provided with an induction unit (floor type where possible) which reheats or recools the primary air supply. Automatic regulation of chilled or hot water flow, achieved by a room thermostat, controls the space temperature. The unit is provided with a drip pan and drain piping is required to remove condensation. Fig. 2 shows a diagram of this system, which is generally confined to passenger accommodations. The average primary (outside) air is about 35 cfm per person. No recirculation is used. The amount of induced air varies with the unit design.

Another system, which may be used to a limited extent, is similar to the third system, except that the room induction unit is used only to reheat the primary air, as all cooling and dehumidifying is done centrally. This simplifies the system and eliminates the unit drainage facilities. However, as no recirculation is used, the primary (outside) air requirements are somewhat larger, averaging about 40 cfm per person.

The duct work for all of the above systems may be designed for conventional velocities. However, if power is available, high velocity systems may be used providing suitably strong duct construction and adequate sound absorbing facilities are provided.

TYPES OF REFRIGERATION SYSTEMS

The kind of refrigeration equipment chosen for a particular vessel depends on the same factors which would govern selection on land.

Small tonnages use reciprocating or radial (Freon) compressors. Ships requiring in excess of approximately 125 tons of refrigeration generally use centrifugal compressors. Steam jet refrigeration has proven satisfactory on several large foreign liners and is being seriously considered in this country for similar applications. The two essential prerequisities of the steam jet—low-cost steam and condenser cooling water—are available. Also, the simplicity of principle and operation is a very desirable feature even for small installations.

Many ships have considerable refrigerated cargo space. The type of equipment used for this application should be kept in mind when selecting equipment to be used for air conditioning. This consideration can reduce materially the over-all cost, because a stand-by unit is always provided for cargo refrigeration. Also, most cargo systems are designed for cooling hot cargoes, and thus a large part of the capacity is available for comfort air conditioning most of the time.

DEHUMIDIFICATION

Merchant Vessels

Many merchant ships are fitted with dehumidification facilities for eliminating damage to the dry cargo by preventing condensation and dampness. The dehumidification load consists of moisture removed from the ventilation air passed through the dehumidifier, and also the moisture on, or given off by, the cargo, packaging, dunnage, battens, and other materials in the ship's holds. The most severe outside condition requires a

moisture removal of 90 grains (140-50) per pound of dry air, with 88 F cooling water. The largest cargo ships are provided with equipment for removing about 250 lb of water per hour.

The dehumidification systems generally utilize silica gel or lithium chloride with inhibitor (see Chapter 38). In most cases central drying equipment is provided, and a simple duct system distributes the dry air to the hold supply ventilation system. These ventilation systems use outside air when weather conditions are favorable. Recirculation and dehumidification are used only when necessary, *i.e.*, when the weather dew-point approaches or exceeds the temperature in the hold On large passenger ships consideration is given to the use of two dehumidifying units, because the cargo-carrying spaces are usually concentrated at the extreme ends of the vessel. With this arrangement, almost all of the dry air distributing system may be eliminated.

Naval Vessels

The end of the war terminated the current need for a great number of naval vessels, and presented the problem of their economical preservation and readiness for speedy return to active service. This problem has been solved by making the ship a warehouse for its own non-perishable stores and supplies, painting the exterior, applying polar type rust preventives (requiring no later removal) to corrodible exposed metal, and sealing and dehumidifying the interior. Such ships can be returned to service in 10 to 30 days.

The dehumidifiers developed for this purpose are dual-bed, electric heat reactivating types, and represent improvements for their special application over commercially available equipment with respect to compactness, weight, reliability, performance, and economy. On most ships dry air from the machine is delivered to the remote portions of the vessel through the fire mains, which provide the most extensive piping system available. The air returns to the machine via accesses and passageways. Humidity is maintained constant at a prescribed value, usually 30 per cent, by a recorder-controller developed especially for the preservation program. This device continuously checks, records, and averages humidity readings of 8 *sensing* stations, located throughout the vessel, and starts or stops the dehumidifier accordingly.

SHIP INSULATION

In order to limit one of the major ventilation heat loads, which is that made necessary by heat transmission, and to prevent condensation, it is necessary to use insulation judiciously. The principal sources of heat in a ship are the power plant and sun load. The confinement or exclusion of this heat in the structure of a ship is not easy, principally because of the complex structural nature of the beams, stiffeners, bulkheads, decks, and hull. The continuous metal paths offer easy means of heat flow throughout the structure.

Insulation cannot be used indiscriminately because of weight and space limitations. Therefore, higher heat transmission coefficients are accepted for insulated structures of certain classes of vessels, than would be considered satisfactory ashore. Particular care must be taken to prevent metal-to-metal contact, which greatly reduces the effectiveness of the insulation.

Hull insulation must be of high quality with particular attention given to fireproofness, low density, low thermal conductivity, ruggedness, vermin

resistant ability, and ease of application. Board or blanket types are most common.

Duct insulation must have the same characteristics as hull insulation. Semi-rigid board is most common, and the blanket type is used only for curved surfaces. Corrugated asbestos is not used because the presence of moisture tends to disintegrate it

Preheaters are frequently located close to the outside air intake in order to conserve insulation, and for the same reason zone reheaters are located as close as possible to the zone Where a reheater serves only one space, the heater is commonly located in the space.

BIBLIOGRAPHY

The Ventilation of Ships, by R. McDonald (*Journal of the Institution of Heating and Ventilating Engineers* (British) October, 1939).

Ventilation and Air Conditioning of the S. S. Panama (*Heating and Ventilating*, September, 1939, p. 47).

Air Conditioning the New Mauretania (*Heating, Piping and Air Conditioning*, July, 1939, p. 431).

Heating, Ventilating and Air Conditioning on Shipboard, by J. H. Clarke (*Heating, Piping and Air Conditioning*, August, p. 467; September, p. 529; October, p. 610, 1940).

Care of Cargo at Sea, by O. D. Colvin, W. H. E. Hahne, and M. R. Colby (*Transactions of the Society of Naval Architects and Marine Engineers;* Part I, Vol. 46, 1938, p. 109; Part II, Vol. 49, 1941, p. 208).

Ventilation of Ships, by J. Dawson (*Journal of The Institution of Heating and Ventilating Engineers*, London, Vol. 10, June-July, 1942, No. III, p. 89).

Modern Marine Engineers Manual, Vol. II, Sections 16 to 19, Incl. (Cornell Maritime Press, 1943).

Warship Ventilating, Heating and Air Conditioning, by Comdr. T. H. Urdahl, U.S.N.R., and W. C. Whittlesey (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 35).

Standardized Heating and Ventilating Equipment for Fighting Ships, by Comdr. T. H. Urdahl, U.S.N.R., and Lt. John Everetts, Jr., U.S.N.R. (*Heating, Piping and Air Conditioning*, July, 1943, p. 333).

Designing Warship Ventilation with Standardized Equipment, by Comdr. T. H. Urdahl, U.S.N.R., and W. C. Whittlesey (*Heating, Piping and Air Conditioning*, August, 1943, p. 419).

Ventilation and Heating of Maritime Commission Ships, by J. W. Markert (*Heating and Ventilating*, February, 1943, p. 32).

Why Ship Ventilation, by J. W. Markert (*Heating and Ventilating*, October, 1943).

Evolution of Ships Ventilation Systems, by J. W. Markert (*The Log*, 1944 Yearbook, p. 202).

Refrigeration, Air Conditioning, Ventilation and Heating, by H. E. Parker (*Marine Engineering*, Vol. II, *The Society of Naval Architects and Marine Engineers*, 1944, Chapter VI, p. 319).

Blackout at Sea, by J. W. Markert (*Heating and Ventilating*, March, 1944, p. 55).

Your Merchant Marine Will be Comfort Air Conditioned, by J. W. Markert (*Heating and Ventilating*, May, 1945, p. 60).

Modern Air Conditioning, by J. W. Markert (*Marine Engineering and Shipping Review*, November, 1945, p. 177).

Modern Marine Refrigeration and Air Conditioning, by W. H. Carrier and L. E. Star (*Marine Engineering and Shipping Review*, April, 1946, p. 132).

Dehumidification Protects U. S. Navy's Inactive Fleet, by Capt. T. H. Urdahl and Comdr. E. R. Queer (*Heating, Piping and Air Conditioning*, March, 1946, p. 71).

CHAPTER 50

WATER SERVICES

Estimate of Demand Load, Estimate of Pipe Capacity, Sizing of Cold Water Supply Piping, Hot Water Supply Piping, Storage Capacity and Heating Load, Methods of Heating Water, Computing Area of Heat Transmitting Surface, Control of Service Water Temperature, Solar Water Heaters

PROPER design of the water distributing system in a building is necessary in order that the various fixtures may adequately perform their function. The amount of either hot or cold water used in any building is variable, depending on the type of structure, usage, occupancy, and time of day. It is necessary to provide piping, water heating, and storage facilities of sufficient capacity to meet the peak demand without wasteful excess in either piping or equivalent cost. Most of the data on the sizing of the water supply pipes have been taken from various reports of the National Bureau of Standards[1].

ESTIMATE OF DEMAND LOAD

The demand load in building water supply systems is based on the number and kind of fixtures installed and on the probable simultaneous use of these fixtures. The essentials for making these estimates, consist principally of Table 1 giving demand weights in terms of fixture units for different plumbing fixtures under different conditions of service and Fig. 1 giving estimated demand in gallons per minute corresponding to any total number of fixture units. Fig. 2 shows an enlargement of Fig. 1 for a range up to 250 fixture units.

The estimated demand load for fixtures used intermittently on any supply pipe will be obtained by multiplying the number of each kind of fixture supplied through that pipe by its weight from Table 1, adding the products, and then referring to the appropriate curve of Figs. 1 or 2 with this sum. In using this method it should be noted that the demand for fixture or supply outlets other than those listed in the table of fixture units is not yet included in the estimate. The demands for outlets—such as hose connections, air conditioning apparatus, etc.—which are likely to impose continuous demand during times of heavy use of the weighted fixtures should be estimated separately and added to the demand for fixtures used intermittently, in order to estimate the total demand.

ESTIMATE OF PIPE CAPACITY

The size of iron pipe required to carry a given rate of flow with a given pressure drop may be determined from Fig. 3 for *Fairly Rough*, or Fig. 4 for *Rough* Pipe, respectively[2].

Ferrous pipe, after a few years of water supply service even with the best of waters in respect to corrosion and caking, will have become sufficiently roughened so that it may be considered as fairly rough pipe and, hence, Fig. 3 may be used to estimate the capacity of ferrous pipe which will carry such water. With badly corrosive or badly caking water, the flow chart in Fig. 4 should be used to estimate capacities of ferrous pipe and no pipe should be smaller than ¾ inch for this type of service.

Table 1. Demand Weights of Fixtures in Fixture Units[a]

Fixture or Group[b]	Occupancy	Type of Supply Control	Weight in Fixture Units[c]
Water closet	Public	Flush valve	10
Water closet	Public	Flush tank	5
Pedestal urinal	Public	Flush valve	10
Stall or wall urinal	Public	Flush valve	5
Stall or wall urinal	Public	Flush tank	3
Lavatory	Public	Faucet	2
Bathtub	Public	Faucet	4
Shower head	Public	Mixing valve	4
Service sink	Office, etc.	Faucet	3
Kitchen sink	Hotel or restaurant	Faucet	4
Water closet	Private	Flush valve	6
Water closet	Private	Flush tank	3
Lavatory	Private	Faucet	1
Bathtub	Private	Faucet	2
Shower head	Private	Mixing valve	2
Bathroom group	Private	Flush valve for closet	8
Bathroom group	Private	Flush tank for closet	6
Separate shower	Private	Mixing valve	2
Kitchen sink	Private	Faucet	2
Laundry trays (1-3)	Private	Faucet	3
Combination fixture	Private	Faucet	3

[a] For supply outlets likely to impose continuous demands, estimate continuous supply separately and add to total demand for fixtures.
[b] For fixtures not listed, weights may be assumed by comparing the fixture to a listed one using water in similar quantities and at similar rates.
[c] The given weights are for total demand. For fixtures with both hot and cold water supplies, the weights for maximum separate demands may be taken as ¾ the listed demand for the supply.

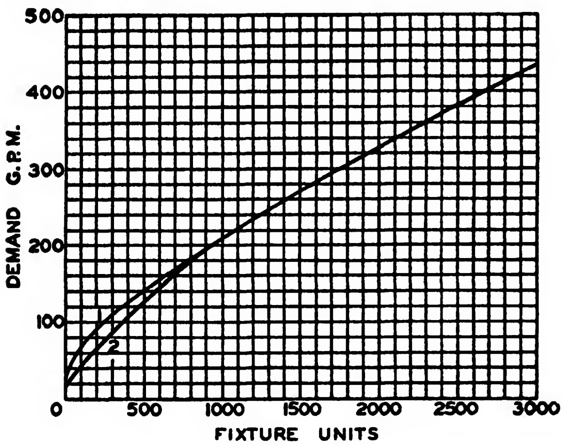


Fig 1. Estimate Curves for Demand Load

No. 1 for system predominantly for flush valves.

No. 2 for system predominantly for flush tanks.

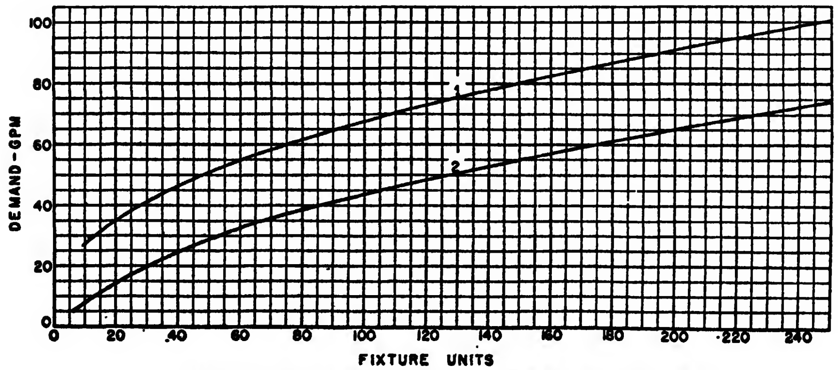


Fig. 2. Section of Fig. 1 on Enlarged Scale

TABLE 2. PERFORMANCE REQUIREMENTS OF WATER METERS[a]

SIZE IN.	NORMAL TEST-FLOW LIMITS GPM	MINIMUM TEST-FLOW GPM
⅝	1 to 20	¼
¾	2 to 34	½
1	3 to 53	¾
1½	5 to 100	1½
2	8 to 160	2
3	16 to 315	4
4	28 to 500	7
6	48 to 1,000	12

[a] *American Water Works Association Standards:*
Registration. The registration on the meter dial shall indicate the quantity recorded to be not less than 98 per cent nor more than 102 per cent of the water actually passed through the meter while it is being tested at rates of flow within the specified limits herein under normal test flow limits: There shall be not less than 90 per cent of the actual flow recorded when a test is made at the rate of flow set forth under minimum test flow.

As copper tubing is not likely to retain interior incrustation to any serious extent, Fig. 5 may be used safely to estimate carrying capacity of copper tubing[2].

SIZING COLD WATER SUPPLY PIPING

In order to apply the foregoing principles to the sizing of the supply pipes in the usual up-feed system, it is necessary to ascertain the minimum pressure in the street main, and from this should be subtracted the minimum pressure required for the operation of the topmost fixture. (A pressure of 15 psi is ample for flush valves but reference should be made to the manufacturers' requirements. A minimum of 8 psi should be allowed for other fixtures.) The pressure differential thus obtained will be available for overcoming pressure losses in the distributing system and in overcoming the difference in elevation between the water main and the highest fixture.

The pressure losses in the distributing system will consist of the pressure losses in the piping itself, the pressure losses in the pipe fittings, and the pressure losses in the water meter, if any. Estimated pressure losses for disc-type meters for various rates of flow are given in Fig. 6. Flow limits

TABLE 3. ALLOWANCE IN EQUIVALENT LENGTH OF PIPE FOR FRICTION LOSS IN VALVES AND THREADED FITTINGS

DIAMETER OF FITTING IN.	EQUIVALENT LENGTH OF PIPE FOR VARIOUS FITTINGS						
	90 Deg Standard Ell Ft	45 Deg Standard Ell Ft	90 Deg Side Tee Ft	Coupling or Straight Run of Tee Ft	Gate Valve Ft	Globe Valve Ft	Angle Valve Ft
⅜	1	0.6	1.5	0.3	0.2	8	4
½	2	1.2	3	0.6	0.4	15	8
¾	2.5	1.5	4	0.8	0.5	20	12
1	3	1.8	5	0.9	0.6	25	15
1¼	4	2.4	6	1.2	0.8	35	18
1½	5	3	7	1.5	1.0	45	22
2	7	4	10	2	1.3	55	28
2½	8	5	12	2.5	1.6	65	34
3	10	6	15	3	2	80	40
3½	12	7	18	3.6	2.4	100	50
4	14	8	21	4.0	2.7	125	55
5	17	10	25	5	3.3	140	70
6	20	12	30	6	4	165	80

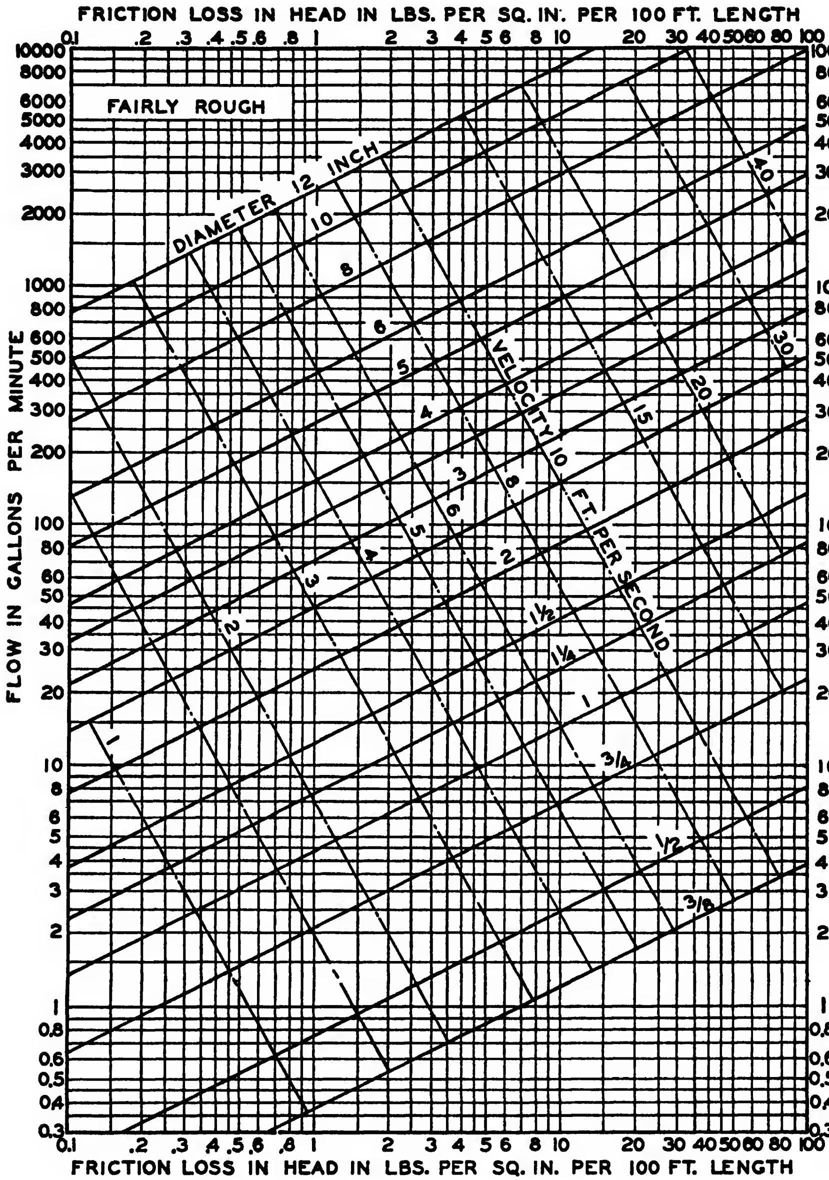


FIG. 3. FLOW CHART FOR FAIRLY ROUGH PIPE

for disc-type meters, which may be regarded as the limits of recommended ranges in capacities, are given in Table 2. For information on other types of meters the manufacturers should be consulted.

Estimated pressure losses for pipe fittings and valves in terms of equivalent pipe length are shown in Table 3.

The pressure loss, in pounds per square inch, caused by the difference in elevation between the street main and the highest fixture may be obtained by multiplying the difference in elevation in feet by 0.43.

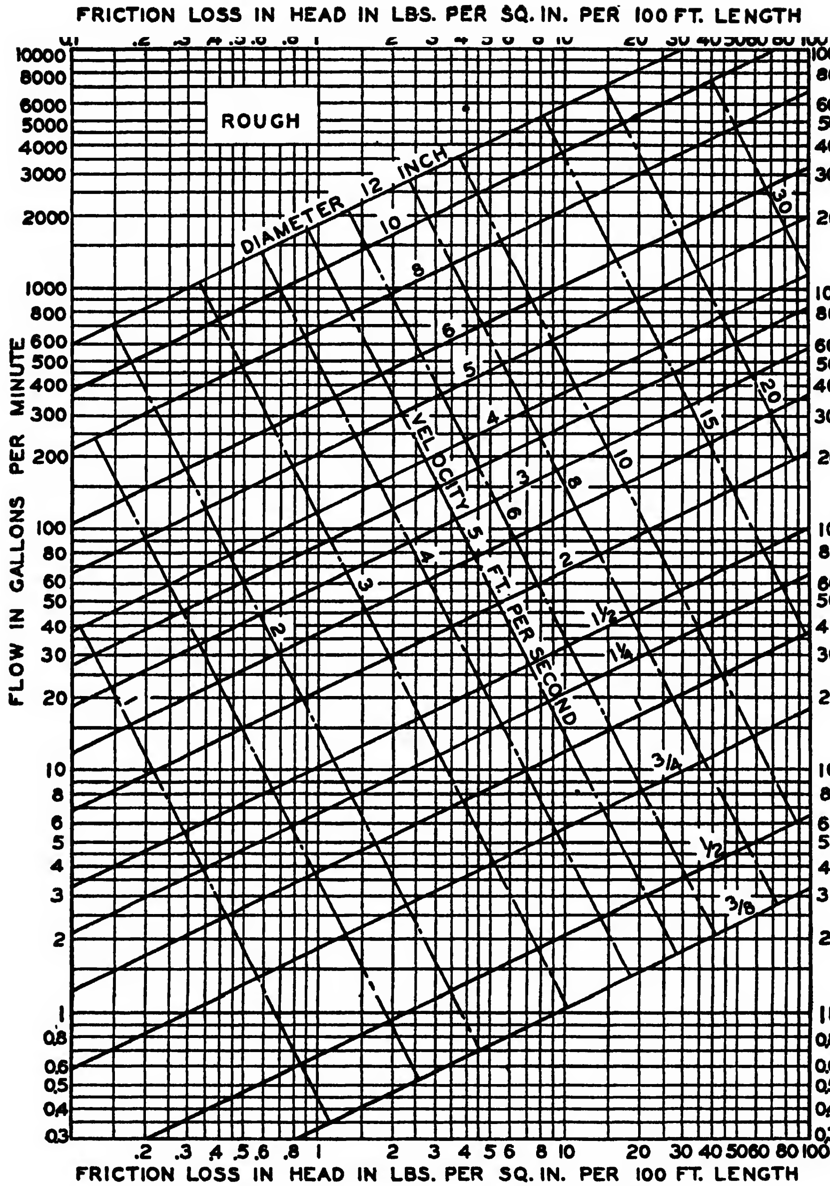


Fig. 4. Flow Chart for Rough Pipe

Example 1. Assume a minimum street main pressure of 55 psi; a height of topmost fixture above street main of 50 ft; a developed pipe length from water main to highest fixture of 100 ft; a total load on the system 50 fixture units; and that the water closets are flush-valve operated. Find the required size of supply main.

From Fig. 2 the estimated peak demand is found to be 51 gpm. From Table 2 it is evident that several sizes of meters would adequately measure this flow. For a trial computation choose the 1½ in. meter. From Fig. 6 the pressure drop through a 1½ in. disc-type meter for a flow of 51 gpm is found to be 6.5 psi.

Then the pressure drop available for overcoming friction in pipes and fittings is $55 - (15 + 50 \times 0.43 + 6.5) = 12$ psi.

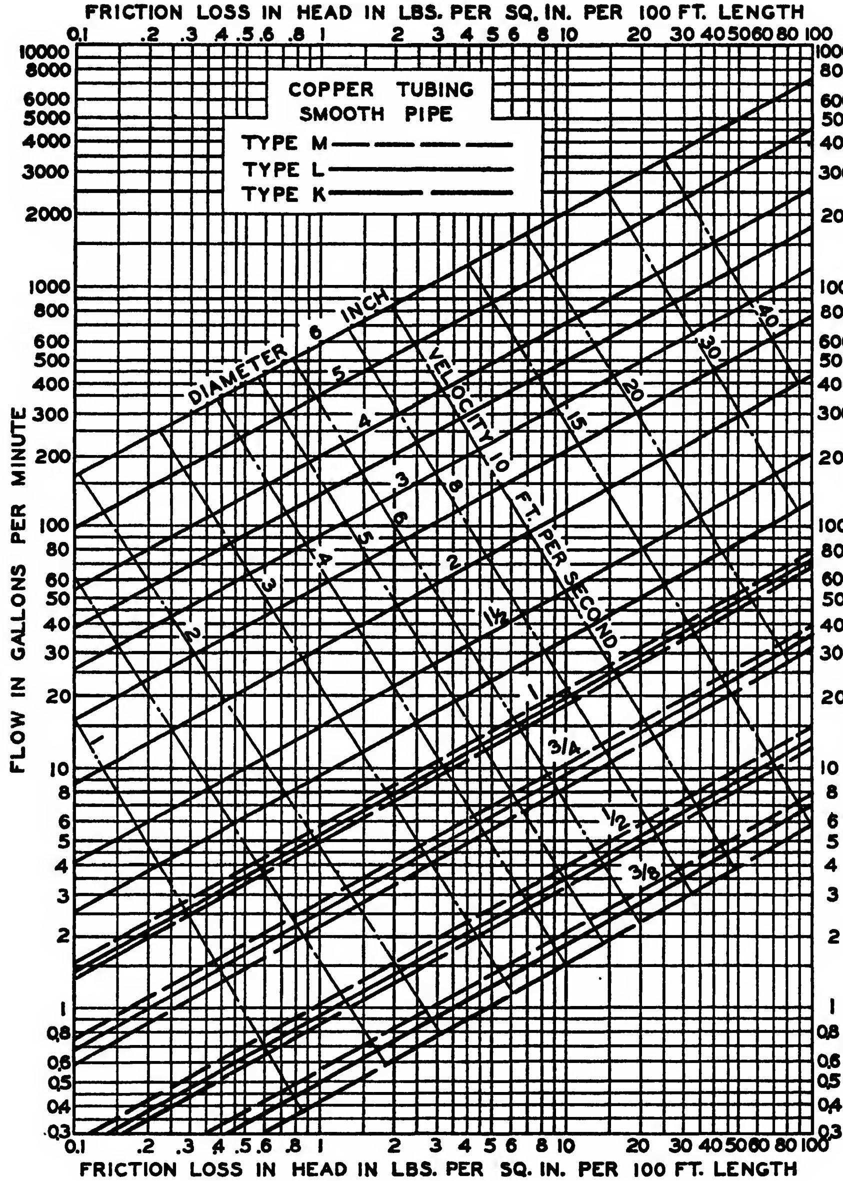


FIG. 5. FLOW CHART FOR COPPER TUBING

At this point it is necessary to make some estimate of the equivalent pipe length of the fittings on the direct line from the street main to the highest fixture. The exact equivalent length of the various fittings cannot now be determined since the pipe sizes of the building main, riser, and branch leading to the highest fixture are not known as yet, but a first approximation is necessary in order to make a tentative selection of pipe sizes. If the computed pipe sizes differ from those used in determining the equivalent length of pipe fittings, a recalculation will be necessary, using the computed pipe sizes for the fittings. For the purposes of this example assume that the total equivalent length of the pipe fittings is 50 ft.

Then the permissible pressure loss per 100 ft of equivalent pipe is $12 \times 100/(100 + 50) = 8$ psi.

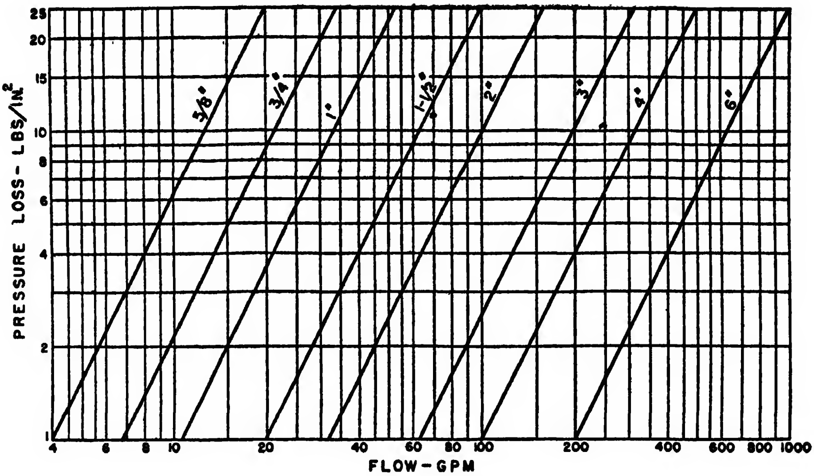


FIG. 6. PRESSURE LOSSES IN WATER METERS

Assuming that the corrosive and caking properties of the water are such that Fig. 3 for fairly rough pipe is applicable, a 2 in. building main is found adequate.

The sizing of the *branches* of the building main, the *risers*, and *fixture branches* follow the principles outlined. For example, assume that one of the branches of the building main carries the cold water supply for 3 water closets, 2 bath tubs, and 3 lavatories. Using the permissible pressure loss of 8 psi per 100 ft, the size of branch determined from Table 1 and Figs. 1 and 3 is found to be 1½ in. Items entering the computation of pipe size are given in Table 4.

The principles involved in sizing either up-feed or down-feed systems are the same. The principal difference in procedure is that in the down-feed system the difference in elevation between the house tank and the fixtures provides the pressure required to overcome pipe friction.

The water demand for hose bibbs or other large demand fixtures taken off the building main is frequently the cause of inadequate water supply to the upper floor of a building. This robbing of upper floor fixtures of water may be prevented by sizing the distribution system so that the pressure drops from the street main to all fixtures are the same. It is good practice to maintain the building main of ample size (not less than 1 in. where possible) until all branches to hose bibbs have been connected. Where the street main pressure is excessive and a pressure reducing valve is used to prevent water hammer or excessive pressure at the fixtures, it is frequently desirable to connect hose bibbs ahead of the reducing valve.

TABLE 4. COMPUTATION OF BRANCH SIZE IN EXAMPLE 1

FIXTURES NO. AND KIND	FIXTURE UNITS (FROM TABLE 1 AND NOTE c)	DEMAND (FROM FIG. 2) GPM	PIPE SIZE (FROM FIG. 3) IN.
3 flush valves	3 x 6 = 18		
2 bath tubs	¾ (2 x 2) = 3		
3 lavatories	¾ (3 x 1) = 2.25		
Total	23.25	38	1½

HOT WATER SUPPLY PIPING

It is common practice to provide circulating piping in all hot water supply systems in which it is desirable to have hot water available continuously at the fixtures. In average sized and small residences and systems, in which the piping from the heater to the fixtures is short, return circulating piping is generally omitted in order to reduce installation cost and to reduce heat loss from the piping, particularly during periods of no water demand.

The hot water supply may be distributed by either an up-feed or down-feed piping system. Three common methods of arranging the circulating lines are shown in Fig. 7. Although the diagrams apply to multi-story buildings the arrangements (*a*) and (*b*) are sometimes used in residential designs.

A check valve should be provided in the run-out from each return riser to prevent temporary reversal of flow in the piping when a faucet is open.

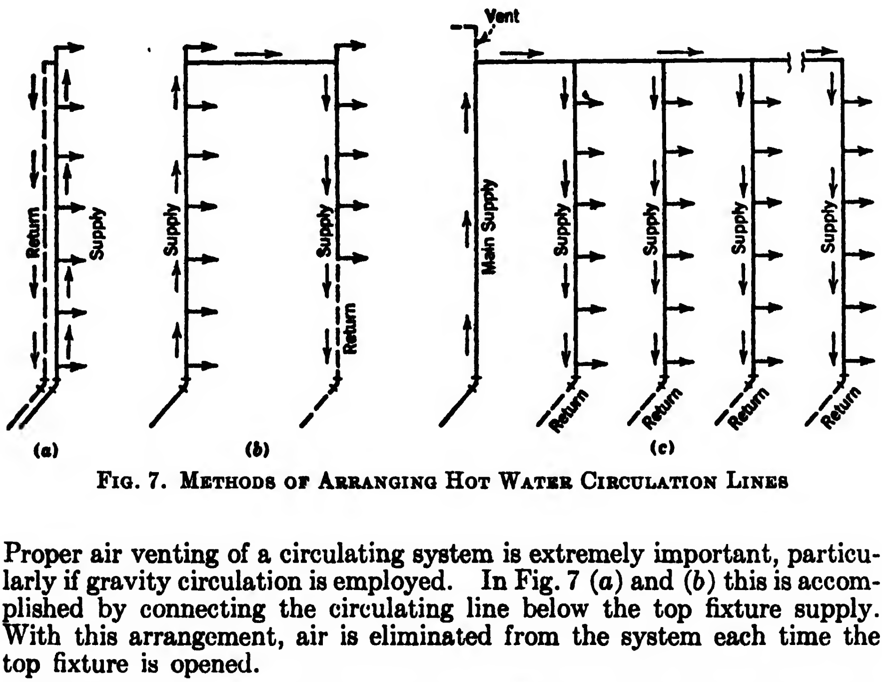


Fig. 7. Methods of Arranging Hot Water Circulation Lines

Proper air venting of a circulating system is extremely important, particularly if gravity circulation is employed. In Fig. 7 (*a*) and (*b*) this is accomplished by connecting the circulating line below the top fixture supply. With this arrangement, air is eliminated from the system each time the top fixture is opened.

Where an overhead supply main is located above the highest fixture as in Fig. 7 (*c*), an automatic float type air vent is installed at the highest point of the system or a fixture branch is connected to the top of the main where air venting is desired and then dropped to the fixture outlet.

It is sometimes necessary to make an allowance for pressure drop through the heater when sizing hot water lines, particularly where instantaneous hot water heaters are used and the available pressure is low.

The principles involved in the sizing of the hot water supply pipes are the same as those for the sizing of cold water supply lines. For small and medium sized installations a ¾-in. hot water return will be ample. For larger installations the size of the hot water return may be computed from considerations of the heat losses in the hot water piping[3]. A throttling

valve should be placed in the hot water return pipe so that the rate of circulation may be adjusted.

Where the hot water piping system is exceedingly long, a water circulator is frequently installed and controlled from an immersion thermostat, in the return line, set to start and stop the pump over approximately a 20 F deg temperature range.

STORAGE CAPACITY AND HEATING LOAD

In estimating the size of hot water storage tank required and the heating capacity to be provided either from the boiler or from an independent domestic hot water heater, it is necessary to know the total quantity of water to be heated per day, and the maximum amount which will be used in any one hour, as well as the duration of the peak load.

In cases where the requirements for hot water are reasonably uniform, as in residences, apartment buildings, hotels, and the like, smaller storage

TABLE 5. ESTIMATED HOT WATER DEMAND PER PERSON FOR VARIOUS TYPES OF BUILDINGS

Type of Building	Hot Water Required at 140 F	Max. Hourly Demand in Relation to Day's Use	Duration of Peak Load Hours	Storage Capacity in Relation to Day's Use	Heating Capacity in Relation to Day's Use
Residences, apartments, hotels, etc.	40 gal per person per day	1/7	4	1/5	1/7
Office buildings	2 gal per person per day	1/5	2	1/5	1/6
Factory buildings	5 gal per person per day	1/3	1	2/5	1/8
Restaurants \$0.50 meals \$1.00 meals \$1.50 meals	1.5 gal per meal 2.5 gal per meal 4.5 gal per meal			1/10	1/10
Restaurants 3 meals per day		1/10	8	1/5	1/10
Restaurants 1 meal per day		1/5	2	2/5	1/6

capacity is required than in the case of factories, schools, office buildings, etc., where practically the entire day's usage of hot water occurs during a very short period. Correspondingly, the heating capacity must be proportionately greater with uniform usage of hot water than with intermittent usage where there may be several hours between peak demands during which the water in the storage tank can be brought up to temperature. As a general rule it is desirable to have a large storage capacity in order that the heating capacity, and consequently the size of the heater, or the load on the heating boiler, may be as small as possible.

In estimating the hot water which can be drawn from a storage tank it should be borne in mind that only about 75 per cent of the volume of the tank is available, as by the time this quantity has been drawn off the incoming cold water has cooled the remainder down to a point where it can no longer be considered hot water.

Where steam from the heating boiler is used to heat domestic hot water, the computed load on the boiler should be increased by 4 sq ft EDR (equivalent direct radiation) for every gallon of water per hour heated through a 100 F rise. The actual requirement is $(100 \times 8.33)/240 = 3.48$ sq ft per

gallon of water heated 100 F. The value of 4 allows for transmission losses, etc.

There are two ways in common use of estimating the hot water requirements of a building; first, by the number of people and second, by the number of plumbing fixtures installed. Where the number of people to be served or can be reasonably estimated, the data in Table 5 may be used.

Example 2. From Table 5, a residence housing five people would have a daily requirement of 5 × 40 = 200 gal per day, and a maximum hourly demand of 200 × 1/7 = 28.5 gal. The heater should have a storage capacity of 200 × 1/5 = 40 gal and a heating capacity of 200 × 1/7 = 28.5 gal per hour.

The conditions given in Example 2 may be cited as average. It is possible to vary the storage and heating capacity by increasing and decreasing one over the other. Such a condition is illustrated in Example 3.

Example 3. Assume an apartment house housing 200 people. From the data in Table 5: Daily requirements = 200 × 40 = 8000 gal. Maximum hours demand = 8000 × 1/7 = 1140 gal. Duration of peak load = 4 hr. Water required for 4-hr peak = 4 × 1140 = 4560.

If a 1000 gal storage tank is used, hot water available from the tank = 1000 × 0.75 = 750. Water to be heated in 4 hr = 4560 − 750 = 3710 gal. Heating capacity per hour = 3710/4 = 930 gal.

If instead of a 1000 gal tank, a 2500 gal tank had been installed, the required heating capacity per hour would be $\frac{4560 - (2500 \times 0.75)}{4} = 671$ gal.

Table 6 may be used to determine the size of water heating equipment from the number of fixtures. To obtain the *probable maximum demand* multiply the total quantity for the fixtures by the Demand Factor in line 11. The heater or coil should have a water heating capacity equal to this *probable maximum demand.* The storage tank should have a capacity equal to the *probable maximum demand* multiplied by the storage capacity factor in line 12. Example 4 will illustrate the procedure.

Example 4. Determination of heater and storage tank size for an apartment building from number of fixtures.

60 lavatories	× 2 =	120 gal per hour
30 bath tubs	× 20 =	600 gal per hour
30 showers	× 75 =	2250 gal per hour
60 kitchen sinks	× 10 =	600 gal per hour
15 laundry tubs	× 20 =	300 gal per hour
Possible maximum demand		= 3870 gal per hour
Probable maximum demand	= 3870 × 0.30	= 1161 gal per hour
Heater or coil capacity		= 1161 gal per hour
Storage tank capacity	= 1161 × 1.25	= 1450 gal

METHODS OF HEATING WATER

Hot water may be heated either by the direct combustion of fuel, by an intermediate carrier such as steam or hot water, or by electrically heated surfaces. The simplest method is to have the fire on one side of a metal barrier and water on the other. In such a method if the water surfaces of heat transfer are small, and if the water carries a heavy proportion of precipitable salts, the water passages may soon clog and then burn out. A familiar example of such trouble is the water back of the firebox in the kitchen stove or the pipe coil inserted into the firebox of a warm air furnace

or small boiler. The critical water temperature, at which the lime, magnesia, etc., collect on hot surfaces, varies with the character and proportions of the solids, but generally such deposits are not a serious trouble with water temperatures lower than 140 F.

Coal-burning, direct-fired water heaters may be constructed of cored cast-iron sections or of steel. In some cases the external appearance of the cast-iron sections is the same as in heating boilers, but internally the cores are changed to enable the sections to withstand the city water pressure. In small capacity water heaters, efficiency is not considered so important as low first cost and ability to maintain a fire at a low rate of combustion, and consequently such heaters are generally built with a *dry* section or fire-brick lining at the base of the fire-pot to prevent too much chilling of the fuel. While mud and scale will eventually clog the water ways of any direct-fired heater, increased life may be obtained by pro-

TABLE 6. HOT WATER DEMAND PER FIXTURE FOR VARIOUS TYPES OF BUILDINGS
Gallons of water per hour per fixture, calculated at a final temperature of 140F

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	Y.M.C.A.
1. Basins, private lavatory	2	2	2	2	2	2	2	2	2	2
2. Basins, public lavatory	4	6	8	6	8	12	6	....	15	8
3. Bathtubs	20	20	30	20	20	30	....	20	....	30
4. Dishwashers	15	50–150	....	50–150	50–200	20–100	....	15	20–100	20–100
5. Foot basins	3	3	12	3	3	12	....	3	3	12
6. Kitchen sink	10	20	....	20	20	20	....	10	10	20
7. Laundry, stationary tubs	20	28	....	28	28	....	....	20	....	28
8. Pantry sink	5	10	....	10	10	....	....	5	10	10
9. Showers	75	150	225	75	75	225	....	75	225	225
10. Slop sink	20	20	....	20	30	20	15	15	20	20
11. Demand factor	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
12. Storage capacity factor[a]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

[a] Ratio of storage tank capacity to probable maximum demand per hour.

viding a three-way cock in the return line between the heater and the bottom of the storage tank, so that water can be blown through the heater or the tank separately at full line pressure to clean out loose sediment. Clean-out openings in the bottom of the heater are advantageous if used by operators of water heaters for periodic cleaning out of sediment.

Oil-burning, direct-fired water heaters usually are of steel and operate with higher flame temperature and better efficiency than commensurate sized coal-burning heaters. They have the same tendency as coal boilers to to *lime up*, and the water passages should be large in cross-section and accessible for periodic cleaning.

Gas-burning, direct-fired water heaters may be of the instantaneous or storage type. Instantaneous heaters are generally constructed of spiral water tubes of copper around which the products of combustion circulate upward from high capacity burners. Storage-type heaters may include in one unit an insulated storage tank, a combusion chamber, flues, burner

equipment, and controls, or may consist of a separate storage tank and external direct-fired water heater, which may be a so-called *side-arm* heater for small capacity or a gas-fired boiler for larger capacity. Gas boilers used for direct hot water supply must be able to withstand the city water operating pressure. While direct-fired gas heaters are used generally for residences and small installations of 100 gal storage capacity or less, indirect heaters are recommended for larger installations.

Chimney connections for all direct-fired, fuel burning water heaters are an important consideration. Refer to Chapter 19.

Electric water heaters for domestic hot water supply are described in the section Heating Domestic Water by Electricity in Chapter 30.

In the indirect method either steam or hot water is used for heating the water. With steam the water to be heated is preferably circulated around the outside of the steam tubes which are submerged within a tank. A

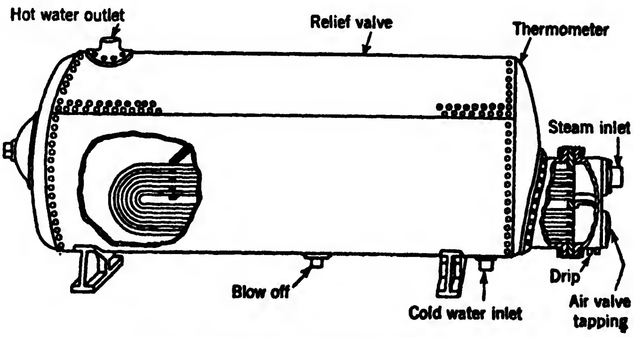


FIG. 8. INDIRECT WATER HEATER

typical indirect heater using steam is shown in Fig. 8. The coils usually are of copper and are *U*-shaped to permit expansion and contraction. The shell may be of steel, copper, or with a special inside protective lining. Where straight heating tubes are used, one end of the tube is usually expanded into a *floating* head to take care of expansion. The coils should be capable of easy withdrawal for inspection and for removal of scale. Instead of steam, the heating medium may also be hot water inside the tubes.

Another method of transferring heat from a heating boiler to the domestic water is illustrated in Fig. 9. The water heater is generally a cast-iron shell within which there is located a spiral copper coil. Hot water from the boiler circulates inside the shell and around the coil, and returns to the boiler, while domestic water from the storage tank circulates inside the coil. The storage tank should be installed with the bottom of the tank as far above the boiler as possible. Horizontal storage tanks smaller than 18 or 20 in. diameter are not recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn. In Fig. 10 the heat transfer surface is placed inside the boiler instead of in a separate vessel, but otherwise the operation is similar to that of Fig. 9. This arrangement with vertical tank is commonly used for small domestic installations.

Sometimes the heating element is located inside of the larger type fire tube boilers and small residential boilers. In this case the heat transfer surface

is in the form of a number of straight copper tubes with rear U bends or a floating head, inserted through a special opening in the boiler. While the coil may be located in the steam space above the water line of a steam boiler, it operates more satisfactorily when below the water line since clogging of the water tubes may thereby be delayed. This method is widely used without storage tanks since the intimate contact and efficient circulation of the water in this arrangement permit the utilization of the heat stored in the water of the boiler. A thermostatic three-way mixing valve is frequently used to maintain a uniform temperature of the hot water supply to the plumbing fixtures.

In order to reduce clogging by precipitated solids, water heating plants sometimes develop steam in a closed circuit, transferring the heat through

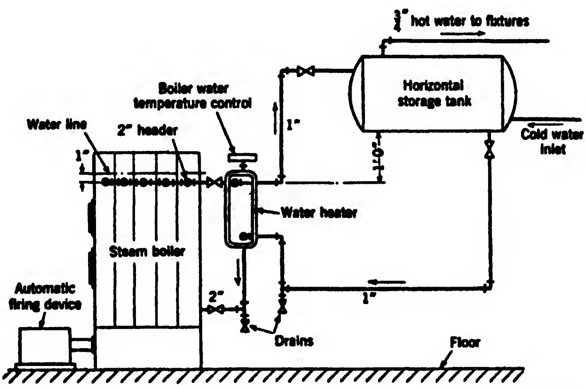


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

a tubular heater to the domestic water. The water in the primary heater, exposed to the high temperature of the fire, is repeatedly used and hence has no appreciable tendency to deposit scale, while the domestic water, heated by steam at a much lower temperature than that of the fire, also exhibits a much reduced tendency to precipitate dissolved salts.

COMPUTING HEAT TRANSMITTING SURFACE

The area of the inside surface of a heating coil may be determined from Equation 1.

$$A = \frac{Q \times 8.33(t_2 - t_1)}{U \times t_m} \tag{1}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_2 = hot water outlet temperature, degrees Fahrenheit.
t_1 = cold water inlet temperature, degrees Fahrenheit.
U = coefficient of heat transmission, Btu per (hour) (square foot) (degree Fahrenheit logarithmic mean temperature difference).
For copper or brass coils U = 240 (steam) and 100 (hot water).
For iron coils U = 160 (steam) and 67 (hot water).

t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water-temperature and is *approximately*:

$$t_s - \left[\frac{(t_2 + t_1)}{2}\right]$$

t_s = temperature of the heating medium, degrees Fahrenheit.

Equation 1 may be used to check the heating coil ratings under temperatures other than those stated in the manufacturer's published ratings.

Example 5. What area of copper transfer surface will be required to heat 70 gal of water per hour from 40 to 180 F with boiler water at 220 F?

$$t_m = \left[220 - \frac{(180 + 40)}{2}\right] = 110 \qquad A = \frac{70 \times 8.33(180 - 40)}{100 \times 110} = 7.42 \text{ sq ft.}$$

For instantaneous submerged heaters the surface required will depend upon (1) the velocity of water in the tubes, (2) the boiler water temperature, (3) the inlet water temperature, (4) the outlet water temperature, (5) the cleanliness of the coil surface, and (6) the condition of the boiler water surrounding the coil. If the heater is located in the water of an actively steaming part of a boiler, the heat transfer may be twice as great as would be obtained if the water surrounding the coil were circulating slowly. Ratings of instantaneous water heating coils will therefore vary greatly, depending upon the assumptions made regarding the conditions of operation. The values of the coefficient of heat transmission for instantaneous heaters shown in Table 7 are conservative.

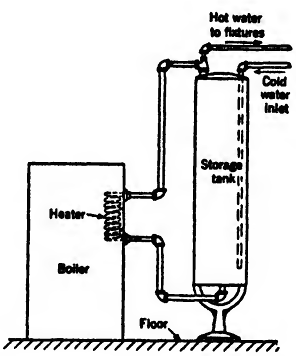


Fig. 10. Indirect Water Heater Placed in Boiler

For a coil in which heat is transferred from steam to water the value of $U = 300\sqrt{v}$ may safely be used (v = velocity of water in feet per second).

The rate of heat transfer between steam or water as the carrier and the domestic water is influenced by the rate of movement of both the carrier and the water which receives the heat. For this reason, where the transfer occurs from heating system water to domestic water, it is good practice to install a circulating pump to insure rapid movement of the boiler water.

In view of the high condensation rates obtained when steam is used with gravity circulation from the boiler, as when there is a sudden demand followed by an inflow of cold water, the bottom of a steam heating transfer

TABLE 7. COEFFICIENT OF HEAT TRANSFER OF INSTANTANEOUS WATER HEATERS
U = Btu per (hr) (sq ft) (Fahrenheit degree logarithmic mean temperature difference)

Boiler Water Temperature	210	200	180
U	225	175	150

element always should be at least 30 in. above the boiler water line, and the steam and condensate return pipes should be of liberal size. Otherwise water hammer and reduced capacity may result due to imperfect drainage of condensate.

When connecting a transfer-type hot water heater below the waterline of a cast-iron steam boiler having vertical sections, there should be a separate tapping for water circulation into every section of the boiler, as shown in Fig. 9, unless the boiler has large top nipple ports providing inter-sectional circulation. If the top nipples are entirely within the boiler steam space, no internal circulation occurs between sections and steaming may occur in sections not connected to the heater and further the unconnected sections will not deliver any heat to the heater.

CONTROL OF SERVICE WATER TEMPERATURE

Coal-fired boilers are usually controlled by an immersion thermostat located in the heated water, which opens or closes draft dampers at the boiler to adjust the rate of fuel combustion. With oil or gas-fired boilers the immersion thermostat controls the oil burner or the automatic gas valve. The gas pilot flame usually burns continuously. With electric heaters the immersion thermostat operates a switch on the source of energy.

When steam or hot water is the medium for heating the water in the tank, an immersion thermostat is used to control a valve in the steam or hot water supply line. In small residence installations, using water as the carrier, a combined immersion thermostat and butterfly valve in one simple fitting may be installed in the transmitting circuit to prevent over-heating of the service water.

In residences heated by pump circulated hot water, the house temperature is controlled by operating the circulating pump intermittently, while domestic hot water is warmed by transfer from the house boiler, independent of the pump operation. The domestic water is heated from the heating boiler the year 'round. Under such an arrangement, to prevent overheating the house by thermal circulation when the pump is not running, it is usual to insert a weighted check-valve in the house heating main, so that no circulation to the house heating system can occur unless the pump operates. In summer the fire may be controlled to maintain a boiler water temperature lower than when heating, and generally about 20 F warmer than that desired in the domestic hot water system.

In buildings which have restaurants it is generally desirable to install two separate service hot water systems so that water at about 180 F minimum may be available for dish washing, while water at 140 F maximum may be used for lavatory and bath purposes.

The immersion thermostat in a hot water storage tank should be located no higher than the center of the tank, and possibly should be even closer to the bottom since water in a tank stratifies proportionally to the temperature. When hot water is removed, the cold water entering to replace it quickly reduces the temperature in the lower parts of the tank.

SOLAR WATER HEATERS

Solar heaters utilize the energy of the sun for heating hot water. The successful operation of such heaters requires the availability of sunshine practically every day in the year, which has limited their use to Florida and the southern portions of California. When supplemented with some other means of gas, coal, or oil water heating, solar heaters may be used in climates where sunshine may be more or less intermittent. They have been used in summer homes as far north as Chicago. When properly installed and proportioned, solar water heaters render satisfactory service especially in climates where the outside temperatures are high and extremely hot water is not necessarily desirable. Such installations consist essentially of a storage tank, heating coil, and hot box. The coil is installed in the hot box and is arranged to circulate water to and from the storage

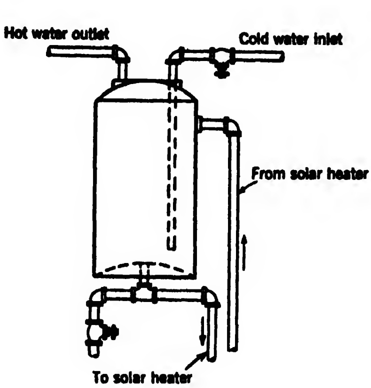


FIG. 11. SOLAR HEATER TANK CONNECTIONS

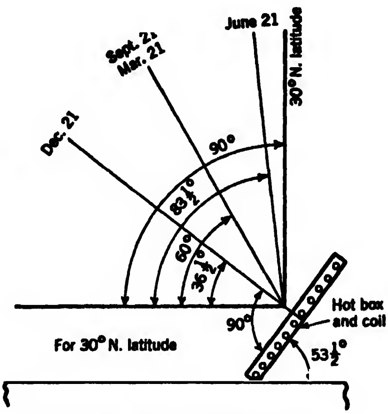


FIG. 12. SOLAR HEATING COIL INCLINATION

tank. The advantage in the use of this type of heater is the fact that it requires no fuel. The same materials should be used for the coil, circulation lines, and tank. A copper coil is more efficient in absorbing heat in the box, but galvanized iron or steel may be substituted, depending on the local water conditions, cost, and other considerations.

The storage tank must be able to store sufficient heated water for the night period of about 16 hr when the coil is not functioning or is operating under such poor sun conditions as to make its heating effect negligible. Due to the fact that the *no sun* period includes the night period when little or no hot water is used, an available storage of 50 per cent of the average daily usage is considered adequate. Since about 25 per cent of stored hot water cannot be drawn out of a storage tank before the incoming cold water reduces the temperature of all of the water in the tank to an unsatisfactory point for usage, the equation for calculating the storage capacity of the tank becomes:

$$S = \frac{Q \times 0.50}{0.75} = 0.666Q \qquad (2)$$

where

S = storage capacity of tank, gallons. Q = average daily usage, gallons.

Thus for a family of four persons using an average of 40 gal of hot water per person per day the size of the tank would be 4 persons x 40 gal x 0.666 or 106 gal, and the nearest standard size of tank to this theoretical capacity would be used. The tank should be well insulated to prevent undue loss of heat during the 16-hour period when the coil is inoperative, and it should be located as high as possible in the building (under the peak of the roof if such exists) so as to secure a maximum circulation head from the coil. The hot water supply to the house, as shown in Fig. 11, is located at the top of the tank, which serves to vent the air from the tank through the hot water faucets as fast as it accumulates.

The coil should be of the return-bend type, square or slightly rectangular in form, and should have the pipes running east and west, with the coil on

TABLE 8. SUGGESTED SOLAR HEATER DESIGN DATA[a]

Design Item	Based on Rate of 30 Gal per Day per Person								Based on Rate of 40 Gal per Day per Person							
No. of Occupants in Residence	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
Hot Water Used at Night, gal per person	15	15	15	15	15	15	15	15	20	20	20	20	20	20	20	20
Hot Water Used at Night, gal total	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Retained in Tank, 25 per cent, gal	4	8	11	15	19	23	27	30	5	10	15	20	25	30	35	40
Tank Capacity Required, gal	20	40	59	75	94	113	130	150	25	50	75	100	125	150	175	200
Hot Water Used During Day, gal	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Total Water to be Heated:																
Gal per 8 hr period	35	70	104	135	169	203	235	270	45	90	135	180	225	270	315	360
Gal per hour	4.5	9	13	17	21	26	29	34	6	12	17	23	28	34	39	45
Copper Coil Required:																
Surface area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Equivalent length 1 in. coil, ft	100	200	300	400	484	580	664	768	128	256	384	512	640	768	896	1024
Box Size:																
Area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Width, ft	4	6	7	8	9	10	10	11	4	6	8	9	10	11	12	12
Length, ft	6	8	11	12.5	13.5	14.5	16.5	17.5	8	10	12	14	16	18	19	21

[a] Sun Effect and the Design of Solar Heaters, by H. L. Alt (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 131).

the south side of the building where it can receive the full sun effect all day long without shadows from the building itself, or from adjacent obstructions such as trees or other structures. The coil should be placed as low as possible in relation to the storage tank level, such as on a porch roof, the roof of a one-story extension or, if necessary, even on the ground. Both the coil and the circulation lines should be designed to facilitate the circulation flow as much as possible, using long radius copper fittings or recessed galvanized iron fittings to match the materials of the coil, circulation lines, and tank. The coil should be inclined as shown in Fig. 12 so that the north end is raised above the south end to secure an angle with the horizontal of about 53 deg. This will result in the inlet end of the coil being on the south side (or bottom) and the outlet end being on the north side (or top). This will satisfy conditions along the 30 deg N latitude which includes the portions of Florida and Southern California where these heaters are most frequently used.

The hot box is usually constructed of wood on the four sides and bottom,

and is insulated. Over the top of the box glass sash are placed and the box should be constructed as near air-tight as possible. The interior surfaces should be painted white to reflect the heat, while the coil should be painted black to absorb the heat. The box need not be deeper than necessary to house the coil and to protect it from the weather.

The addition of a light gage copper plate on the bottom of the box to which the pipe of the coil is soldered, for good metallic contact, will add to the amount of heat received by the coil due to the fact that this plate will receive all of the sun rays which fail to directly strike the coil. The heat from this source is transmitted to the coil through the plate instead of by heating the air surrounding the coil and from which only part of the heat enters the coil, the balance being transmitted through the glass sash.

Design data given in Table 8 may be used with some judgment in selecting the size of solar heater coil and box for a particular application. These data are based on consumptions of 30 and 40 gal of hot water per day per person.

REFERENCES

[1] Methods of Estimating Loads in Plumbing Systems, by R. B. Hunter (*National Bureau of Standards*, Report BMS65, 1940). Plumbing Manual, Report of the Subcommittee on Plumbing, Central Housing Committee on Research, Design and Construction (*National Bureau of Standards*, Report BMS66, 1940). Water-Distributing Systems for Buildings, by R. B. Hunter (*National Bureau of Standards*, Report BMS79, 1941).

[2] Water Distributing Systems for Buildings, by R. B. Hunter (*National Bureau of Standards*, Report BMS79, p. 6-9). (Charts extended to flow of 0.39 gpm.)

[3] Plumbing Practice and Design, by Svend Plum (John Wiley and Sons, Inc., 1943).

BIBLIOGRAPHY

Laundry, Kitchen and Hospital Equipment, by H. C. Russell (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 45).

Water Consumption, Cost and Savings, by G. C. St. Laurent (*American Hotel Association, Hotel Engineering*, Vol. 1, 1940).

Water-Supply Piping for the Plumbing System, by F. M. Dawson and A. A. Kalinske (Technical Bulletin No. 3, *National Association of Master Plumbers*).

Use of Solar Energy for Heating Water, by F. A. Brooks, *Smithsonian Institution*, Washington, D. C.

CHAPTER 51

CORROSION AND WATER FORMED DEPOSITS CAUSES AND PREVENTION

Definitions, Classification and Characteristics of Water, Causes and Prevention of Scales and Sludges, Causes and Prevention of Slimes, Under-Water Corrosion, Atmospheric Corrosion, Buried Pipe Lines, Handling Water Treating Chemicals, Legal Regulations

THE surfaces of heating and ventilating equipment that are in intimate contact with water sometimes are affected by the chemical characteristics of the contacting waters to such extent that prohibitive amounts of insoluble materials are formed or corrosion ensues at an insufferable rate. To avoid or to correct such troubles, it is desirable that heating and ventilating engineers have a general appreciation of industrial water chemistry. The principal purpose of this chapter is to provide those criteria by which the average engineer may judge whether a problem is one that will yield to rather simple remedies or will require the skill of an experienced water technologist.

DEFINITIONS

The following definitions for water-formed deposits, corrosion, and closely allied terms have been proposed:

Water-Formed Deposits. A water-formed deposit[1] is any accumulation of insoluble material derived from water or formed by the reaction of water upon surfaces in contact with water.

Deposits formed from or by water in all of its phases may be further classified as scale, sludge, corrosion products, or biological deposits.

Scale. Scale[1] is a deposit formed from solution directly in place upon a confining surface. It is a deposit which will retain its physical shape when mechanical means are used to remove it from the surface on which it is deposited. Scale, which may or may not adhere to the underlying surface, is usually crystalline and dense, frequently laminated, and occasionally columnar in structure.

Sludge. Sludge[1] is a water-formed sedimentary deposit. It usually does not cohere sufficiently to retain its physical shape when mechanical means are used to remove it from the surface upon which it deposits. Sludge is not always found at the place where it is formed. It may be hard and adherent and baked to the surface on which it has been deposited.

Biological Deposits. Biological deposits[1] are water-formed deposits of biological organisms or the products of their life processes. Biological deposits may be microscopic in nature, such as slimes, or macroscopic, such as barnacles or mussels. Slimes are usually composed of deposits of a gelatinous or filamentous nature.

Corrosion. Corrosion[2] is destruction of a metal by chemical or electrochemical reaction with its environment. In the corrosion process, the reaction products formed may be soluble or insoluble in the contacting environment. Insoluble corrosion products may deposit at or near the attacked area or be carried along and deposited at considerable distance from the attacked area.

Corrosivity. Corrosivity[2] is the capacity of an environment to bring about destruction of a metal by the process of corrosion. Corrosivity is a property of the environment, but it has no significance until the metal in question is specified.

CLASSIFICATION AND CHARACTERISTICS OF WATER

For industrial use there is no accepted conventional classification of water. Rather, each industry usually develops a body of ideas applicable to its own water problems[4]. For heating and ventilating engineers, it is perhaps most convenient to distinguish between mineralized waters and condensates.

Mineralized Waters

All the waters found in streams, wells, lakes, and the ocean are mineralized. The same is true of all municipal supplies even though they may have been *treated*. For a given area ground waters are likely to be more highly mineralized than are surface waters. Conversely, surface waters

TABLE 1. MINERAL ANALYSES TYPIFYING COMPOSITION OF WATERS AVAILABLE AND USED INDUSTRIALLY IN THE USA

Substance	Unit	Location or Area[a,b]								
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Silica	SiO_2	2	6	12	37	10	9	22	14	———
Iron	Fe	0	0	0	1	0	0	0	2	———
Calcium	Ca	6	5	36	62	92	96	3	155	400
Magnesium	Mg	1	2	8	18	34	27	2	46	1,300
Sodium	Na	2	6	7	44	8	183	215	78	11,000
Potassium	K	1	1	1	——	1	18	10	3	400
Bicarbonate	HCO_3	14	13	119	202	339	334	549	210	150
Sulphate	SO_4	10	2	22	135	84	121	11	389	2,700
Chloride	Cl	2	10	13	13	10	280	22	117	19,000
Nitrate	NO_3	1	——	0	2	13	0	1	3	———
Dissolved Solids	———	31	66	165	426	434	983	564	948	35,000
Carbonate Hardness	$CaCO_3$	12	11	98	165	287	274	8	172	125
Non-Carbonate Hardness	$CaSO_4$	5	7	18	40	58	54	0	295	5,900

[a]All values are parts per million of the unit cited to nearest whole number (see Reference 5).

[b]Numbers indicate location or area as follows:

(1) Catskill supply—New York City
(2) Swamp Water (Colored) Black Creek, Middleburg, Florida
(3) Niagara River (Filtered) Niagara Falls, New York
(4) Missouri River (Untreated) Average
(5) Well Waters—Public Supply—Dayton, Ohio—30-60 ft.
(6) Well Water—Maywood, Illinois—2090 ft.
(7) Well Water—Smithfield, Va.—330 ft.
(8) Well Water—Rosewell, N. Mexico
(9) Ocean Water—Average

are more likely to be contaminated with municipal sewage and trade wastes. Virtually all mineralized waters also contain biological organisms.

Mineralogical Characteristics. The character and amount of extraneous inorganic materials—including deleterious gases—dissolved and suspended therein describe the mineralogical characteristics of any water. Revealing such information is the function of a mineralogical chemical analysis.

The analyses in Table 1 disclose the composition of the public water supplies used by about 45 per cent of the total population of the cities, in the United States, having more than 20,000 inhabitants[5].

All values recorded are in terms of parts per million*. This is the approved standard terminology for reporting the results of mineral analyses[6]. Values reported in the other terms commonly used may be converted into the standard form by using the factors listed in Table 2.

*Parts per million are hereinafter abbreviated ppm. A part per million signifies a unit weight of material per million unit weights of the solution.

TABLE 2. CONVERSION FACTORS FOR WATER ANALYSES

To Convert	Into	Multiply by
Grains per U. S. gallon	ppm	17
Grains per Imperial gallon	ppm	14¼
Grams per liter	ppm	1000
Mg per liter	ppm	1

Data for dissolved gases or pH values have been omitted in Table 1 because even waters of the same mineral contents may vary widely in these respects. Unpolluted natural waters usually have pH values within the range 5 to 8, depending upon their *free* CO_2 contents. Polluted waters, which include those derived from wells or swamps in marshy ground, may have pH values well below 5.

Biological Characteristics. The slime-forming organisms are mostly lower plants, grouped by botanists into the Phylum Thallophyta. This group is distinguished by the absence of leaves, roots, or stems from the mosses, ferns, and seed plants which comprise the three other phyla of the plant kingdom.

The Thallophyta (see Table 3) are divided into algae, which can synthesize chlorophyll for the production of sugar, and the fungi which lack chlorophyll and must therefore secure already synthesized carbohydrates.

All Thallophyta are of universal distribution and many of them are slime forming. Of the five divisions of algae, only three (the green, the blue-green, and the diatoms) are found in fresh water. Of the five divisions of fungi, all may occur in fresh water, the principal slime formers being indicated in Table 3.

The methods of analysis commonly used in the sanitary examination of a water have, as their principal object, to identify and count pathogens.

Most slime-forming organisms are not pathogens. When a water is subjected to a biochemical analysis for the purpose of evaluating its slime producing characteristics, tests, widely different from sanitary bacteriological tests, must be made. Tests upon the water itself are seldom satisfactory, and true indications of the sliming characteristics of a water can only be determined by the analysis of deposits on surfaces having the temperature close to the temperature of the final design equipment. The

TABLE 3. PRINCIPAL SLIME FORMERS

Phyla	Rough Division of Phyla
Algae	Single celled, sometimes forming slimy sheets. Many celled in either sheets or fronds.
Fungi	*Bacteria* (Schizomycetes) frequently forming slimy surface coatings. *Slime Molds* (Myxomycetes) forming slimy sheets as one stage of their life history. *Sac fungi* (Ascomycetes) of which one division, the yeasts, occasionally form slimy aggregates. The alga-like fungi (*Phycomycetes*) and the stalked fungi (Basidomycetes) rarely form slimes but their filaments may hold together the slimes of other organisms.

TABLE 4. PLANT WATER SUPPLY EXAMINATION

WATER SUPPLY—	200 feet deep well—average water temperature 53 F—water is producing a brown stain in plumbing fixtures.
SAMPLE—	The sample was scraped from the surface of the shell and tube condenser of #2 Freon Compressor on the meat chilling room. The sterile sample bottle was filled one-half with deposit and the balance with circulating water. No preservative was added—pH at time of collection was 7.4.
ANALYSIS REQUIRED— MACROSCOPIC ☑ MICROSCOPIC ☑ BACTERIOLOGICAL ☑ ORGANIC CONTENT ☑	
PROBLEM—	A 25-ton Freon—12 Compressor has head pressure about 10 lb higher than during initial operation, without change in water temperature. Deposits have been observed on heat exchanger surfaces. It is desired to know the nature of these deposits and if they are the cause of this increased head pressure.
MACROSCOPIC EXAMINATION—	Heavy brown flocculent material settles rapidly in clear water. pH—7.3 Odor—woody, mouldy.
MICROSCOPIC EXAMINATION—	INORGANIC MATERIAL—small amount white crystals. AMORPHOUS MATERIAL—small amount—brown. IRON BACTERIA—profuse growth of crenothrix—(Photo usually included).
CULTURAL EXAMINATION—	SABOURAUD'S AGAR 1. Aerobic gram positive spore-forming rod with mucoid sheaths. (Photo usually included). 2. Short gram negative coccibacilli (Photo usually included).
TOTAL COUNT	100,000 organism/cc.
ORGANIC CONTENT (by WEIGHT, DRY BASIS)—60%	
DISCUSSION—	The presence of common slime-forming organisms in the deposit combined with high organic content indicates that the deposit is bacterial in origin. Heat transfer reductions would be caused by such a deposit. These deposits, combined with crenothrix, can cause corrosion of both ferrous and non-ferrous metals.
RECOMMENDATIONS—	It is recommended that the water be treated at the suction side of the deep well pump with chlorine in quantities sufficient to maintain a free chlorine residual of 1.0 ppm at the discharge of the shell and tube cooler. This treatment can be scheduled on an intermittent basis.

results of such a test are commonly reported in the manner illustrated in Table 4.

Condensates

All condensates result from the chilling of water vapor. Such chilling may result from natural causes, thus producing *dews*, *sweats*, *rain*, *and snow*, or from artificial causes, as in steam condensing equipment, producing condensate or return water.

In the heating and ventilating field, the biological characteristics of condensates are likely to be of concern *only* where the condensate is used

as cooling water in recirculating systems. The deleterious gas contents of condensates, however, very often create serious corrosion troubles.

The data in Table 5 typify the chemical composition of the atmosphere in rural and metroplitan areas, and of stack gases when various types of common fuels are used. The curves in Fig. 1 disclose the solubility of the major deleterious gases present in such atmospheres in otherwise pure water, when the partial pressure of the gas is one pound per square inch absolute.

The most common deleterious gases entrained by steam are oxygen and carbon dioxide. In rare instances, hydrogen sulphide, sulphur dioxide, or ammonia are present.

In most steam condensing equipment[7,8], the non-condensable gases entrained with steam accumulate so that the amount present in the vapor space is several hundred times higher than in the incoming steam and, the amount dissolved in the condensate may therefore approach, or even exceed for short periods, the amount entrained by the steam.

TABLE 5. DATA TYPIFYING THE DELETERIOUS GAS CONTENT OF DIFFERENT ATMOSPHERES

Name of Gas	Chemical Formula	Air				Flue Gases					
		Rural		Metropolitan		Bitum. Coal		Fuel Oils		Natural Gas	
		% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia
Oxygen	O_2	21	3.143	21	3.143	2	0.299	7	1.048	10	1.497
Carbon Dioxide	CO_2	0.03	0.004	0.06	0.009	15	2.245	13	1.946	10	1.497
Sulphur Dioxide	SO_2	None	None	0.003	0.004	0.07	0.010	0.03	0.004	0.0001	0.0015

CAUSES AND PREVENTION OF SCALES AND SLUDGES

Scales may be formed on surfaces of equipment in contact with water, and sludges in the body of the water, by the separation from the water of dissolved or suspended solids. According to the nature of a particular piece of equipment and the method of its operation such separation may be promoted by one or more than one, of several factors:

a. The concentration of solids may be increased by the evaporation of water.

b. The dissolved solids may be rendered less soluble in the water by changes in temperature.

c. Conditions may favor the decomposition of unstable compounds with the formation of less soluble compounds.

Figs. 2 and 3 show that the solubilities of both calcium carbonate and calcium sulphate decrease with the rising temperature within a moderate range of temperatures. Surfaces transferring heat into water, such as condensers and coolers, are more susceptible to scale and sludge formation than are the cold parts of the same system using the same water.

The most common of the unstable soluble salts are the bicarbonates of calcium, magnesium, and occasionally iron. Under conditions favoring the removal of carbon dioxide, as when the water is strongly aerated or when it is boiled, the bicarbonates are readily converted to the relatively

insoluble carbonates (or, in the case of iron in the presence of oxygen, ferric hydroxide or oxide may be formed). Conversely, carbonates are readily converted to the more soluble bicarbonate by the addition of carbon dioxide or other acidic materials. This explains the increase in the apparent solubility of calcium carbonate at decreasing pH values (increasing concentration of hydrogen ion) shown in Fig. 2, the carbonate really going into solution largely as bicarbonate.

It is sometimes desired to evaluate the tendency in a particular water toward the separation of calcium carbonate, which may be desirable as a

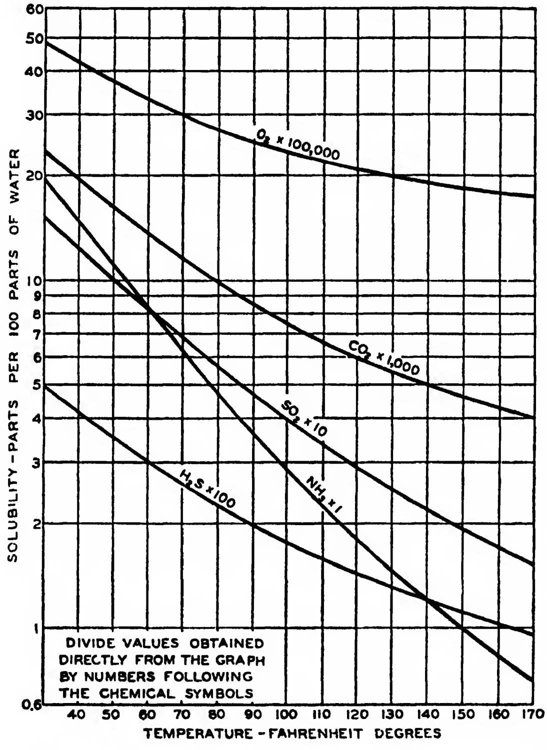


FIG. 1. SOLUBILITY OF GASES AT PARTIAL PRESSURE OF 1 PSI

means of establishing a corrosion-resistant film on metal surfaces, or in other circumstances may be undesirable because of the impedance of the calcium carbonate film to heat transfer. This tendency is indicated approximately by the Langelier Index[9], which is obtained by subtracting the actual pH of a particular water from the pH at which it is estimated precipitation of calcium carbonate would just begin. This estimate may be made by the use of Fig. 4.

There are various expedients which may be employed for avoiding or mitigating difficulties due to scales:

a. The water may be treated before use to remove elements such as calcium, magnesium, and iron, which form relatively insoluble compounds. In the various softening processes this removal of these elements is accompanied by the addition of other elements, particularly sodium, the compounds of which are relatively soluble.

b. The water may be treated within the equipment to promote the separation of dissolved solids as sludges, rather than as scale which is, in most cases, more objectionable.

c. The increase in total solids due to the evaporation of water may be controlled by the displacement, continuously or intermittently, of some of the used water by fresh supply.

d. Substances, such as the polyphosphates, having the property of inhibiting the precipitation of calcium carbonate from solutions supersaturated with it, may be added.

e. The pH of the water may be lowered (hydrogen ion concentration raised) to reduce the tendency for precipitation of carbonate. This is permissible only to such an extent as will not cause a serious increase in rate of corrosion.

The choice of the best expedient must be made for each type of equipment and will be affected by local considerations.

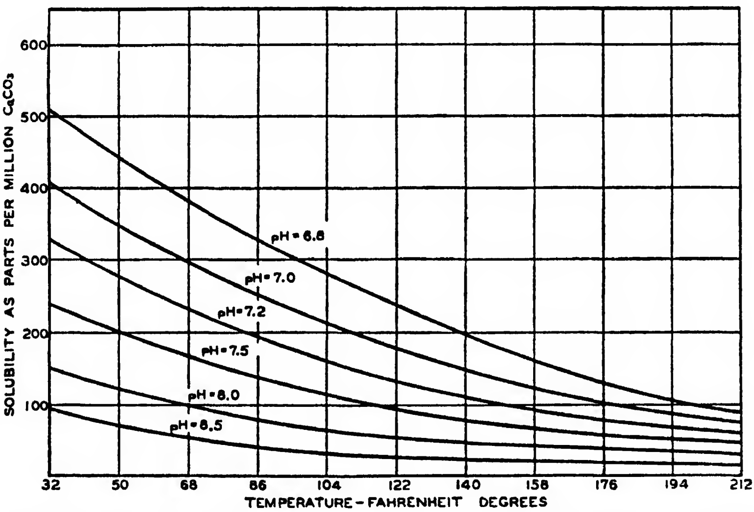


FIG. 2. SOLUBILITY OF CALCIUM CARBONATE IN DISTILLED WATER CONTAINING CARBON DIOXIDE

(pH Values at Approximately 73 F)

Fig. 2 Adapted from (1) Ind. & Eng. Chem., 20 (1928) 1197—by Baylis. (2) J.A.C.S. 50 (1929) 2086—Frear & Johnson.

Once-Through Equipment and Closed Recirculating Systems

Where abundant supplies of water are available at low cost, the cooling water may pass through the equipment once, undergoing a slight rise in temperature. Little difficulty from scale should be experienced in this case unless the carbonate hardness is more than 200 ppm, or the water has been treated to induce incipient calcium carbonate precipitation. Closed recirculating systems in which the water is cooled indirectly, as in radiators, and returned to the equipment, should usually be little troubled with scale. However, in both once-through and closed recirculating systems, slimes may cause trouble.

If there is some tendency for scaling, it may usually be prevented by the addition of small amounts, about 5 ppm or less, of polyphosphate[10]. Alternately, a minor lowering of pH by the addition of carbon dioxide or sulphuric acid may be effective if permissible from corrosion standpoint.

Open Recirculating Systems

Where water from condensers and similar equipment is passed through a spray pond or cooling tower and then returned to the equipment, there is an increase in the concentration of solids because of the evaporation of some of the water into the cooling air, and, moreover, the aeration removes carbon dioxide. Both factors promote the tendency to deposit scale. If the conditions are particularly adverse, it may be necessary to subject the water to a softening treatment before use, this being the more feasible because of the reduced water requirement in such a recirculating system. When this is not practicable, or when the tendency to scale formation is only moderate, a considerable improvement may be effected by the

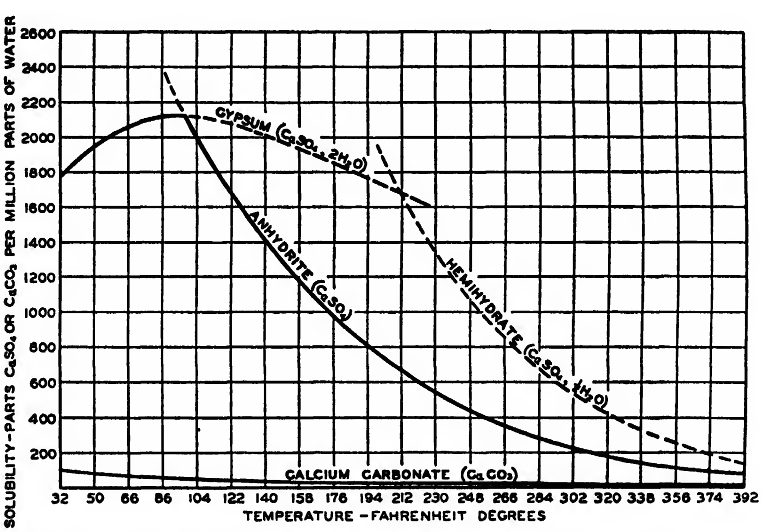


FIG. 3. SOLUBILITY OF CALCIUM SULFATE AND OF CALCIUM CARBONATE FOR COMPARISON

($CaCO_3$ in Equilibrium with Normal CO_2 Content of the Atmosphere)

Fig. 3 Adapted from Bul. No. 15, Univ. of Mich. "Formation and Properties of Boiler Scales" by P. E. Partridge.

addition to the water of organic compounds such as gelatine, glucosates, dextrine, and tannin which tend to prevent precipitated material from forming adherent scales. In systems of this kind, the loss of liquid as spray from the cooling towers or spray ponds may limit adequately the final concentration of solids in the cooling water. If not, provision must be made for sufficient purging of used water.

Heating Systems

In hot water heating systems or in steam heating boilers where all condensate is returned, troubles from scaling should not be severe. If necessary, sodium phosphate or sodium carbonate may be added to the water to prevent the formation of adherent calcium sulphate scale.

Boilers and High Temperature Equipment

Where temperature exceeds 250 F, complete softening of the water is the only practical method for minimizing sludge formation. This is

usually accomplished by artificial or natural zeolites (called also ion-exchange materials) or by hot-process precipitation softeners.

In boilers operating at pressures above 100 psi virtually all the calcium, magnesium, silica, iron, and manganese salts entering with the feed water are potential scale or sludge formers.

In low pressure boilers (100–250 psig), the formation of adherent calcium sulphate (anhydrite) scales is most to be feared. Such deposits form on the *hottest* evaporative surfaces. It is a material of low heat conductivity. Even a layer of egg shell thickness may so impede the rate of heat transfer as to bring about over-heating of the metal.

The ortho-phosphates of sodium are most frequently used to prevent sulphate scales. The concentration of phosphate required is such as to cause the precipitation of calcium phosphate as sludge, thus keeping the boiling water under-saturated with respect to calcium sulphate. To a lesser extent, sodium carbonate (called also soda ash and sal soda) is also used. Most of the effective boiler compounds contain either phosphates or soda ash, or both. Certain organic materials and colloids are sometimes found to minimize scale formation. Where chemicals are introduced directly into the boiler in amounts adequate to prevent scale, sludge is formed in amounts proportionate to the calcium and magnesium salts entering with the feed water. To prevent troublesome accumulation of this sludge, as well as soluble salts, as evaporation occurs some *blowdown* of boiler water is necessary.

CAUSES AND PREVENTION OF SLIMES

A water containing slime-producing organisms will produce prohibitive amounts of slime *only* when the conditions of use are such as to propagate their life processes. Whenever sufficient food material from normal water or from air-borne dust combines with optimum temperature conditions such as exist on cooling surfaces and air washers, serious quantities of slime will be produced.

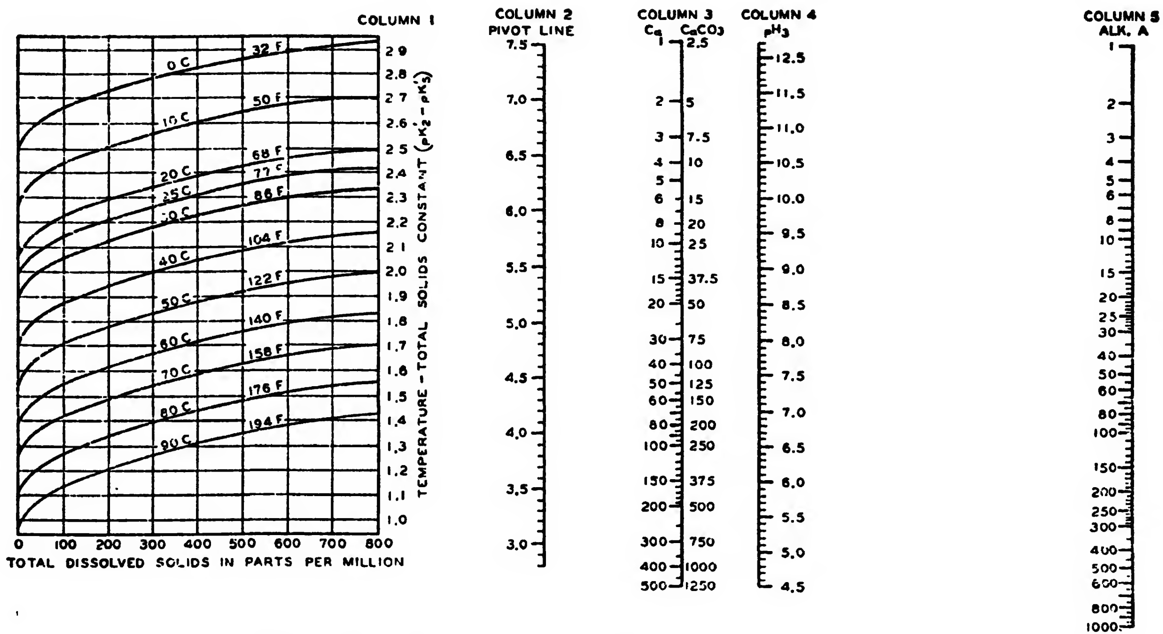
Some natural well waters do not contain sufficient foods to support luxuriant slime growths. Algae which require light for carrying on their life processes are likely to cause difficulty in cooling towers and other areas where sunlight is abundant. The ordinary slime-forming bacteria are capable of using a wide variety of nitrogenous and cellulose material as food sources. These bacteria thrive best under dark conditions such as exist in condensers and other heat transfer surfaces. Other organisms capable of causing similar difficulties use such a wide variety of food material as algae[11], iron compounds[12], and inorganic sulphates[13].

At present, the use of toxic chemicals and irradiation are the two general means employed in slime control. The value of ultra violet light, used so broadly in the beverage industry, is somewhat in dispute.

Anti-fouling paints have been developed and are fairly satisfactory for the prevention of the growth of macro-organisms such as *barnacles* and *mussels*, but these paints must be renewed at frequent intervals and are not applicable to inaccessible areas such as the inside of pipe lines and cooling towers. Satisfactory *anti-sliming* paints have *not* been found.

Names and other pertinent data relating to some of the more common chemicals used in slime control are shown in Table 6.

Chlorine is the only chemical to which is attributed the ability to destroy slime-forming organisms. The others are presumed to poison marine organisms, most of which *recover* when the chemical is not used regularly.



(Data required for determining pH saturation.

(1) Total alkalinity, as parts per million of $CaCO_3$.

(2) Calcium in parts per million.

(3) Total dissolved solids, in parts per million.

(4) Temperature, in degrees centigrade, at which pH saturation is desired.

FIG. 4. GRAPH AND NOMOGRAM FOR DETERMINATION OF pH SATURATION BY LANGELIER'S FORMULA

(Applicable within pH Range 7.0—9.5)

[a]Based on article in Oct. 1936 issue of American Water Works Association Journal and later corrections for Tables 2 and 4. Prepared for Charles P. Hoover of the Columbus, Ohio, Water Softening and Purification Plant by M. L. Riehl.

Instructions for Using Chart:

1. Knowing temperature and total dissolved solids, find temperature and total solids constant on Col. 1.
2. Align this constant with given value of calcium on Col. 3 of Chart, then locate point on Col. 2 of Chart (Pivot Line).
3. Align this point on Pivot Line with given alkalinity on Col. 5; read pH saturation on Col. 4.

Saturation index is pH actual minus pH saturation.

E. G.—pH actual.	pH saturation.	Saturation index.
7.6	8.1	−0.5 (corrosive)
8.4	7.8	+0.6 (scale forming)

While chlorine is the most generally used chemical, others may occasionally prove to be more practicable. Choice of the chemical is conditioned largely by the design and operation of the system.

Open Recirculating Systems

In spray ponds and cooling towers of the open type, light-loving algae growths are likely to cause blocking of the distribution piping and troughs. These organisms are most troublesome in areas accessible to sunlight. Algae slimes are usually stringy in character.

In open recirculating systems, continuous use of small quantities of chlorine is generally most satisfactory. In once-through systems, where large quantities of water are used, intermittent treatment a few times each day will usually result in satisfactory slime removal and chemical economies.

Neither the phenols nor copper sulphate may be used for the removal of slime already formed. For this purpose, chlorine gas is used. After being cleaned, the other chemicals may be used to prevent the reestablishment of slime in the system. The removal of green algae from a cooling tower should never be used as an indication that the true slime-forming organisms on heat exchanger surfaces have been removed. The more resistant slime formers, which so materially reduce heat transfer efficiency, will often be unaffected by treatment which completely eliminates algae.

Table 6. Common Chemicals Used for Slime Control

Chemical	Trade Name	Physical State[a]
Chlorine	Chlorine	Gas
Hypochlorites	Calcium Hypochlorites Sodium Hypochlorites	Crystalline
Chlorinated Phenols Sodium—	Chlorophenylphenate Tetrachlorophenate Pentachlorophenate	Briquettes Briquettes Briquettes
Potassium Permanganate	Permanganate of Potash	Crystalline
Copper Sulphate	Blue Vitriol	Crystalline

[a] As shipped.

Closed Once-Through Systems

In equipment where light is excluded, slime formations are due to fungi. Usually, they predominate on the heat exchange surfaces. Bacteria form thick, soft slime. Yeast and molds form tough rubbery slimes. Chlorine and hypochlorite solutions, fed intermittently, are used to prevent such slimes.

UNDER-WATER CORROSION

When deleterious substances are present in water, the corrosivity of the solution is increased in proportion to the amount of deleterious substances present, the temperature, and usually the rate of flow of the solution over the metal surfaces. There are other relevant factors, but their influence in general is subordinate to those mentioned. Dissolved oxygen, acid

gases, and chloride salts are the corrosion accelerators most frequently encountered.

Neutral and slightly alkaline waters saturated with air, corrode iron at a rate about triple that for the same water free of air. Hot water containing oxygen will corrode iron at a rate three to four times that for the same water when cold.

Corrosion of iron decreases as the pH of water solutions increases and practically ceases at a pH of 11. If the metal contains film forming agents, such as chromium, nickel, and silicon or if the water contains inhibitors such as silicates and chromates, corrosion may in some instances be minimized.

Cold Water Services

Where water from municipal supplies is used industrially in a closed system with little or no increase in temperature, it is seldom necessary or feasible to treat the water to reduce its corrosivity. When it is mandatory, the addition of caustic soda to *maintain* a pH over 11 *plus* the addition of sufficient sodium sulphite to maintain a residual of over 100 ppm (as Na_2SO_3) usually suffices to prevent serious troubles. However, in some cases the cost may be prohibitive.

When the use of sodium sulphite or a comparable chemical for oxygen removal is prohibited, as in potable waters, the addition of small amounts of lime to maintain a Langelier Index (See Fig. 4) of 0.5 or more may prove helpful.

In systems exposed to the atmosphere, as for example air washers or storage tanks, both laboratory[14] and field tests[15] have shown that the addition of alkalies to maintain a pH greater than 8.5, plus the addition of other chemicals that produce protective films on the metal surface, will measurably decrease corrosion. Sodium dichromate, sodium silicate, and tri-sodium orthophosphate have been shown to be effective film formers in the order mentioned.

Caustic soda is usually used to raise the pH value, and sodium dichromate is most often employed as a film former in industrial waters. In old systems, not previously inhibited, about 500 ppm of sodium dichromate are usually maintained at the start. After two or three months and in new systems, a residual of about 300 ppm of dichromate usually proves effective. When insufficient dichromate is employed, pitting is sometimes accelerated. Aeration does not impair the efficiency of dichromates, but does deplete the caustic soda concentration.

In large industrial systems, the use of vacuum deaeration has been shown to be effective[16]. In small systems, the equipment required can seldom be justified, economically.

Soft water, as for example the effluent from zeolite softeners, is likely to be several times more corrosive to iron than hard waters. In small installations, the use of copper or brass pipe usually is a practical expedient. Cement lined pipe and tanks suitably resist attack.

Where the water contains slime-forming organisms, especially those bacteria that thrive on iron, chlorination of the water is imperative to inhibit tuberculation and subsequent pitting.

Bitumastic paints, applied at *regular* intervals upon *well cleaned* surfaces, will measurably prolong the life of equipment handling cold waters.

Hot Water Services

As a usual thing, corrosion does not create important troubles when temperatures are maintained below 140 F.

In closed systems where little fresh water is introduced, such as in a hot water space heating system, corrosion is usually negligible because the oxygen released in heating the water is purged through the vents.

Where large amounts of fresh water are constantly entering and are being heated, the use of mechanical deaeration is the most universally satisfactory expedient to employ. Where the use of such equipment cannot be justified economically, anti-corrosive chemicals, and the use of corrosion resistant metals, are the more practical expedients to be used.

Treating Chemicals. Alkalies, such as lime and caustic soda, silicates of soda (water glass), the poly-phosphates of soda, sodium sulphite, and sodium dichromate are usually used. Organic compounds, such as the glucosates, dextrines, and tannins are sometimes used, but their value is still a controversial matter. When any chemical is used, so many relevant factors are involved that it is always advisable to seek adequate technical counsel in inaugurating the treatment. Very often, where such precaution is not taken, new troubles are created that are more aggravating than the original difficulty[17].

Silicate of soda is used to protect iron, lead, and brass water pipe[18]. For most waters, a solution of $Na_2O{:}3SiO_2$ is recommended. Sodium silicate, equivalent to about 10 ppm added silica, should be fed to the water for the first month after which it may be reduced to give 5 or 6 ppm added silica. Where careful control of the silicate feed is exercised, the water is not injured for domestic use by this treatment. The rate of corrosion of iron pipe has been reduced by 70 per cent and dezincification of brass pipe practically stopped by this simple treatment. The amount required and the effect are not the same in all waters.

Pipe Materials. Brasses with 60 to 67 per cent copper are dezincified in some corrosive waters and in certain localities are not much more serviceable than galvanized iron or steel pipe. The zinc in brass pipes is leached out locally, leaving a plug of porous copper. The weakening of such pipe is especially noticeable under the threads. Dezincification is retarded by the use of silicate of soda (8 ppm added silica)[19].

In salt or fresh water, there is no material difference in rate of pitting of wrought iron, steel, low metalloid steels, or copper bearing steels. This is contrary to the relative performance of these metals in atmosphere.

Refrigerating Systems

Corrosion in refrigerating systems is confined to surfaces in contact with brines or those in contact with the refrigerant.

Brines. Refrigerating brines usually are comprised of sodium chloride, calcium chloride, or calcium and magnesium chlorides. The corrosivity of dilute brines is higher than their more concentrated solutions. The corrosivity of sodium brines, other conditions being fixed, is about 1.5 times greater than brines of the alkaline earth metals.

Brines are excellent electrolytes. Contact of dissimilar metals of wide potential differences when in contact with brines results in rapid corrosion by galvanic action.

The leakage of air, acid refrigerants, or both, accelerates the corrosivity

of brines. Ammonia precipitates calcium and magnesium salts thus clogging the system at restricted points.

The addition of caustic soda and sodium dichromate to brine solutions to inhibit corrosion of iron is a more or less general practice. Sodium silicate and sodium phosphate are also used at times, but tests indicate they are not as effective as is sodium dichromate. It has been suggested[20] that 125 lb of sodium bichromate per 1000 cubic feet of calcium chloride brine, and 200 lb per 1000 cubic feet of sodium chloride brine be added to inhibit brines; that when salt or calcium chloride is added to "strengthen" brine, sodium dichromate also be added in the amounts shown in Table 7.

Refrigerants. The common refrigerants, except those of the hydrocarbon type, will attack the common metals and alloys if *moisture is present.* Even a very small amount of water may cause severe corrosion

TABLE 7. QUANTITIES OF SODIUM DICHROMATE TO BE ADDED TO MAINTAIN INITIAL CONCENTRATION

SPECIFIC GRAVITY OF BRINE TO BE STRENGTHENED	LB SODIUM DICHROMATE PER 100 LB $CaCl_2$ ADDED
1.16	0.695
1.18	0.621
1.20	0.556
1.22	0.502
1.24	0.455
	LB OF SODIUM DICHROMATE PER 100 LB NaCl ADDED
1.12	1.79
1.14	1.47
1.16	1.32
1.175	1.18

with certain refrigerants. The amount required need only be sufficient to produce a water film on the metal surface.

With the halogenated hydrocarbons, complete elimination of water is much to be desired. Where ammonia is used, copper and its alloys, aluminum and zinc, are attacked especially at elevated temperatures. When sulphur dioxide is used more than 50 ppm (0.005 per cent) of water will cause appreciable corrosion of virtually all the common materials.

Minimizing Condensate Corrosiveness

There are four expedients that may be utilized to minimize corrosion in steam condensate systems: (1) treatment of the boiler feedwater so as to eliminate deleterious gases entrained with the steam, (2) design of the condensing equipment to minimize dissolution in the condensate of the deleterious gases entrained with the steam, (3) chemical treatment of the condensate, (4) use of resistant metals.

Boiler Feedwater Treatment. Elimination of oxygen from boiler feedwater and, therefore, from the steam developed, can be accomplished either mechanically or chemically. In some steam generating stations, both expedients are employed.

Tests[21] have indicated that in small low-pressure heating boilers, where the boiler input contains less than about 50 ppm of carbonate hardness,

the CO_2 in the steam can be controlled by adding calcium hydroxide to the boiler. In Fig. 5 are shown the equilibria conditions proposed for boilers operating at pressures up to about 5 psi gage. This expedient may *not* be used in higher pressure boilers, because of the possibilities of scale and sludge formations. In the latter, the only method used to date for treating the feedwater consists (a) in removing the alkaline earth salts, *i.e.*, softening with (b) subsequent acidulation followed by deaeration at temperatures near the atmospheric boiling point of water[22].

Design of Condensing Equipment. In the design of water heaters and comparable types of condensing equipment[23], it is possible to shift the accumulation of non-condensible gases to a location away from the condensate level and, subsequently, vent these gases to the atmosphere.

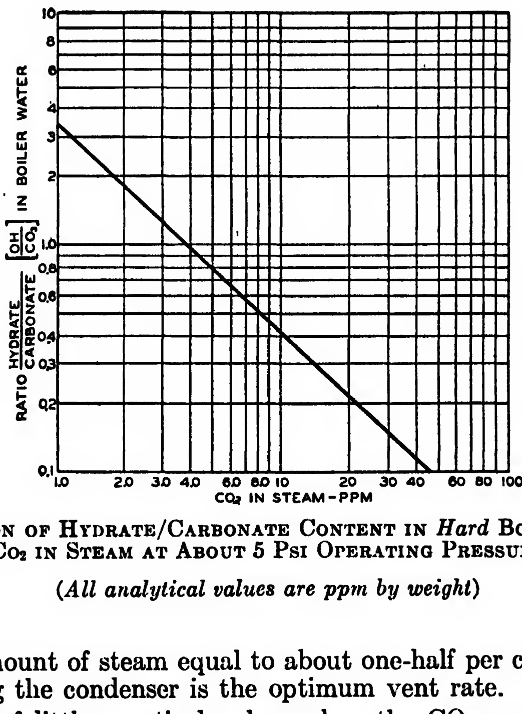


FIG. 5. RELATION OF HYDRATE/CARBONATE CONTENT IN *Hard* BOILER WATER AND CO_2 IN STEAM AT ABOUT 5 PSI OPERATING PRESSURE

(All analytical values are ppm by weight)

Venting an amount of steam equal to about one-half per cent of the total steam entering the condenser is the optimum vent rate.

Venting, is of little practical value, when the CO_2 content of the incoming steam is below about 5 ppm. When the steam contains more than 5 ppm, venting provides a means of producing a condensate containing a minimum of about 3 ppm. However, even as little as 3 ppm of dissolved CO_2 can produce active corrosion if large amounts of condensate are flowing.

Chemical Treatment of Condensate. Condensates containing comparatively large amounts of oil, are practically non-corrosive, due to the protective film provided by the oil. When oil is intentionally added to condensate[24], inadequate quantities may accelerate rather than decelerate corrosion on those surfaces not covered by the oil. Sodium silicate added to CO_2-bearing condensate has been shown to decrease, but not entirely prevent, corrosive action. It is not definite whether the protection afforded by silicate solutions is due to the establishment of a protective

film on the metal surface or to neutralization of the CO_2 by the alkali in the silicate solution.

It has been postulated that ammonia[25], cyclohexylamine[26], ethylene diamine, and morpholine[27] will retard corrosion of condensate lines. Tests with benzylamine have also been reported[28]. Where copper and its alloys are involved, the use of alkaline inhibitors is believed inadvisable. The use of small amounts of sodium hexametaphosphate has been suggested too, but tests[29] indicate that this salt accelerates rather than decelerates, the rate of attack of steel by condensate containing CO_2 and oxygen. Whether chemical treatment of steam or condensate is feasible, must be determined not only upon the basis of the acuteness of corrosion troubles, but also upon the uses to which the steam or condensate is put.

Use of Resistant Metals. For economic reasons, the metals known to resist corrosion can seldom be used exclusively for condensate lines in any

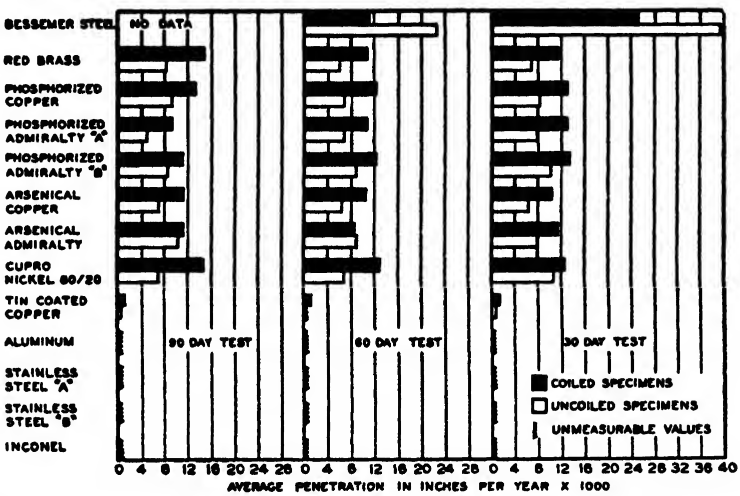


FIG. 6. COMPARATIVE CORROSION RESISTIVITY OF 10 MATERIALS EXPOSED TO CONDENSATE

sizeable enterprise. Nevertheless, there may be instances where the use of a limited amount of the more costly, but resistant, materials can be justified. The data in Fig. 6 are the results of tests[30] designed to reflect the corrosion resistance of the more commonly used metals to attack by condensate containing oxygen and CO_2.

In contemplating the use of a resistant metal, as a section of a condensate line, it should be remembered that, if other conditions are *right,* corrosive attack will merely be transferred down stream in the system. Galvanic corrosion resulting from the contact of dissimilar metals in a condensate line seldom occurs. No paint or similar protective coating has thus far proven satisfactory. Tests of cement lined and vitreous lined pipe have shown the linings to be readily dissolved by hot condensates.

ATMOSPHERIC CORROSION

Most of the problems originated by atmospheric corrosion occur in connection with the fire-side of boilers and furnaces (including their flues

and stacks), sewer vents, air ducts, coal and ash handling equipment. Usually such equipment is fabricated from common types of ferrous metals.

Generally little or no atmospheric corrosion occurs at temperatures higher than the boiling point of water, because at such temperatures little or no condensate is formed. If it does form at the higher temperatures, only negligible amounts of carbon dioxide and oxygen, present in the atmosphere, will dissolve in the hot liquid but sulphur gases may dissolve and cause rapid attack. Oxygen, sulphur dioxide, sulphur trioxide, and carbon dioxide are the deleterious gases most frequently accountable for corrosion in moist atmospheres.

Coal Storage and Handling Equipment

Virtually all coals contain sulphur in the form of pyrite, and some moisture. In storage, the pyrite is likely to be decomposed by oxidation. Moisture dissolves the products of decomposition forming sulphurous and sulphuric acid. The acid solutions vigorously attack the supporting metal.

Rubber linings have been developed for coal chutes and bins that effectively resist corrosion and the abrasive action of the coal, but they are expensive[31]. Concrete linings for steel bunkers have also been effectively employed[32].

The use of high chromium steels is not always a sure cure, especially with coals treated with dust allaying agents high in chlorides.

Flues, Stacks, and Fire-side of Boilers

The surfaces of flues and boilers contacting the products of combustion, seldom experience corrosive attack when the equipment is in operation. Breechings, smoke hoods and canopies in contact with flue gas may, however, be subject to attack during the warming-up period of an appliance or when the rate of operation is so low that the temperature of the flue gas is below the dew-point. It is common practice to use cast-iron or acid resistant vitreous enameled steel in flue gas connections to appliances to prolong the life of these parts. The shut-down period when condensation of moisture occurs on the metal surfaces is usually the time when most damage is done[33]. In those sections of the stacks where flue gas temperature drops below the dew-point, corrosion is inevitable during operation.

It is clear that where long shut-down periods are anticipated, a practical method for mitigating corrosion is to clean the surface thoroughly and to provide adequate clean, dry air circulation to prevent condensation (See also Care of Idle Heating Boilers, Chapter 18).

Protective coatings with organic binders are destroyed rather rapidly above 400 F because of the decomposition of the organic materials. The surfaces of metals, whose temperature does not exceed 400 F, may be protected by periodically applying paints such as those specified in the following paragraphs entitled Air Ducts.

Air Ducts

The most practical method for protecting air duct surfaces made of steel from atmospheric corrosion is to apply protective paints. One of the most effective protective coatings is red lead paint.

Three coats of paint should be applied, of which the first two coats should be rust inhibitive paint such as red lead paint with the second coat tinted to a light brown color with carbon black, and the finishing coat may

be red lead paint tinted to a black or brown color, black paint made according to Federal Specification TT-P-61, red iron oxide paint conforming to Federal Specification TT-P-31, or white or light tinted paint made according to Federal Specification TT-P-40.

Another paint which has had some use for priming iron and steel is zinc chromate paint.

Under some conditions, a chlorinated rubber base paint made according to Federal Specification TT-P-91 may be used for the finishing coat, particularly where the presence of highly corrosive gases or contact with strong alkaline water would injure the standard paints. Rubber base paints should be used only for the finishing coat over regular priming and second coats.

BURIED PIPE LINES

Lines that are cold and in intimate contact with the earth are corroded from the same causes as in mineral waters, but pitting is usually more intense due to variations in concentration of salts and oxygen in solution and presence of solid materials, such as cinder in contact with metal pipe. Galvanic currents, induced by contact of certain dissolved constituents in the soil, often act over a large area, and accelerate corrosion where they leave the pipe line.

Certain bacteria that thrive in the absence of oxygen have the power to obtain hydrogen and dissociate sulphates in the soil with a resultant production of hydrogen sulfide which attacks the iron to form iron sulphide. Other bacteria have preference for cathodic hydrogen and by removing it keep natural corrosion going on at a rapid rate.

Stray electric currents from electric power generating stations sometimes find their way into buried steel structures and do damage in proportion to the current density where the current leaves the metal to enter the ground.

Pipe Materials

Under many conditions where steel would be corroded, the use of corrosion-resistant metals other than steel may be desirable even if greater in first cost. Stainless steel, copper, red brass, and bronze will resist corrosion and may, at times, be used to advantage. Sheet lead linings are excellent to prevent corrosion, and lead pipe will resist corrosion both on the exterior and interior when used for buried pipe lines as well as for pipe for general purposes. Galvanized iron pipe will resist corrosion for various periods of time depending on the soil and how long the galvanized coating lasts. The zinc used for the galvanized coating is on the electrochemical protective side of the iron and the zinc is corroded and changed to zinc compounds before the iron is attacked. This accounts for the protection afforded by galvanized iron. Even if some protection is obtained, eventually the galvanized coatings are destroyed by chemical action and the corrosion of the steel begins.

Protective Coating

Protective coatings for buried pipe lines are in a class by themselves because of the unusual service conditions and because it it not possible to maintain them by recoating when necessary. Buried steel pipe lines have been protected against corrosion with considerable success by the use of very thick bituminous coatings applied in molten condition. The best

results are obtained by applying the bituminous coatings over a standard priming coat such as red lead or a bituminous paint, and for long service it has been found that after the bituminous coatings are applied, a wrapping of asbestos fabric saturated with bitumens will prevent movement and displacement of the bituminous coatings and add greatly to the length of time satisfactory protection will be maintained.

Cathodic Protection

Protection is obtained by rendering the structure cathodic to the surrounding water or soil by means of a controlled difference of potential. This method, which has proved satisfactory and economical on a number of gas and oil pipe lines underground, has also been applied with some success to the protection of the inside of water storage tanks and other structures that are in contact continuously with water. Protective coatings that insulate a large portion of the metal surface will reduce very materially the total amount of protective current that it is necessary to impress on bare anodic areas to arrest corrosion.

For each structure, it is necessary to determine or estimate the minimum current density required and design the anode or anodes so that the necessary protection can be obtained most economically. In water having relatively high electrical conductivity such as in sea water, this is comparatively easy compared with fresh water. In the latter, the composition of the water is a major factor. It is therefore desirable to obtain an accurate estimate of the minimum current density required. The current is then controlled by the potential between the anode and the structure to be protected.

Rectifiers have generally proved to be the most practical means for supplying the necessary current for protection of surfaces in contact with neutral waters[34].

HANDLING WATER TREATING CHEMICALS

Virtually all the chemicals used in water conditioning are injurious if taken internally in large doses. Many also cause severe skin irritation. Thus, they should be handled with caution.

Caustic soda, lime, and concentrated sulphuric acid will burn the flesh. In addition, if mixed with small amounts of water, sufficient heat may be generated so that spattering occurs or the container becomes too hot to handle.

The chlorophenol compounds, even in the low concentrations used in water conditioning, have been reported[35] to produce dermatitis. Chrom-itch is not uncommon among workers handling chromates. The amines are said to be absorbed through the skin[36]. Morpholine is said to cause kidney and lung trouble when so absorbed.

Chlorine gas irritates the skin, eyes, and mucous membranes. Concentrations as low as 0.004 per cent by volume in air cause dangerous illness in 0.5 to 1 hour.

When relatively large amounts of the non-gaseous chemicals are to be handled, protective clothing, including goggles, should always be provided, and a shower head or its equivalent provided at or very near the point where the chemicals are mixed. *Chemicals should always be washed from the skin with large volumes of water.*

For the handling of chlorine and chlorinators, the U. S. Public Health Service[37] stipulates the following safety requirements:

1. Suitable gas masks and a small bottle of ammonia for testing for leaks should be kept at convenient points immediately outside the room or enclosure in which chlorine is being stored or is in use. Gas masks should be inspected at regular intervals and kept in serviceable condition. Note:—All purpose masks offer adequate protection only when the concentration of acid gases does not exceed two per cent—See "Safe Practises Pamphlet #64 National Safety Council".

2. Chlorinating equipment and cylinders of chlorine should be housed preferably n separate buildings above the ground level.

3. The room or building housing chlorinators in service should be maintained at a temperature above 60 F, but never in excess of the normal summer temperature. The cylinders of chlorine should be shielded, where necessary, from excessive heat or cold. Direct heat should not be applied to cylinders of chlorine nor should hot water be poured over them or come in contact with the cylinder valve.

4. Adequate ventilation should be provided for all enclosures in which chlorine is being fed or stored.

5. All joints of tubing connecting chlorine cylinder and chlorinators should be kept absolutely tight and inspected frequently to insure tightness. Tubing should slope upward from the cylinder.

LEGAL REGULATIONS

In a number of states, the water used for humidification, even in industrial plants, is required to meet drinking water standards insofar as bacteriological quality is concerned. A ruling of the *U. S. Department of Agriculture*, Meat Inspection Division prohibits the use of chromate in water used for air washing when the air later contacts foodstuffs[38].

There is an ever growing consciousness on the part of public health officials of the necessity for regulations to *protect* potable water supplies. Attesting this is an ordinance[39] now in effect in Detroit, Michigan, which stipulates in part:

"No physical connection shall be maintained between lines carrying city water and pipes, pumps, or tanks supplied from any other source. Where dual supplies are necessary or desired, lines carrying city water must be protected against back flow of polluted water by an *atmospheric gap*—Secondary supplies and emergency sources shall include: surface waters from rivers, lakes, ponds lagoons, and reservoirs; well waters both deep and shallow; any supply of water which has been stored, held, or reserved after being used for industrial purposes; cooling water, or water which has in any way been treated, processed, or has been subjected or exposed to any contamination of a bacteriological or chemical nature; and water from any other source than the city supply."

The U. S. Public Health Service stipulates:

"Salts of barium, hexavalent chromium, heavy metal glucosides, or other substances with deleterious physiological effects, shall not be allowed in the water supply system."

The same agency recommends that the concentration of the substances listed be held below the the values cited in following table:

Substance	Max Concentration, ppm	Substance	Max Concentration, ppm
Copper	3.0	Arsenic	0.05
Iron & Manganese (Total)	0.3	Selenium	0.05
Magnesium	125.0	Phenols (Total)	0.001
Zinc	15.0	Poly-phosphate of Sodium	10.0
Lead	0.1	pH Value @ 25C	10.6
Fluorine	1.0		

The Board of Directors of the *American Water Works Association* has accepted these values as standard for *all* public water supplies in the United States[40]. While their action is not binding, prudence dictates that no form of treatment should be used that will result in raising the concentration of the substances listed above the value cited.

Since virtually all of the permissible chemicals used for scale, slime, and corrosion control have deleterious, physiological effects if taken internally in *relatively large doses*, they should always be carefully *proportioned.* To insure this, the Detroit ordinance stipulates that the chemical feeding device must have the following major characteristics:

"1. There shall be a visible means of checking the quantity of material being applied by the feeding device.

2. A water metering device, sealed to prevent tampering, shall be installed to measure the flow of water being treated.

3. The device shall be constructed so that in the event of back-flow or vacuums, the maximum amount of material that may be possibly back-siphoned from the device or any of its attachments or parts shall not exceed one fluid ounce.

4. Should there be a failure of the water metering device or the water supply, the feeding device shall automatically cease operating."

REFERENCES

[1] Annual Report (1947) Committee D-19 (*American Society for Testing Materials*)

[2] Handbook on Corrosion, *Electrochemical Society* (Unpublished).

[3] Discussion, *Corrosion and Material Protection*, May 1945, p. 2.

[4] Round-Table Discussion on Organizing the Classification of Industrial Waters, by W. C. Schroeder (Proceedings, *American Society for Testing Materials*, Vol. 44, 1944, p. 1051).

[5] Typical Water Analyses for Classification with Reference to Industrial Use, by W. D. Collins (Proceedings, *American Society for Testing Materials*, Vol. 44, 1944, p. 1057).

[6] A.S.T.M. Standard D-596—41 (*American Society for Testing Materials*).

[7] Discussion on Preventing Solution of CO_2 in Condensates, by E. W. Guernsey (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 69).

[8] Studies of the Mechanism of Solution of CO_2 in Condensates Formed in Steam Heating Systems of Buildings, by L. F. Collins (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 39).

[9] The Analytical Control of Anti-Corrosion Water Treatment, by W. F. Langelier (*Journal, American Water Works Association*, Vol. 28, 1936, p. 1500).

[10] Surface-Active Properties of Hexametaphosphate, by G. B. Hatch and Owen Rice (*Industrial and Engineering Chemistry*, Vol. 31, 1939, p. 51).

[11] Slime Control in Cooling Equipment with Phenol Derivatives, by J. A. Holmes (Proceedings, Annual Water Conference, *Engineers Society of Western Pennsylvania*, 1944, p. 61).

[12] Tuberculation of Mains as Affected by Bacteria, by H. G. Reddick and S. E. Linderman (*Journal, New England Water Works Association*, Vol. 46, 1932, No. 42).

[13] Microbiological Anerobic Corrosion of Steel Pipe Lines, by R. F. Hadley (*The Oil and Gas Journal*, September, 1939).

[14] Tests of Corrosion Inhibitors for Water Treatment in Air Conditioning Equipment, by James H. Wilson and E. C. Groesback (Research Paper 1305, *National Bureau of Standards Journal of Research*, Vol. 24, 1940, p. 665).

[15] The Control of Corrosion in Air-Conditioning Equipment by Chemical Methods, by C. M. Sterne (Proceedings, *American Society for Testing Materials*, Vol. 38, 1935, Part 2, p. 261).

[16] Cold Water Vacuum Deaeration, by S. T. Powell (Proceedings, Water Conference, *Engineers' Society of Western Pennsylvania*, 1945, p. 51).

[17] Corrosion—Causes and Prevention, by F. N. Speller (McGraw Hill Book Co., 1935, p. 366).

[18] Corrosion Control with Threshold Treatment, by G. B. Hatch and Owen Rice (*Industrial and Engineering Chemistry*, Vol. 32, 1940, p. 1572).

[19] Refrigeration Data Book (*American Society of Refrigerating Engineers*, 1936, p. 404).

[20] Engineering Problems of Water Treatment, by L. F. Collins (*Power Plant Engineering*, Vol. 50, July 1946, p. 78–81, 120).

[21] See Ref. 18, p. 470.

[22] Studies in The Detroit Edison Co. (Unpublished)

[23] Preventing the Solution of CO_2 in Condensates by Venting of the Vapor Space of Steam Heating Equipment, by D. S. McKinney, J. J. McGovern, C. W. Young and L. F. Collins (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 53).

[24] Corrosion in Steam Heating Systems, by Leo F. Collins and Everette L. Henderson (*Heating, Piping and Air Conditioning*, October 1939, p. 620).

[25] U. S. Patent 1,395,730.

[26] U. S. Patent 2,053,024.

[27] U. S. Patent 1,903,287.

[28] Treating Steam Chemically to Reduce Return Line Corrosion, by A. A. Berk, (*Industry and Power*, Vol. 53, Nov. 1947, p. 79).

[29] Tests in The Detroit Edison Co. (Unpublished)

[30] More Information Concerning Corrosion in Steam Heating Systems, by L. F. Collins (Proceedings, Water Conference, *Engineers' Society of Western Pennsylvania*, 1943, p. 37).

[31] Rubber Linings and Coatings, by J. J. McNeil (*Corrosion and Material Protection*, March-April, 1947).

[32] Protection of Steel Bins from Corrosion, by J. V. Schaefer (*Power Plant Engineering*, Vol. 26, 1922, p. 632).

[33] Some Notes on Corrosion of Cast-Iron Sectional Boilers, by E. R. Walters (*The Institution of Heating and Ventilating Engineers*, Preprint, 1944).

[34] Cathodic Protection of Steel Equipment Submerged in Water, by L. P. Sudrabin (Proceedings, Water Conference, *Engineers' Society of Western Pennsylvania*, 1944).

[35] A discussion by D. W. Haering (*See Ref. 11, p. 66*).

[36] Cyclohexylamine and Dicyclohexylamine, by T. S. Carswell and H. L. Morrill (*Industrial and Engineering Chemistry*, Vol. 29, 1937, p. 1247).

[37] Drinking Water Standards, etc. (Reprint #2440, *Public Health Reports*, Vol. 58, No. 3, January 15, 1943).

[38] Discussion of Ref. 15, by R. M. Palmer.

[39] Official Plumbing Code of the City of Detroit, Article V.

[40] Private Communication from H. S. Jordan, A.W.W.A.

CHAPTER 52

CODES AND STANDARDS

THE Codes and Standards listed in Table 1 represent accepted practice, methods, or standards prepared and accepted by the organizations indicated. They are valuable guides for the practicing engineer in determining test methods, ratings, performance requirements, and limits applying to equipment used in heating, ventilating, and air conditioning. Copies can usually be obtained *from the organization listed in the reference column.*

TABLE 1. CODES AND STANDARDS PREPARED AND ACCEPTED BY VARIOUS SOCIETIES AND ASSOCIATIONS

SUBJECT	TITLE	SPONSOR	REFERENCE
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning (1938)	A.S.H.V.E. *A.S.R.E.*	A.S.H.V.E.
Air Conditioning (120,000 Btu/Hr or less)	Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (1945).	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.* Manual No. 7
Air Conditioning (Above 120,000 Btu/hr)	The Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (1942).	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.* Manual No. 9
Airplane	Aeronautical Recommended Practice for Heating and Ventilating Airplanes (1943)	*S.A.E.*	*S.A.E.* ARP 85
Airplane	Aeronautical Recommended Practice for Internal Combustion Type Airplane Heaters (1945).	*S.A.E.*	*S.A.E* ARP 143A
Attic Ventilation	Attic Ventilation Code (1946)	*P.F.M.A.*	*P.F.M.A.*
Boilers	I=B=R Testing and Rating Code for Low Pressure Heating Boilers (1947).	*I.B.R*	*I.B.R.*
Boilers	Net Square Feet Radiation Loads in 70 Deg Fahr, Recommended for Low Pressure Heating Boilers (1943).	*H.P. & A.C.C.N.A.*	*H.P. & A.C.C.N.A.*
Boilers	Net Load Recommendations for Heating Boilers.	*H.P. & A.C.C.N.A.*	*H.P. & A.C.C.N.A.*
Boilers	Standard and Short Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2) (1929).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) (1929).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel (1932).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers (Revised April 1930).	A.S.H.V E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers (1938).	A.S.H.V.E.	A.S.H.V.E.
Boilers	*A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.	*A.S.M.E.*	*A.S.M.E.*
Boilers (Miniature)	*A.S.M.E.* Miniature Boiler Code (1946).	*A.S.M.E.*	*A.S.M.E.*
Boilers	*A.S.M.E.* Boiler Construction Code (Combined Edition) (1946 with 1947 Addenda).	*A.S.M.E.*	*A.S.M.E.*

TABLE 1. CODES AND STANDARDS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Boilers (Power)	*A.S.M.E.* Power Boiler Code, Including Rules for Inspection (1946 with 1947 Addenda).	*A.S.M.E.*	*A.S.M.E.*
Boilers (Power)	Suggested Rules for Care of Power Boilers (1946).	*A.S.M.E.*	*A.S.M.E.*
Boilers (Steel)	Steel Boiler Institute Rating Code for Commercial Steel Boilers and Residential Steel Boilers (1948).	*S.B.I.*	*S.B.I.*
Boilers (Steel)	Simplified Practice Recommendation for Steel Firebox Heating Boilers (1937).	*B.S.* *S.B.I.*	*B.S.* R157-37
Boilers (Steel)	SBI Code for Testing Oil-Fired Residential Steel Heating Boilers (1948).	S.B.I.	S.B.I.
Building Requirements	American Standard Building Requirements (1946).	*N.H.A.* *U.S.P.H.S.*	*A.S.A.* A53.1-1946
Burners (Gas)	American Standard Testing Requirements for Gas Conversion Burners (1941).	*A.G.A.*	*A.S.A.* Z21.17-1940
Burners (Gas)	American Standard Requirements for Installation of Domestic Gas Conversion Burners	*A.G.A.*	*A.S.A.* Z21.8-1948
Burners (Gas)	American Standard Requirements for Installation of Gas Burning Equipment in Power Boilers (1942).	*A.G.A.*	*A.S.A.* Z21.33-1942
Burners (Anthracite)	Commercial Standard for Domestic Burners for Pennsylvania Anthracite (Underfeed Type) (1940).	*B.S.* *A.I.L.*	*B.S.* CS48-40
Burners (Oil)	Commercial Standard for Mechanical-Draft Oil Burners Designed for Domestic Installations (1942).	*B.S.* *O.B.I.*	*B.S.* CS75-42
Chimneys (Flue Linings)	American Standard Sizes of Clay Flue Linings (1947).	*A.I.A.* *P.C.*	*A.S.A.* A62.4-1947
Cleaners (Air)	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (1934).	A.S.H.V.E.	See A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1945).	*H.P. & A.C.C.N.A.*	*H.P. & A.C.C.N.A.* Engrg. Stds., Sec. 2, Part V
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1928).	*A.S.M.E.*	*A.S.A.* A13-1928
Coils	Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Non-Volatile Refrigerants (1945).	*B.C.M.I.* *B.S.*	*B.S.* T.S. 4044
Compressors	Tentative *A.S.R.E.* Standard Methods of Rating and Testing Refrigerant Compressors.	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.*	*A.S.R.E.* Circular No. 23
Condensers	*A.S.R.E.* Standard Methods of Rating and Testing Evaporative Condensers.	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.*	*A.S.R.E.* Circular No. 20
Condensers	Tentative *A.S.R.E.* Standard Methods of Rating Water-Cooled Refrigerant Condensers.	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.*	*A.S.R.E.* Circular No. 22
Condensing Units	*A.S.R.E.* Standard Methods of Rating and Testing Mechanical Condensing Units (1940).	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.*	*A.S.R.E.* Circular No. 14-41
Conductivity	Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative) (1942).	A.S.H.V.E. *A.S.R.E.* *A.S.T.M.* *N.R.C.*	A.S.H.V.E.

TABLE 1. CODES AND STANDARDS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Control Equipment (Industrial)	Underwriter's Laboratories, Inc., Standard for Industrial Control Equipment (July 1938, reprinted Sept. 1945).	*U.L.*	*U.L.* Subject 508
Controls	Underwriter's Laboratories, Inc., Standard for Temperature Indicating and Regulating Equipment (Jan. 1947).	*U.L.*	*U.L.* Subject 873
Convector	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)(1931).	A.S.H.V.E.	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Convector	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section) (1933).	A.S.H.V.E.	A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237
Convector	Commercial Standard for Testing and Rating Convectors (1947).	*B.S.* *C.M.A.* *I.B.R.*	*B.S.* CS 140-47
Coolers (Air)	Proposed *A.S.R.E.* Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration (1945).	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.* *R.E.M.A.*	*A.S.R.E.* Circular No. 25-44
Coolers	Tentative *A.S.R.E.* Standard Methods of Rating and Testing Water and Brine Coolers.	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.*	*A.S.R.E.* Circular No. 24
Cooling Units	Standard Methods of Rating and Testing Self Contained Air Conditioning Units for Comfort Cooling (1940).	*A.S.R.E.* A.S.H.V.E. *R.M.A.* *N.E.M.A.* *A.C.M.A.*	*A.S.R.E.* Circular No. 16
Ducts and Fittings	Simplified Practice Recommendation for Pipes, Ducts and Fittings for Warm Air Heating and Air Conditioning (1945).	*Mfrs.* *B.S.*	*B.S.* R207-45
Exchangers (Heat)	Standards of *Tubular Exchanger Manufacturers Association* (1941).	*T.E.M.A.*	*T.E.M.A.*
Exhaust Systems	American Standard for Grinding, Polishing, and Buffing Equipment Sanitation (1941).	*A.F.A.*	*A.S.A.* Z43-1941
Exhaust Systems	Tentative Code of Recommended Practices for Testing and Measuring Air Flow in Exhaust Systems (1937).	*A.F.A.*	*A.F.A.* Preprint 36-27
Exhaust Systems	Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems.	*A.F.A.*	*A.F.A.*
Fans	Definitions and Terms in Use by the Blower Industry (1946).	*N.A.F.M.*	*N.A.F.M.* Bulletin No. 105
Fans	Standard Test Code for Centrifugal and Axial Fans (1938).	*N.A.F.M.** A.S.H.V.E.	*N.A.F.M.* Bulletin No. 103
Fans	Standards for Fans (1947).	*N.E.M.A.*	*N.E.M.A.* Publ. 47-128
Fire Prevention	Building Code Recommended by the *National Board of Fire Underwriters* (1943).	*N.B.F.U.*	*N.B.F.U.*
Fire Prevention	National Fire Codes (1944).	*N.F.P.A.*	*N.F.P.A.*
Fire Prevention	National Fire Code for the Prevention of Dust Explosions (1943).	*N.F.P.A.*	*N.F.P.A.*
Furnaces (Duct)	American Standard Approval Requirements for Gas-Fired Duct Furnaces (1942).	*A.G.A.*	*A.S.A.* Z21.34-1942

* Also endorsed by *P.F.M.A.*

Table 1. Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Furnaces (Gas, Floor)	Commercial Standard for Gas Floor Furnaces—Gravity Circulating Type (1942).	*B.S.* *A.G.A.E.M.*	*B.S.* CS99-42
Furnaces (Gas)	American Standard Approval Requirements for Central Heating Gas Appliances (1943).	*A.G.A.*	*A.S.A.* Z21.13-1943
Furnaces (Forced Air, Solid-Fuel)	Commercial Standard for Solid-Fuel-Burning Forced Air Furnaces (1944).	*F.H.A.* *N.W.A.H. & A.C.A* *A.I.L.*	*B.S.* CS109-44
Furnaces (Oil-Fired)	Commercial Standard for Warm Air Furnaces Equipped with Vaporizing Pot Type Oil Burners (1943).	*Mfrs.* *B.S.*	*B.S.* CS104-43
Furnaces (Oil)	A Tentative Code for Testing Oil-Fired Furnaces.	*N.W.A.H & A.C.A*	*N.W.A.H. & A.C.A.*
Furnaces (Oil)	Commercial Standard for Oil Burning Floor Furnaces Equipped with Vaporizing Pot-Type Burners (1944).	*B.S.* *O.P.A.*	*B.S.* CS113-44
Garages	Code of Minimum Requirements for Heating and Ventilating Garages (1935).	A.S.H.V.E.	A.S.H.V.E.
Gases (Toxic) and Dust	American Standard Allowable Concentration of Harmful Gases:	*A.S.A.*	*A.S.A.*
	Carbon Monoxide		Z37.1-1941
	Hydrogen Sulfide		Z37.2-1941
	Carbon di-sulfide		Z37.3-1941
	Benzene		Z37.4-1941
	Cadmium		Z37.5-1941
	Manganese		Z37.6-1942
	Chromic Acid and Chromates		Z37.7-1943
	Mercury		Z37.8-1943
	Metallic Arsenic and Arsenic Trioxide		Z37.9-1943
	Xylene		Z37.10-1943
	Lead and Certain Inorganic Lead Compounds		Z37.11-1943
	Toluene		Z37.12-1943
	Oxides of Nitrogen		Z37.13-1944
	Methanol		Z37.14-1944
	Styrene-Monomer		Z37.15-1944
	Formaldehyde		Z37.16-1944
Heat Transfer (Walls)	A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls (1928).	A.S.H.V.E	A.S.H.V.E.
Homes (Prefabricated)	Commercial Standard for Prefabricated Homes	*PHMI* *B.S.*	*B.S.* CS 125 47
Mineral Wool	Commercial Standard for Mineral Wool: Blankets, Blocks, Insulating Cement, and Pipe Insulation for Heated Industrial Equipment (1944).	*B.S.* *I.M.W.I.*	*B.S.* CS117-44
Mineral Wool	Commercial Standard for Mineral Wool: Loose, Granulated, or Felted Form, in Low Temperature Installations (1943).	*B.S.* *I.M.W.I.*	*B.S.* CS105 43
Mineral Wool	Recommended Commercial Standard for Industrial Mineral Wool Products—All Types—Testing and Reporting (1946).	*I.M.W.I.* *B.S.*	*B.S.* CS131-46
Motors	Nema Motor and Generator Standards (June 1945).	*N.E.M.A.*	*N.E.M.A.* 45-102
Piping	American Standard Code for Pressure Piping (1942).	*A.S.M.E.*	*A.S.A.* B31.1-1942
Pumps	Hydraulic Institute Test Code for Centrifugal Pumps. Hydraulic Institute Test Code for Rotary Pumps (1943).	*H.I.*	*H.I.* Section F

Table 1. Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Radiators	Code for Testing Radiators (1927).	A.S.H.V.E.	A.S.H.V.E.
Radiators	Simplified Practice Recommendation for Cast Iron Radiators (1943).	*I.B.R.* *B.S.*	*B.S.* R174-43
Refrigeration (Equipment)	Underwriter's Laboratories, Inc., Standard for Air Conditioning and Commercial Refrigerating Equipment (Feb. 1946).	*U.L.*	*U.L.* Subject 207A
Refrigeration (Mechanical)	American Standard Safety Code for Mechanical Refrigeration (1939).	*A.S.R.E.*	*A.S.A.* B9-1939**
Refrigeration (Unit Systems)	Underwriter's Laboratories, Inc., Standard for Unit Refrigerating Systems (Feb. 1946).	*U.L.*	*U.L.* Subject 207C
Refrigerators (Gas-Fired)	American Standard Approval Requirements for Refrigerators Using Gas Fuel (1941).	*A.G.A.*	*A.S.A.* Z21.19-1941
Refrigerators (Household)	American Standard Test Procedures for Household Electric Refrigerators (Mechanically Operated) (1944).	*A.S.R.E.* *U.S.D.A*	*A.S.A.* B38.2-1944
Sound	Sound Measurement Test Code for Centrifugal and Axial Fans (1942).	*N.A.F.M.*	*N.A.F.M.* Bulletin No. 104
Sound	American Recommended Practice for the Calibration of Microphones (1938).	*A.S. of A.*	*A.S.A.* Z24.4-1938
Space Heaters	American Standard Approval Requirements for Gas Space Heaters (1942).	*A.G.A.*	*A.S.A.* Z21.11-1942
Space Heaters	Commercial Standard for Flue Connected Oil-Burning Space Heaters Equipped with Vaporizing Pot-Type Burners (1943).	*I C. & H.A.M.*	*B.S.* CS101-43
Stokers	Code for Determination of Rated Capacities of Anthracite Underfeed Stokers (1944)	*S.M.A.*	*S.M.A.*
Stokers	Code for Determination of Rated Capacities of Bituminous Underfeed Stokers (1944).	*S.M.A.*	*S.M.A.*
Stokers	Recommended Minimum Firebox Dimensions and Base Heights (1944).	*S.M.A.*	*S.M.A.*
Stokers	Recommended Standards Governing Minimum Setting Heights (1944).	*S.M.A.*	*S.M.A.*
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters (1930).	A.S.H.V.E. *I.U.H.A.*	A.S.H.V.E. *I.U.H.A.*
Unit Heaters	Standard Code for Testing Hot Water Unit Heaters (1942).	*I.U.H.A.*	*I.U.H.A.*
Unit Heaters	American Standard Approval Requirements for Gas Unit Heaters (1940).	*A.G.A.*	*A.S.A.* Z21 16-1940
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (1934).	A.S.H.V.E.	A.S.H.V.E.
Unfired Pressure Vessels	Unfired Pressure Vessel Code (1946 with 1947 Addenda).	*A.S.M.E.*	*A.S.M.E.*
Vacuum Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (1934).	A.S.H.V.E.	A.S.H.V.E.
Warm Air (Gravity)	Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems (1945).	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.* Section No. 5
Water Heaters	*N E.M.A.* Standards for Electric Water Heaters (1945).	*N.E.M.A.*	*N.E.M.A.* 45-104

** Also designated A.S.R.E. Circular No. 15.

Table 1. Codes and Standards—(Concluded)

Subject	Title	Sponsor	Reference
Water Heaters	American Standard Approval Requirements for Gas Water Heaters (1944).	*A.G.A.*	*A.S.A.* Z21.10-1944
Water Heaters	Testing and Rating Hand-Fired Hot Water Supply Boilers (1948).	*F.H.A.*	*B.S.* CS145-47
Wiring	National Electrical Code, Standard of N.B.F.U. and N.F.P.A. (1947).	*N.B.F.U.* *N.F.P.A.*	*N.B.F.U.*

ABBREVIATIONS AND ADDRESSES

A.C.M.A.	Air Conditioning Manufacturers Association, superseded 1940 by A.C.R.M.A.
A.C.R.M.A.	Air Conditioning and Refrigerating Machinery Association, Southern Bldg., Washington, D. C.
A.F.A.	American Foundrymen's Association, 222 W. Adams St., Chicago, Ill.
A.G.A.	American Gas Association, 420 Lexington Ave., New York, N. Y.
A.G.A.E.M.	Association of Gas Appliance and Equipment Manufacturers, superseded 1945 by G.A.M.A.
A.I.A.	American Institute of Architects, 1741 New York Ave., Washington, D C.
A.I.L.	Anthracite Industries Laboratory, 237 Old River Rd., Wilkes Barre, Pa.
A.S.A.	American Standards Association, 70 East 45th St., New York, N. Y.
A.S. of A.	Acoustical Society of America, 919 N. Michigan Ave., Chicago, Ill.
A.S.H.V.E.	American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York, N. Y.
A.S.M.E.	American Society of Mechanical Engineers, 29 West 39th St., New York, N. Y.
A.S.R.E.	American Society of Refrigerating Engineers, 40 West 40th St., New York, N. Y.
A.S.T.M.	American Society for Testing Materials, 1916 Race St., Philadelphia, Pa.
B.S.	National Bureau of Standards, Washington, D. C.
C.M.A.	Convector Manufacturers Association, 400 W. Madison Av., Chicago, Ill.
F.H.A.	Federal Housing Administration, Washington, D. C.
G.A.M.A.	Gas Appliance Manufacturers' Association, 60 East 42nd St., New York, N. Y.
H.I.	Hydraulic Institute, 90 West St., New York, N. Y.
H.P. & A.C.C.N.A.	Heating, Piping and Air Conditioning Contractors National Association, 1250 Avenue of the Americas, New York, N. Y.
I.B.R.	Institute of Boiler and Radiator Manufacturers, 60 East 42nd St., New York, N. Y.
I.C.H.A.M.	Institute of Cooking and Heating Appliance Manufacturers, Shoreham Hotel, Washington, D. C.
I.M.W.I.	Industrial Mineral Wool Institute, 441 Lexington Ave., New York, N. Y.
I.U.H.A.	Industrial Unit Heater Association, 5-157 General Motors Bldg., Detroit, Mich.
N.A.F.M.	National Association of Fan Manufacturers, 5-157 General Motors Bldg., Detroit, Mich.
N.B.F.U.	National Board of Fire Underwriters, 85 John St., New York, N. Y.
N.E.M.A.	National Electrical Manufacturers Association, 155 East 44th St., New York, N. Y.
N.F.P.A.	National Fire Prevention Association, 60 Batterymarch St., Boston, Mass.
N.H.A.	National Housing Agency, Washington, D. C.
N.R.C.	National Research Council, 2101 Constitution Ave., Washington, D. C.
N.W.A.H. & A.C.A.	National Warm Air Heating and Air Conditioning Association, 145 Public Square, Cleveland, Ohio.
O.B.I.	Oil Burner Institute, superseded 1942 by O.H.I.A.
O.H.I.A.	Oil Heat Institute of America, 6 East 39th St., New York, N. Y.
O.P.A.	Office of Price Administration, Washington, D. C.
P.C.	Producers Council, 815-15th St., N.W., Washington, D. C.
P.F.M.A.	Propeller Fan Manufacturers Association, 2-225 General Motors Bldg., Detroit, Mich.
P.H.M.I.	Prefabricated Home Manufacturers Institute, 908 20th St., N.W., Washington, D. C.
R.E.M.A.	Refrigeration Equipment Manufacturers Association, 1107 Clark Bldg., Pittsburgh, Pa.
R.M.A.	Refrigerating Machinery Association. See A.C.R.M.A.
S.A.E.	[illegible]
S.B.I.	[illegible]
S.M.A.	[illegible]
T.E.M.A.	Tubular Exchanger Manufacturers Association, 366 Madison Ave., New York, N. Y.
U.L.	Underwriter's Laboratories, 207 East Ohio St., Chicago, Ill.
U.S.D.A.	United States Department of Agriculture, Washington, D. C
U.S.P.H.S.	United States Public Health Service, Washington, D. C.

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HEATING VENTILATING AIR CONDITIONING

GUIDE

1949

On pages 1027-1381 will be found the Catalog Data of 249 manufacturers whose products are described and illustrated.

For the convenience of the user of THE GUIDE 1949 there are eight main divisions:

On pages 1001-1024, under each of the index headings—Air Cleaning Equipment, Fans, Humidifiers, Ventilators, etc., will be found a list of manufacturers of any desired products, fully cross-indexed, and the page numbers in the Catalog Data Section where the products are described.

By reference to these indices, the manufacturers names and the page numbers, any item of equipment or materials, and the producers address, may be located quickly.

Air & Refrigeration Corporation

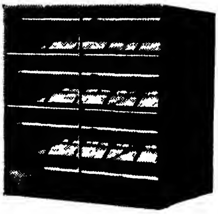
475 Fifth Avenue, **New York 17, N. Y.**

Atlanta, Ga. • **Detroit, Mich.**

Air Conditioning, Humidifying, Dehumidifying, Cooling, Scrubbing, Air Washing and Purification Apparatus

Air & Refrigeration Corporation specializes in the design and manufacture of industrial and comfort-conditioning apparatus where maintenance of suitable humidity and temperature within closely controllable limits is essential. This specialization is based on technical knowledge and ingenuity born of extensive experience in the solution of the more difficult problems of air conditioning. A complete line of air conditioning equipment is available to contractors and owners for all phases of humidifying, dehumidifying, cooling and washing.

Capillary Air Washers provide a superior type humidifying, dehumidifying, air washing, cleaning and cooling unit for central station apparatus. For most purposes the Capillary Washer requires $\frac{1}{3}$ the volume of water at $\frac{1}{3}$ the pressure used by conventional spray equipment. They are available with factory insulated casings and tank for central station applications. *For complete data, see Bulletin G-3.*

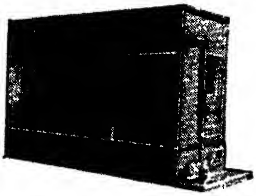


Factory Insulated Class I Capillary Air Washer

Capillary Unit Conditioners are factory insulated and assembled, ready for use. They include fan, motor, drive, heating coils, Capillary Cells with suitable sprays, spray pump and mixing dampers. Units are designed for floor mounting or for ceiling suspension, and can be arranged for the reception of cooling coils, if required. *Complete description and engineering data will be found in Bulletin G-3.*

Factory Insulated Size #3-4 Capillary Unit Conditioner

Spray Type Air Washers for washing, humidifying and dehumidifying air are all basically the same. A & R Spray Washers include special features of design developed to insure more efficient and dependable operation, lower maintenance costs, and, in many cases, lower installation costs. Such features relate especially to eliminators, collecting tanks, flooded baffles, nozzle arrangement, etc. Spray Washers can be supplied with factory insulated casings and tank for central stations applications. *For details, see Bulletin AW-1.*



Factory Insulated Prewasher for Lint Removal

Sprayed Coil Dehumidifiers for year-round treatment of air are complete with cooling coils, sprays, circulating pump and glass mat eliminators. Sprayed coil dehumidifiers are factory insulated and complete, ready for assembly in the field. Special features in design insure continuous washing and cleaning of finned surfaces and easy accessibility to all parts. *For engineering information and detailed description, see Bulletin SC-1.*

A & R Insulated Panels consist of insulation between metal sheet on one side and hard fiber board on the other, the three laminated and cemented together under pressure. This unique panel design includes the structural frame to form units which require *only bolting together* to make enclosures of any required shape for plenum chambers and many other purposes. Panels are available in widths from 3 in. to 48 in., and in lengths to 12 ft. Their use insures tremendous economies in field labor. *For details, see Bulletin P-1.*

Factory Insulated Plenum Chamber

Submit design and capacity for special recommendations; write for catalog and engineering data.

American Blower Corporation

Detroit 32, Michigan

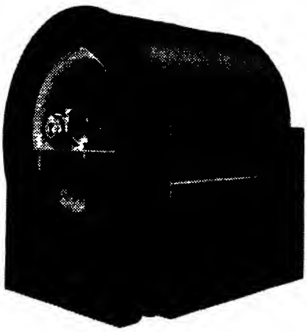
CANADIAN SIROCCO COMPANY, LTD.

310 Ellis Street, Windsor, Ontario

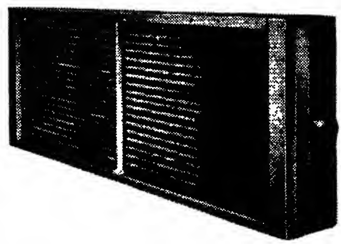
Branch Offices in Principal Cities

Division of AMERICAN RADIATOR & Standard Sanitary CORPORATION

AIR CONDITIONING — HUMIDIFYING — DEHUMIDIFYING — COOLING — VENTILATING — HEATING — VAPOR-ABSORPTION — DRYING — AIR WASHING AND PURIFICATION — EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS



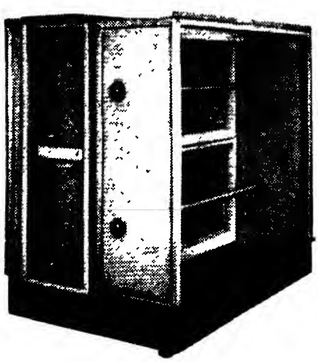
Double Inlet "ABC" Multiblade Fan—above, is a heavy duty ventilating fan. The wheel has narrow, forward pitched blades. Low tip speeds assure quiet operation. Request Bulletin A-801. Bulletin A-603 describes backwardly inclined, nonoverloading HS Fan.



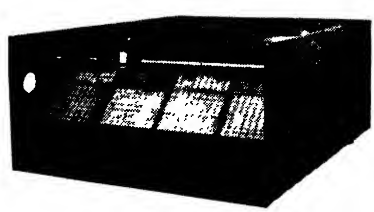
Heating & Cooling Coils—right, American Blower heating and cooling coils offer a number of improvements in design and construction. Available in a complete range of sizes and types, including:

Bulletin o	**Type S steam coils** **Type D double tube coils** **Type U return blend coils** **Type B booster coils**
Bulletin 1521	**Type W water coils** **Type C cleanable water coils** **Type X direct expansion coils**
Bulletin B-1318	**Type H heavy duty coils**

"ABC" Utility Sets—complete packaged units, directly connected or V-Belt short coupled drive for duct applications. Sizes for wide variety of ventilating problems. Quiet, compact. Bulletin 2814.



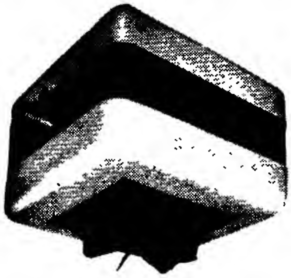
American Blower Air Washer—above, cleans, purifies and freshens air, removes dust, odors and bacteria, cools if desired and provides an effective method of controlling humidity. Bulletin 3623.



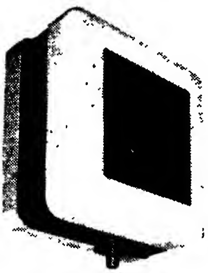
Capillary Air Washers—above, for high efficiency in cleaning, humidification, cooling and dehumidification of air. Air is forced at low resistance through long, irregular passages of small size formed by a large amount of thoroughly wetted glass surface. Write for Bulletin 3723.

TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

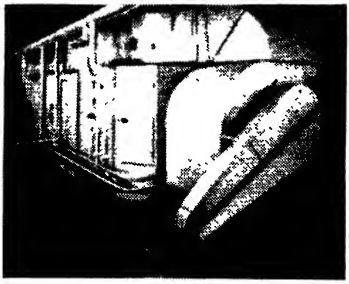
All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic and kitchen ventilation for homes.



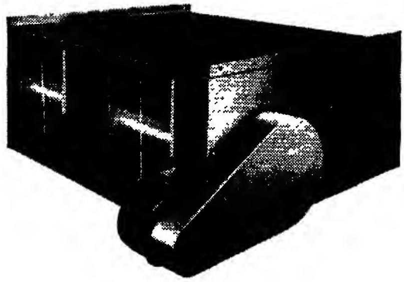
"ABC" Vertical Unit Heaters—for ceiling applications, give an even, wide floor area distribution of heat. For either steam or hot water heating systems. Variable speed, 2-speed and constant speed models. Write for Bulletin 6417.



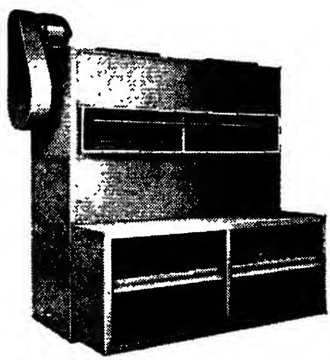
Venturafin Unit Heaters—for many general purpose heating jobs. Wall or ceiling mounted. Streamline construction, rugged heating elements. Steam or hot water. Request Bulletin 6317.



Air Conditioning Central Systems—provide an effective way of cooling, heating, humidifying, dehumidifying and purifying air in all classes of business and public buildings where a duct system is desirable. Write for special data.



Heating and Ventilating Units—with air filters and Aileron control. Ideal wherever attractive, quiet and economical heating and ventilating units are required. Wall, floor or ceiling mounting. Offer great flexibility of design and arrangement to meet specific needs. Bulletin 6017.



American Blower Air Conditioning Units.—*Type A* for all normal unitary type commercial and industrial applications. Cooling, heating, humidifying. Capacities 1000 cfm—13600 cfm. *Type S* for commercial and industrial applications desiring washed air or high relative humidities. Capacities 1000 cfm—13600 cfm. *Type M*, large capacity for central system installation with separately mounted fan. Cooling, dehumidification, heating, humidifying. Capacities 1000 cfm—41000 cfm. Bulletin 6527.

314 East St.
Crown Point, Ind.
1 Clarina St.
Wakefield, Mass.
703 Embree
Crescent
Westfield, N. J.
553 S. Figueroa St.
Los Angeles, Calif.

The Bahnson Company

Winston-Salem, N. C.

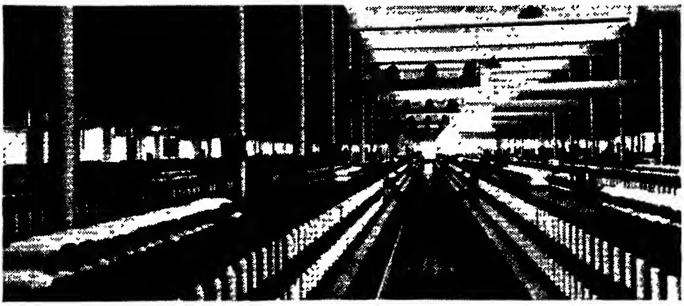
Bahnson System

AIR CONDITIONING ENGINEERS

886 Drewry St.
Atlanta, Ga.
43 Virginian Apts.
Greenville, S. C.
1001 S. Marshall St.
Winston-Salem,
N. C.

HUMIDUCT

AIR CONDITIONING SYSTEM



The Bahnson Humiduct is a unit system of air conditioning for humidifying, ventilating, evaporative or refrigerative cooling, heating, filtering, electrostatic air cleaning, and dehumidifying, in any desired combination. Through the application of delivering saturated air plus entrained moisture, wherein part of the evaporation may take place in the room, supplementary evaporation is not required with the Humiduct. The use of this principle allows a more accurate control of humidity with a lower volume of air than is required with saturated air types of systems.

BAHNSON CENTRAL STATION AIR CONDITIONING

Air washers and other components for evaporative cooling or refrigerative cooling systems are designed, manufactured, and installed by Bahnson for applications requiring central station air conditioning. Conventional air washers of single or double bank sprays and the Bahnson Centrispray which utilizes the principle of centrifugal atomization of water are manufactured in required sizes.

Bahnson Central Station with Refrigeration

Bahnson System
AIR CONDITIONING ENGINEERS

The Bahnson Air Vitalizer air conditioning system employs a dry duct unit distribution system combined with the Bahnson Centrifugal Humidifier or the Bahnson Economizer Atomizer to obtain evaporation. Ventilation, humidification, evaporative cooling, heating and air filtering are combined with the flexibility of the Air Vitalizer system to permit any air handling or evaporative capacity. Positive circulation affords even distribution with additional sensible Cooling.

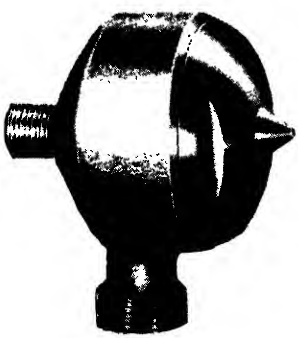
CENTRIFUGAL HUMIDIFIER

The Bahnson Centrifugal Humidifier is a completely self-contained unit humidifier requiring only water supply, drain, and electrical connection for installation. Evaporative capacity ranges up to 12 gal of water per hour dependent upon the size of the Humidifier and conditions under which it must operate. The Centrifugal Humidifier may be equipped with the Type J individual humidity control, or the highly sensitive Bahnson Master B control may be used to actuate an electric motor valve on the feed water line to a group of humidifiers.

Type H with J Control

1. **Atomizes water into a fine mist.**
2. **Diffuses the mist with room air until absorbed.**
3. **Distributes the humidified air uniformly.**
4. **Automatically controls evaporation for constant relative humidity.**

ATOMIZER



Type ESC
Pat. Pending

The Type ESC Bahnson Economizer is a pneumatic Atomizer employing air and water under pressure to produce a fine spray for humidification purposes. The Type ESC Atomizer is very simple in construction and dependable in operation. It includes a self-cleaning pin operating in conjunction with the water pressure to remove any dirt from the water line. The capacity of this Atomizer may be adjusted for amount of evaporation and quality of spray by regulating the pressure of the air and water supplying the unit. **When operating at from 20 lb to 25 lb air pressure, the unit uses as little as one-half the amount of compressed air required for aspirating type Atomizers for comparable spray quality and evaporative capacity.**

Carrier Corporation • Syracuse 1, N. Y.

MARINE DIVISION:
385 Madison Ave.
New York 17, N. Y.

Carrier

INTERNATIONAL DIVISION:
385 Madison Ave.
New York 17, N. Y.

Offices and Dealers in principal cities—refer to your telephone directory.

Carrier AIR CONDITIONING

Room Air Conditioners—compact units in attractive cabinets; two sizes in window type models and one in floor or console model. Built to provide summer comfort air conditioning, year round ventilation and air circulation for individual rooms, private offices and other similar small enclosures.

Self-Contained Air Conditioners—completely enclosed in neat cabinets, these units provide summer comfort for residences, retail shops, general offices, beauty salons, and other commercial spaces of medium size.

Assembled Air Conditioners—fully enclosed, compact units designed for installation outside the space to be air conditioned, and using ducts to distribute the air. Ideal for year round air conditioning of laboratories, offices, stores, and similar interiors.

Unitary and Central Station Air Conditioners—for groups of rooms such as offices and laboratories, and for large spaces such as stores, factories, theaters, industrial plants, and other interiors requiring year round air conditioning. Units available in floor or suspension models. Supplemented by refrigeration where cooling and dehumidifying is required.

"Weathermaster" Systems—for air conditioning of multi-story, multi-room buildings such as apartments, hospitals, hotels and office buildings. System consists of room units in decorative cabinets or for furring in under windows, each with individual control of temperature, and a central station apparatus. In one system the air is distributed through conduits requiring but little space—practical for new or old buildings. A special development of Carrier Corporation.

Blast Freezers and Cold Diffusers—for food freezing and storage, meat packing operations, and other industries requiring low temperatures. Units are available in suspension or floor models and can be used within the space to be refrigerated or remotely located and connected by ducts.

Write for descriptive literature on any of the above equipment

Carrier REFRIGERATION

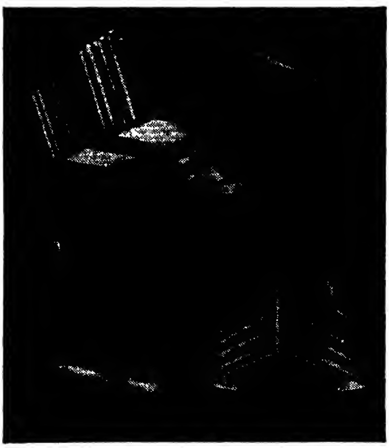
Centrifugal Refrigerating Machine—for large comfort and industrial air conditioning applications and for cooling processes down to below −100 F. These efficient machines operate with any standard drive, are simple in operation and require little maintenance. Uses safe refrigerant. Available in capacities from 100 to 1200 tons cooling.

Reciprocating Refrigerating Machines—to provide refrigeration for industrial and comfort air conditioning of moderate size and for process cooling. These new 5 Series machines are available in direct connected or belt drive, water or evaporative cooled types and are job-assembled to give widest variety of capacities. Sizes from 5 to 100 hp.

Commercial Refrigerating Machines—for storage refrigerators, display cases, milk coolers, ice makers, farm and home food freezers in a wide range of capacities and temperatures. Units use "Freon" or Methyl Chloride refrigerants and are complete with compressor, drive, air or water cooled condenser, and controls mounted on a one-piece base.

Evaporative Condensers—for condensing refrigerants and cooling liquids by a process of blowing air over wetted coils. The evaporation of the water from the surface of the coils removes a maximum amount of heat with a minimum of water consumption. Can be used in place of a cooling tower. Afford savings wherever water costs are high. May be placed outdoors without protection. Capacities range from 6 to 75 tons condensing.

Carrier INDUSTRIAL HEATING



Unit Heaters—for commercial and industrial heating uses. Available in two suspended types: one horizontal discharge model for small space heating and one vertical 4-way discharge model for larger areas. Both models have adjustable louvres for directional control. Units consist of propeller type fan, coils for steam or hot water, and drive, all neatly encased. Available with manual or thermostatic controls. Capacities range from 21,000 to 502,000 Btu per hour at 2 lb steam pressure.

Heat Diffusing Units—for commercial and industrial buildings. Suspended or floor models with centrifugal type fan, coils for steam or hot water, and selective air distribution, all factory assembled for easy installation. Capacities are from 115,000 to 1,570,000 Btu per hour at 2 lb steam pressure.

Write for descriptive literature on any of the above equipment.

Clarage Fan Company

Kalamazoo, Michigan

Application Engineering Offices **In Principal American Cities**

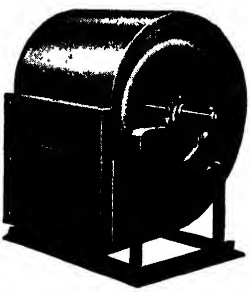
(Consult Telephone Directory)

Clarage Air-Handling and Conditioning Equipment

Fans for ventilating and air conditioning. Capacities; 200 to 200,000 cfm.

Fans • Blowers • Air Washers Air Conditioning Systems and Units • Unit Heaters & Coolers

For over thirty-five years Clarage has been a leading manufacturer of equipment for ventilating, heating, cooling, drying, air cleaning, humidifying, dehumidifying, complete air conditioning, exhausting, pneumatic conveying and mechanical draft. Clarage equipment is designed to meet all types of industrial, commercial, public building and power plant requirements. Whatever your air-handling or conditioning problem, Clarage is an excellent source of supply.



Fans for warm air furnaces, oil burners, stokers, etc. 200 to 9000 cfm.

We build many other types of fans and allied equipment. Write for a Clarage catalog covering our complete line.

Unit Heaters for factories, stores, offices, etc.—floor and suspended type units. Offered in 19 standard sizes.

Fans for exhaust systems, pressure blowing, etc. 100 to 50,000 cfm.; built for a wide range of pressures.

Air Conditioning central systems and units to solve any temperature and humidity control problem.

GENERAL REFRIGERATION
DIVISION
YATES-AMERICAN MACHINE CO., Beloit, Wisconsin

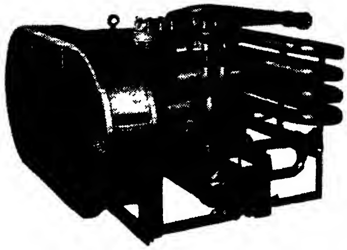
TWO GOOD NAMES TO REMEMBER

Lipman
AUTOMATIC REFRIGERATION

GR
AIR CONDITIONING

LIPMAN FOR REFRIGERATION

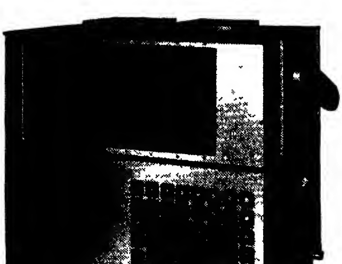
You can choose from a wide range of LIPMAN automatic refrigeration units—¼ thru 40-hp capacity—air or water-cooled—with ammonia, freon or methyl chloride refrigerant. Every unit is backed by more than a quarter century of LIPMAN service in every field of commercial refrigeration—your guarantee of dependable, economical performance.



MODEL F1509 WATER COOLED FREON—12 CONDENSING UNIT

A compact, self-contained 15-hp Freon-12 condensing unit.

GENERAL REFRIGERATION FOR AIR CONDITIONING



During its thirty years in the field, General Refrigeration has developed an outstanding line of air conditioning equipment, with a size and type for every requirement in theaters, hotels, factories, offices, shops, laboratories, etc.

MODEL 10-4000 DUCT-TYPE SELF-CONTAINED AIR CONDITIONER

Equipped with 10-hp condensing unit. Requires relatively small operating charge of refrigerant.

Let experienced GR-LIPMAN-engineers study your refrigeration or air conditioning needs and recommend the correct unit for the job. No obligation. Write General Refrigeration Division, Yates-American Machine Company, Beloit, Wis., for descriptive literature and nearest distributor.

GENERAL ELECTRIC

AIR CONDITIONING DEPARTMENT

BLOOMFIELD NEW JERSEY

DISTRICT AND LOCAL OFFICES

ATLANTA 3, GEORGIA
410 Red Rock Building

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700 Commonwealth Ave.

CHICAGO 54, ILL.
The Merchandise Mart
Room 1127

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617 Vine St., Room 1328

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925 Euclid Ave., Room 524

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903 Ross Avenue

DETROIT 2, MICH.
8735 Lyndon Ave.

KANSAS CITY 6, MO.
106 W. 14th St., Suite 2510

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1233 South Hope St.

MINNEAPOLIS 2, MINN.
12 South 6th Street

NEW ORLEANS 12, LA.
837 Gravier St., Room 1004

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200 South Main Street

SAN FRANCISCO 6, CALIF.
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806 15th Street, N.W.

G-E AUTOMATIC HEATING EQUIPMENT

A COMPLETE LINE

G-E OIL-FIRED BOILER. Available in 5 sizes, from 100,000 to 450,000 Btu. Shipped with controls and jacket dismounted, to permit easy handling of the boiler. Assembly and installation on site is a quick, simple job. Listed by *Underwriters' Laboratories*.

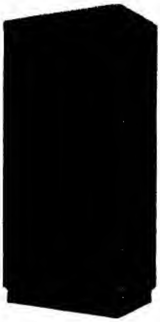
G-E CONVERSION OIL BURNER. Comes in 2 sizes, 3 capacities (1 to 3 gal). Listed by *Underwriters' Laboratories*.

G-E OIL-FIRED WARM AIR FURNACE. Small, packaged units, easily and economically installed. Four sizes, from 60,000 to 155,000 Btu. Listed by *Underwriters' Laboratories*. Models of 60,000 and 85,000 Btu output are *Underwriters' Laboratory* listed for installation with 2 inch clearance on both sides and back.

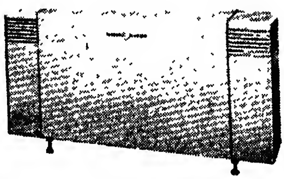
G-E GAS-FIRED BOILER. Comes in 8 sizes, with Btu output ranging from 76,800 to 345,600. *A.G.A.* approved, *Underwriters' Laboratory* listed.

G-E GAS-FIRED WARM AIR FURNACE. Compact, packaged, easily and economically installed units in 5 sizes. Btu outputs ranging from 48,000 to 168,000. All units listed by *Underwriters' Laboratories*—small units listed for alcove installation with 2 to 3 inch wall clearance.

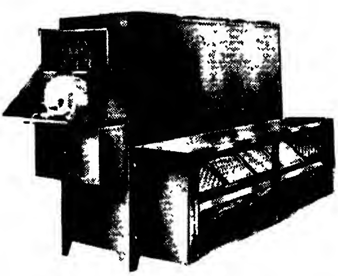
AIR CONDITIONING EQUIPMENT



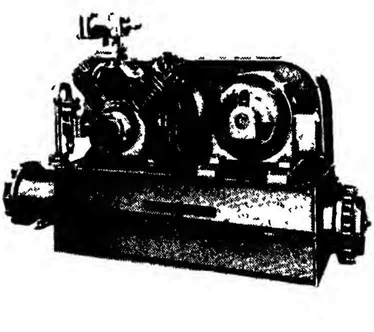
PACKAGED AIR CONDITIONERS—To provide air conditioning directly in the space or remotely installed with simple ductwork, for stores, beauty shops, restaurants, large offices, office suites, apartments, and all kinds of retail and commercial establishments. Available in 2, 3, 5, 7½ and 10 hp models.



REMOTE ROOM AIR CONDITIONERS, individually controlled, for summer cooling, or year 'round air conditioning. General Electric Type AD room units are designed for use in offices, hospitals, hotels, apartment buildings, and other places where individual control of room temperature is desired. Application and arrangement can be modified to meet a wide range of specific requirements.



CENTRAL PLANT AIR CONDITIONERS—for summer, winter, or year 'round air conditioning. These modern, radically different air conditioners are built in horizontal and vertical models, with five basic frame sizes in each type, covering a cooling range of from 0.8 to 58 tons, and a heating range from 28,100 to 1,310,000 Btu's per hour. The architect and engineer can select from 28 different arrangements, made possible by G-E Building Block Design which saves space, engineering time, and installation time. These Type HDH and HDV air conditioners are factory-built in these sections: fan, filter, coil, sump, and motormount. Simple assembly completes the job. All sections pass through standard 30 in. doorway.



REFRIGERATING CONDENSING UNITS—A complete line of reciprocating type condensing units for use with "Freon" refrigerants. Air-cooled models available from 1/6 hp to 3-hp. Water cooled models available from 1/2 hp to 125 hp. Motor compressor units for use with evaporative condensers or cooling towers available from 3 hp to 125 hp.

Hastings Air Conditioning Co., Inc.

Hastings, Nebr.

Manufacturers of

Air Conditioners.

Unit Heaters.

Utility and Package Blowers.

Dealers and Representatives in Principal Cities

A Complete Line of Highly Specialized Air Conditioners For DX, Cold City Water, Chilled or Well Water Operation. All equipment may be used for combination cooling and heating. Watercoils are six rows deep and DX coils four rows deep. Developed and designed for utmost efficiency. Constructed of copper tubing expanded and metallically bonded to pure copper fins. FLOWMETERS (to visually control flow) are standard on all water equipment.

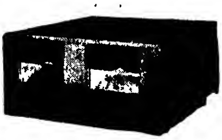
FLOOR MODELS

Floormasters—Unusual design and special features permit maximum installation possibilities with excellent results.

Cooling Capacity—
Water coils, 3 to 6 tons.
DX coils, 5 tons.
Air Delivery—2240 cfm.
Motor—½ hp Filters—three 16x25. Dimensions—Ht. 93 in., Width 48 in., Depth 25 in.

Royal—For offices, homes, hospitals, etc.
Cooling Capacity—
water, 1 to 2 tons.
1/6 hp 590 cfm Filter 16x25. Dimen. Ht. 40 in., Wd. 28., in., Depth 20½ in.

CENTRAL PLANTS



Sectional construction for ease of handling. Motors inside mounted to provide very neat appearing compact units.

SPECIFICATIONS

Size	CFM	Motor Hp	Filters	Capacity Tons
CP 30	3,000	1	5	4- 9
CP 40	4,000	1	8	6-12
CP 60	6,0C0	2	10	9-18
CP 80	8,000	3	12	12-24
CP 120	12,000	5	20	18-36

STEAM UNIT HEATERS

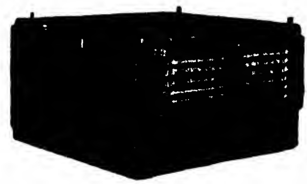
Centrifugal Type for extreme quietness and efficiency.

Steam pressure—to 150 lbs per sq in.

Finish—Brown wrinkle enamel and stainless steel louvers.

GENERAL UTILITY MODELS

Master—Singly or in multiple are suitable for any business or space size. Large jobs handled without duct work by proper location of units.



Cooling Capacity—
Water coils, 3 to 6 tons.
DX coils, 5 tons.
Air Delivery—2,240 cfm.
Motor—½ or ¾ hp. Filters—four 16x23. Dimen. Ht. 29 in., Wd. 49 in., Depth 50 in.

Majestic—Similar to Master except size. Cooling Capacity—Water 1½ to 3 tons. ¼ hp 1120 cfm Two 16x25 filters. Dimensions—Ht. 26 in., Wd. 28 in., Depth 40 in.

GAS UNIT HEATERS

These CENTRIFUGAL gas unit heaters present many fine and unusual features. *AGA* approved.

Squirrel-cage blowers provide SILENT operation and permit air delivery thru duct systems up to ¼ in. S.P.

Stainless steel ribbon burners result in quiet efficient combustion.

Dual directional, individually adjustable stainless steel louvers permit complete control of air delivery.

Write for Catalogs, Literature, or Information

Niagara Blower Company

General Sales Office: 405 Lexington Ave. **New York 17, N. Y.**

CHICAGO—5: 37 W. Van Buren St. BUFFALO—7: 673 Ontario St. SEATTLE—4: 705 Lowman Bldg.
District Engineers in Principal Cities

Over 30 Years' Experience in Industrial Air Conditioning, Liquid Cooling and Air Drying

NIAGARA AERO HEAT EXCHANGER
For cooling industrial liquids, water, oils, solutions, chemicals, compressed air and gases, with Niagara "Balanced Wet-Bulb" temperature control to improve efficiency and obtain precise results. Patented (U. S. Nos. 2,296,946 and R. I. 22,553). Ask for Bulletin 96.

NIAGARA AIR CONDITIONING SYSTEMS
For human comfort and for all industrial applications requiring controlled conditions of temperature, relative humidity, air purity and air movement.

NIAGARA AIR CONDITIONER, TYPE A
High precision apparatus using saturation to obtain control of R. H. to 1 per cent for laboratory work and control of hygroscopic materials. Ask for Bulletin 58.

NIAGARA AIR CONDITIONER, TYPE C
A year around air conditioning unit providing heating and humidifying or dehumidifying. Ask for Bulletin 80.

NIAGARA FAN COOLER AND DISK FAN COOLER
For comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products, fur storage vaults, etc. Bulletin 72.

NIAGARA SPRAY COOLER
For all cooling applications requiring high humidity or high capacity in small space. Ask for Bulletin 72.

NIAGARA "NO FROST" SYSTEM
Using Niagara "No Frost" Liquid in spray coolers, prevents frosting of cooling coils, automatically keeps spray solution at proper concentration, gives freedom from brine troubles, corrosion. Constant, efficient operation. Temperature to −100° F. Ask for Bulletins 95 and 105.

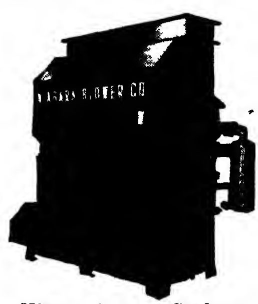
NIAGARA AEROPASS CONDENSER (Illustrated)
Saves power and water cost utilizing atmospheric air to remove heat of condensation. Patented Duo-Pass prevents scaling, saves power. "OILOUT" positively removes oil and dirt from refrigerant lines, assuring always full capacity. Balanced Wet Bulb Control assures operation of refrigeration plant at minimum head pressure regardless of weather or load conditions. Ask for Bulletin 103.

NIAGARA "DUAL" COOLERS
Simultaneously cools a room and furnishes chilled water as a refrigerant. Saves equipment cost, operating expense. Patented. Ask for Bulletin 70.

NIAGARA INDUSTRIAL LIQUID COOLER
Furnishes refrigerated water or aqueous solution in any quantity up to 220 gpm. Positive control of temperature regardless of load variation. Delivers "sweet" water at 33° F without danger of freezing damage. Ask for Bulletin 104.

NIAGARA FAN HEATERS AND HIGH PRESSURE STEAM FAN HEATERS
For heating and ventilating large areas. Units of the highest quality in engineering, material and workmanmanship. Ask for Bulletins 73 and 109.

NIAGARA MOTOR BLOWERS
One, two and three-fan units. High and low static pressure models. Ask for Bulletin 89.



Niagara Aeropass Condenser with "Oilout" and Bal[illegible] Wet Bulb Control

Parks-Cramer Company

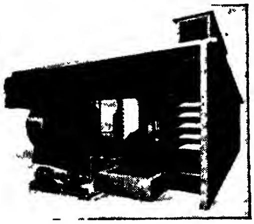
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying or De-humidifying, Air Changing, Refrigeration, Air Filtering, Air Washing

AUTOMATIC REGULATION

Merrill Process System of Hot Oil Circulation for Heating Industrial Materials Jacketed Cocks, Fittings and Jacketed Piping

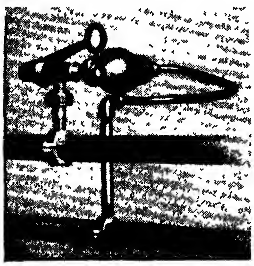


Central Station

Central Station Air Conditioning

A complete system or conditioning air, with positive circulation and controlled ventilation. One or more air washer and fan units. High humidifying and evaporative cooling capacity. Heating, filtering, and refrigerated cooling optional. Ducts with adjustable outlets distribute conditioned air uniformly. Slight air pressure also improves uniformity. No free moisture in room. Centralized maintenance.

Air Washer or Central Station Units.
Nozzles for Central Station Air Washers.



Turbomatic Humidifier

Turbomatic Humidifier

Efficient humidifier of the atomizer type. For direct humidification, as humidity boosters for Central Station systems of all makes. Self-cleaning, both air and water ports. Streamlined to prevent lint and dirt accumulation.

Parks Automatic Airchanger (not illustrated)

A patented system of forced air change used with a direct humidifying system. Insures fixed humidity and maximum evaporative cooling by controlling amount of air change and operation of humidifiers by a psychrometric humidity regulator. Designed for either complete new installations or for supplementing existing direct humidifying equipment.

Psychrostat

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Employs the principle of the Sling Psychrometer, used in all U. S Weather Bureau Stations. Hygrostat (not illustrated) where requirements are not so exacting. An Air Conditioning System is no better than its Regulation.

Pettifogger

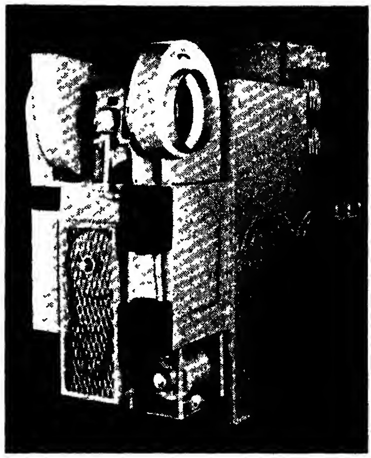
The Pettifogger

A compact humidifier for offices, stores, storerooms, laboratories, or other isolated departments. Self-contained in lacquered copper casing. Permanently though flexibly connected to water and electrical supplies. Automatic control. Adjustable capacity. Neutralizes drying effect of heating.

Pittsburgh Lectrodryer Corporation

Foot of 32nd Street Pittsburgh, Pa.

Small automatic air conditioning type LECTRODRYER used for providing lowered relative humidities.



This machine protects equipment in storage by maintaining a relative humidity of 35 per cent or lower.

FOR INDEPENDENT CONTROL OF DEHUMIDIFICATION IN COMFORT AND INDUSTRIAL AIR CONDITIONING

The results of years of experience in the independent control of industrial dehumidification are now available for comfort air conditioning in the form of sturdy, dependable, thoroughly tested machines for controlled adsorption dehumidification.

LECTRODRYER equipment using Activated Alumina, a solid adsorbent, is widely used in maintaining lower than normal relative humidities in the chemical, pharmaceutical and other industries.

In comfort air conditioning these machines handle the latent heat load with only the sensible heat load left for refrigeration or water cooling. With this type system, only the air needed for the sensible heat load is cooled and no reheat is required.

Machines are available for steam, gas or electric operation, whichever the purchaser specifies. Standard machines are available in several sizes ranging from 350 cfm upward.

LECTRODRYERS are shipped complete as self-contained automatic units in that they require no regular manual attention except for starting. They are built for continuous operation with reactivation being carried on simultaneously with the drying operation.

Write for full details.

United States Air Conditioning Corporation

Heating, Cooling Ventilating and Air Conditioning Equipment

usAIRco

For Industrial, Commercial and Residential Applications

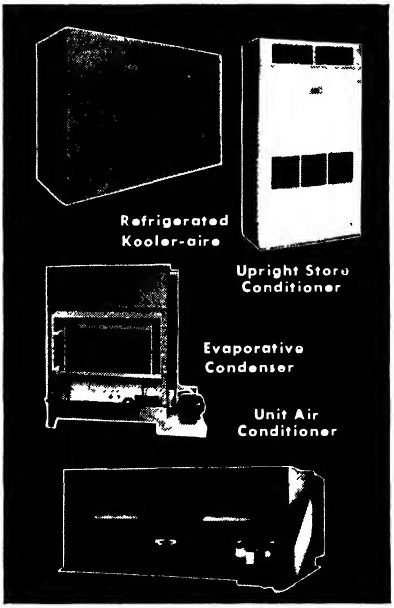
3377 COMO AVENUE SOUTHEAST MINNEAPOLIS 14, MINNESOTA

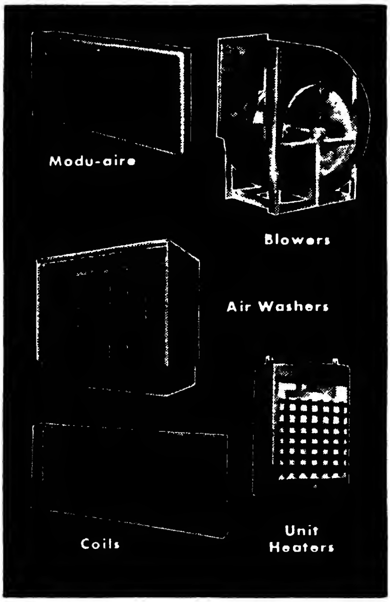
Refrigerated Kooler-aire—Unit combining cooling and dehumidifying units, refrigeration compressor and evaporative condenser all in one balanced assembly. 3 to 40 ton capacities.

Upright Store Conditioners—A completely packaged unit designed for conditioning applications in stores, restaurants, beauty shops. 3 and 5 ton capacities.

Evaporative Condensers—A cooling unit that condenses refrigerants, available capacities from 3 through 100 tons. Permits savings of 95 per cent in water costs, and by eliminating waste disposal problems, evaporative condensers provide substantial economy in cooling refrigerant gases.

Unit Air Conditioners—Units for year around air conditioning or for any combination of air conditioning functions. Made in both single and double fan arrangements, and in 9 sizes from 1000 cfm to 12,000 cfm. Units available with special coils for mechanical cooling or heating.





Modu-aire—Complete units for individual room air conditioning wherever central heating or cooling systems are available. No ducts required. Cooling or heating can be adjusted in individual rooms.

Blowers—Heavy and light duty blowers made in single or double inlet styles, and in a variety of sizes and capacities for all cooling, heating, ventilating and air conditioning requirements.

Coils—usAIRco makes a complete line of coils for every air conditioning requirement—including steam coils for heating, standard and non-freeze type, water coils for heating or cooling, and direct expansion coils.

Air Washers—Single and double stage 2,500 to 100,000 cfm for cleansing, cooling by cold water or refrigerant, humidifying or dehumidifying.

Unit Heaters—Suspension type small area heaters operating on steam, hot water or gas, are designed for supplementary heating in factories, garages, warehouses.

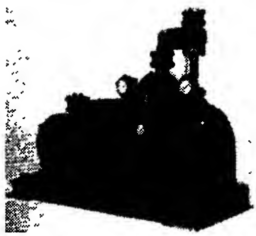
York Corporation

York, Pennsylvania

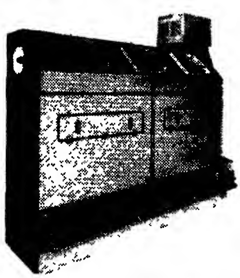
Factory Branches and Distributor Engineering and Sales Offices throughout the World.

Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for industrial processes or comfort requirements. Installations of unit and central systems in a complete range of capacities and types for every design requirement.

York Turbo Compressor



York V-W Condensing Unit



York Sectional Economizer

Yorkaire Unit Air Conditioner

Condensing and Water Cooling Systems—Turbo (centrifugal) brine and water cooling systems available over wide range of capacities—up to 1500 tons refrigeration for Freon-11 water cooling duty—suitable for steam turbine or motor drive.
Self-contained dynamically balanced, non-vibrating V/W type reciprocating compressors available in capacities up to 350 tons refrigeration in a single unit, with water cooled or economizer type condensers. Efficient automatic capacity reduction available for economical operation at reduced load.

The York Economizer—A combined force-draft cooling tower and refrigerant condenser, is available for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with Freon as the refrigerant are furnished, as standard, with a liquid sub-cooling coil. Economizers also designed for cooling of quench oil and other liquid coolants.

Air Conditioning Units: A complete line of finned coil, dry coil, wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round processing.

Yorkaire Unit Air Conditioner—A compact, self-contained model occupying but 21 x 42 inches of floor space and requiring only water, drain and electrical connections to operate. Special features provide utmost flexibility to meet varying conditions. Temperature dial control provides both automatic and manual temperature control. Air volume and motion may also be adjusted by a special control and the directional grille provides *directed air flow*—up, down or from side to side. May be used with ducts if desired.
Yorkaire Conditioners are ruggedly built, quiet in operation, equipped with standard fan and compressor motors for AC or DC.

Dehumidifiers—For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers are available in a full range of capacities for industrial installation.

Worthington Pump and Machinery Corporation

Air Conditioning and Refrigeration Division

General Offices: HARRISON, NEW JERSEY

Albany	Charlotte	El Paso	Louisville	Phoenix	San Francisco
Atlanta	Chicago	Fort Worth	Milwaukee	Pittsburgh	Seattle
Baltimore	Cincinnati	Galveston	New Haven	Portland, Ore.	Springfield, Mass.
Birmingham	Cleveland	Houston	New Orleans	Providence	Syracuse
Boston	Dallas	Kansas City	New York	St. Louis	Tulsa
Buffalo	Denver	Knoxville	Omaha	St. Paul	Washington, D. C.
Butte	Detroit	Los Angeles	Philadelphia	Salt Lake City	Wilmington, Del.

Representatives in all Principal Cities

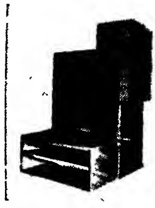
Self-contained Air Conditioners

Cooling (or heating, if desired), dehumidification, ventilation, circulation and air cleaning for commercial and small industrial applications. 3 and 5 ton capacities. Hermetic compressor, finned copper tubing condenser, large-surface finned cooling coils.

Freon Refrigeration Units

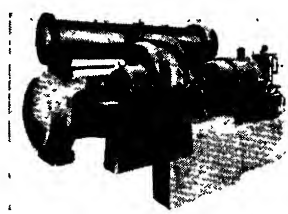
For all air conditioning and refrigeration applications up to 100 tons. *Series HS Compressors*—3 and 5 hp, vertical, two cyl.; 7½ hp four-cyl., V-type with Feather* Valves; splash lubrication. *Series HF Compressors*—10 to 100 hp, four-cylinder V-type and six-cylinder W-type; full force-feed lubrication. Supplied with mounted horizontal cleanable type shell and tube condenser or for use with Worthington Evaporative Condenser.

Air Conditioning Units



Series AHY and AVY Central Station Air Conditioners are for year-round air conditioning; heating coils and humidification can be added. AHY units for horizontal air flow, ceiling mounting; AVY vertical for floor mounting. 5 sizes from 2000 to 12,000 cfm, 4 to 62 tons. With or without internal face and by-pass dampers, also cooling coils.

Centrifugal Refrigeration Water Cooling Systems



Freon-11 centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive. 56 unit sizes . . . 150 to 1200 tons.

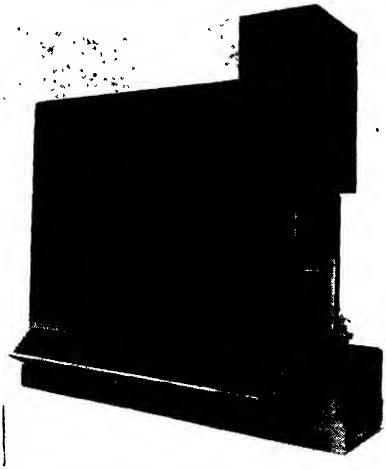
Packaged Air Conditioners

Series RCY for year-round air conditioning provides cooling, dehumidification, ventilation, cleaning and heating of air. RCY-300 has nominal capacity of 127,000 Btu/hr (10.5 tons) at *ASRE* rating and 3600–5100 cfm. RCY-500 has capacity of 204,000 Btu/hr and 5400–7600 cfm. Attractive styling, heavy steel frame with lightweight panels, balanced design, all parts accessible.

Unit Heaters and Ventilators

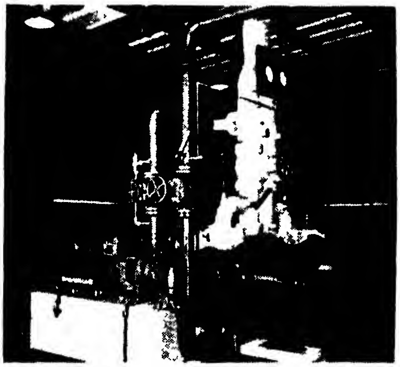
Series UHY unit heaters are blower type, for large spaces requiring uniform heating and good ventilation. Prevent stratification and condensate freezing. Can be arranged for automatically introducing outside air or regulating recirculated air. Five unit sizes, 16 arrangements, capacities from 100,000 to 1,200,000 Btu/hr with air volumes of 1500 to 12,000 cfm.

Evaporative Condensers



Series ECZ Freon-12 illustrated. 10 to 50 tons refrigeration. Units of sectionalized construction; all parts easily accessible. Galvanized steel coils for ammonia. Bare copper for Freon-12.

Horizontal Refrigeration Compressors

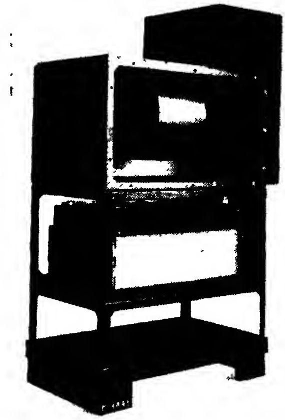


Horizontal Double-Acting Refrigeration Compressor in single or duplex types; for single or two stage compression. Capacities 50 to 1000 tons.

Condensers and Brine Coolers

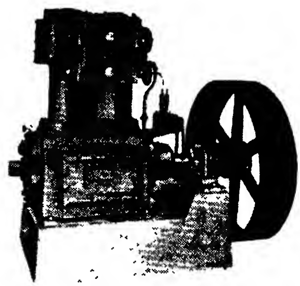
Multi-pass, as illustrated, for closed systems and space saving. Heads efficiently baffled to produce high liquid velocity. Vertical "Spira-Flo" types provided with circular water box and special water distributors. Cleanable without interfering with condenser operation.

Commercial Product Coolers



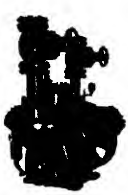
Both dry- and brine-spray types, in many sizes and arrangements for cooling, freezing and storage. Capacities from 2 to 15 tons, 2000 to 17,000 cfm.

Vertical Ammonia Compressors



Pressure-lubricated, roller main bearings; safety heads; patented Feather* Valves; belt drive or direct connected to electric motor, diesel or gas engine; sizes 3 x 3 to 10 x 10, 2-cylinder.

Booster Refrigeration Compressors



Multi-cylinder vertical single-acting for capacities up to 400 cfm. Horizontal double-acting for larger capacity requirements. Designed to produce and maintain low evaporator pressures required for low temperature applications in conjunction with compression or absorption refrigeration systems.

ALLIS-CHALMERS MANUFACTURING COMPANY

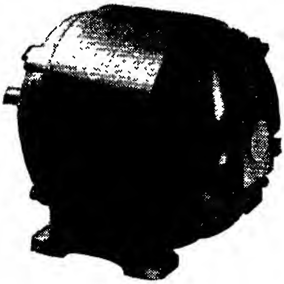
Milwaukee 1
Wisconsin

There is an Allis-Chalmers representative near you.

ALLIS-CHALMERS builds four major components for heating, ventilating and air-conditioning installations—in types and sizes for any requirement. A single source of supply means skilled sales service . . . coordinated installation . . . better unit responsibility. Bulletin 25B5170 describes Allis-Chalmers equipment for heating, ventilating and air-conditioning.

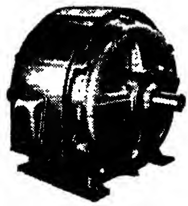
ALL-PURPOSE MOTORS

A complete line of ½ to 200 hp motors is built by Allis-Chalmers for any requirement of speed and torque. Strong, distortion-resistant construction adds to motor life.



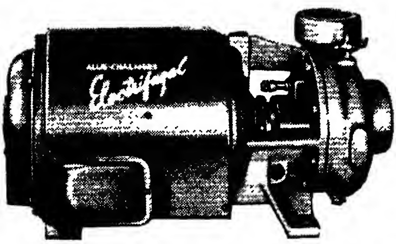
SQUIRREL-CAGE

WOUND ROTOR



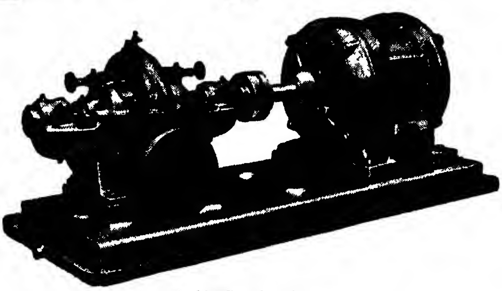
SYNCHRONOUS

CENTRIFUGAL PUMPS

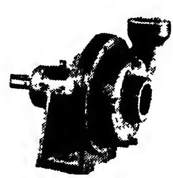


The Electrifugal

Motor and pump are built as one unit in the compact, efficient **Electrifugal** pump. Alignment is no problem. Has deep-groove ball bearings for all-position operation. Takes up ⅓ less space. 15 to 1600 gpm; heads to 500 ft. Bulletin 52B6059D.



Double Suction Pumps



Low Cost Pedrifugal

Motor Controls

Allis-Chalmers supplies motor controls for every requirement of operation. Reversing and non-reversing, manual or magnetic, across-the-line or reduced voltage.

Dry Type Transformers

A compact, quickly installed Class "B" insulated transformer that has proved practical in large heating and ventilating systems. Serves all circuits; all system loads.

SALES OFFICES

ALABAMA Birmingham, 3 ..... 1st Ave. & 20th St.
ARIZONA Phoenix ....45 W. Jefferson St.
CALIFORNIA Los Angeles, 13... 458 S. Spring St.
San Diego ....735 Bank of Amer. Bldg.
San Francisco, 7... ... . 650 Harrison
COLORADO Denver, 2... .. 1755 Glenarm Place
CONNECTICUT New Haven, 10... ..157 Church St.
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New York, 7 50 Church St.
Rochester, 4.........................89 East Ave.
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NORTH CAROLINA Charlotte, 2....212 S. Tryon St.
OHIO Cincinnati, 2 ....617 Vine St.
Cleveland, 14 ....E. 9th & Superior Sts.
Toledo, 4................... Summit at Madison
Youngstown, 3 ....26 Market St.
OKLAHOMA Oklahoma City....901 W. Grand Ave.
Tulsa, 3........................320 E. Archer St.
OREGON Portland, 4............920 S. W. 6th Ave.
PENNSYLVANIA Phila., 3 ... .1617 Pa. Blvd.
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Wilkes-Barre................. Market & Franklin
York .... 42 East King St.
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TENNESSEE Chattanooga, 2. ..... 737 Market St.
Knoxville, 2...................... 531 S. Gay St.
Memphis, 3 ....46 N. Third St.
TEXAS Amarillo................. .Amarillo Bldg.
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El Paso.. . ... Corner Oregon & Mills Sts.
Fort Worth, 1 . ... 407 West 7th St.
Houston, 3 .. .. 1719 McKinney Ave.
San Antonio, 5 . Commerce and Main Ave
Beaumont . . .. 1001 Pearl St.
UTAH Salt Lake City, 1 .... 134 S. Main St.
VIRGINIA Richmond, 19 527 East Main St.
WASHINGTON Seattle, 1. .. ... 1318 4th Ave.
Spokane, 8 .... West 428 Riverside
WEST VIRGINIA
Charleston, 1. . . . Cor. Summers & Lee
Wheeling 500 Hawley Building
WISCONSIN Milwaukee, 2..... 715 N. Van Buren S

Distributors throughout the World

ALLIS-CHALMERS TEXROPE DRIVES

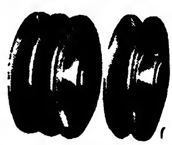
Texrope is the trade name and registered trade mark of V-belt drive products of Allis-Chalmers, pioneer in multiple V-belt drives. Texrope drives are compact, silent, easy on bearings . . . offer speed ranges of 7.1 . . . permit use of lower cost higher speed motors. Duplex-sealed cover protects Texrope V-belt internal cords against dirt, grit, moisture. Oil-Resisting, Static-Resisting, and Oil-Proof types. Bulletin in 20B6051G.

Texsteel Sheaves

Pressed steel construction. Fractional to 25 hp use.

Texdrive Sheaves

Integrally bushed for standard shafts. Of fine grain cast iron.



Adjustable Sheaves

50 per cent speed change with movable threaded plate.

Vari-Pitch Sheaves

9 per cent to 28 per cent speed control for 1 to 300 hp.

Wide Range Vari-Pitch

Speed ranges up to 116 per cent, with wide section V-belts.

"MAGIC-GRIP" SHEAVES

Easy To Put On—Easy To Take Off

Sheave slides on shaft without forcing. *Easy "slide on" results in true alignment.* *Tightening 3 capscrews locks sheave to shaft.* *Use 2 capscrews as jackscrews in tapped holes.* *A twist of the wrist breaks bushing grip.* *Slide sheave off . . . no prying necessary.*

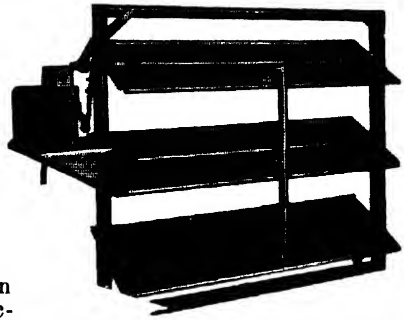
American Foundry and Furnace Co.

General Offices: Bloomington, Illinois
P. O. Box 904

Sales & Engineering Offices in Principal Cities

Auburn, Ind.	Lige Warm. & Vent. Co.	New Orleans	R. K. Rothrock
Boston	Thermal Equipment Co.	New York City	The Demuth Co.
Chicago	Gillespie-Dwyer Co.	Oakland	Aladdin Heating Corp.
Cincinnati	Walter A. Juergens	Philadelphia	Amer. Htg. & Vent. Co.
Cleveland	Amer. Warm. & Vent. Co.	Richmond, Ind.	Harry Hoff S.M. Works
Denver	Kenneth Stegemiller	Richmond, Va.	Richmond Equip. Co.
Des Moines	Iowa Kol-Master Corp.	St. Louis	M. F. Carlock
Elmira, N. Y.	Amer. Warm & Vent. Co.	St. Paul	Amer. Fdry. & Fur Co.
Grand Rapids	R. T. Shuttleworth	Salt Lake City	Williams Gritton & Wilde
Indianapolis	James H. Elliott & Assoc.	San Antonio	Langhammer - Rummel
Kansas City, Mo.	John H. Kitchen & Co.	Toledo	Amer. Warm. & Vent. Co.
Kenmore, N. Y.	Murton J. Rodman	West Lafayette, Ind.	F. H. Speaker & Son
Los Angeles	Harry F. Haldeman, Inc.		
Milwaukee	Amer. Fdry. & Fur. Co.		

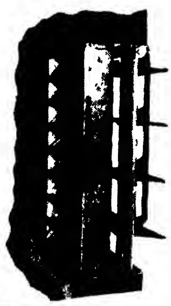
F-12 LOUVER DAMPER



Made to fit any size opening. Adaptable to automatic or hand control. Blades of 16 gage steel. Channel frames 2 x ½ x ⅛ in. standard except in large sizes—optional ¼ x 2 in. iron bar frame. Standard steel painted aluminum—optional galvanized iron. Ball bearing blade pivots standard—optional brass trunions. Made for vertical or horizontal installation. For industrial plants, powerhouses, hotels, schools, theatres, etc.

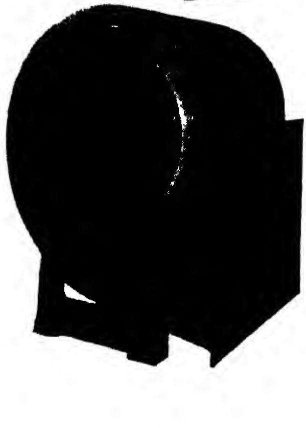
When F-12 is ordered with blades longer than 48 inches, dampers are made in multiple sections operating in unison. Motor brackets for internal or external mounting at extra charge. Motors and connecting linkage furnished by others unless specific arrangements are made. Standard is as illustrated with adjustable extended shaft.

S-454-F COMBINATION STORMPROOF LOUVER and DAMPER



Consists of galvanized iron frame with 26 gage galvanized iron stationary horizontal stormproof louver blades riveted securely to outside frame. Apron extends over sill. Back of stormproof louver is No. 16 mesh, rust-proofed, insect screen in "U" type removable frame. Back of screen is multiple-blade ball bearing louver damper—similar to F-12 but with off-center axle—to control volume of air admitted. Louver damper blades of 16 gage steel galvanized. Frame of 2 x ½ x ⅛ in. galvanized channel iron. Dampers can be automatically or manually controlled. Blades all work in unison. Made to fit opening size specified. Standard for 8 in. deep wall. Entire assembly or any part can be furnished made of aluminum, copper or stainless.

SUPERIOR BLOWERS



Forwardly Curved Multiblade Type
Heavy Duty Construction
Made in Single and Double Widths with wheel diameters ranging from 10 to 65 in., in 5 in. increments. Capacity Range: 800 to 105,000 cfm

HEAVY DUTY HORIZONTAL HEATERS

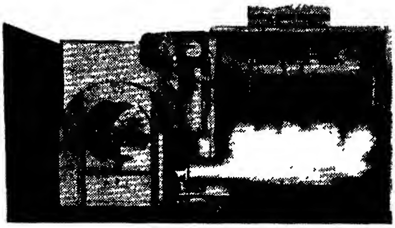
Made of heavy cast iron for long, dependable service. Made in sections for easy assembly, or replacement of parts. Integrally cast fins for additional strength and heating surface. Direct Fired, with long fire travel for minimum fuel consumption. Designed to relieve internal stresses set up by heating and cooling. Licensed Engineer not required to operate. Reduces hazards of explosions, and damage to equipment from freezing.

CENTRAL TYPE FORCED WARM AIR

For schools, churches, theatres, auditoriums, factories, drying plants, etc.
For hand fired coal, stoker fired coal, oil or gas. Heats and ventilates with same system—uses part outside air to maintain air quality. Air Filters, Automatically Controlled Humidifier, and Automatic Temperature Regulation available as optional equipment.
Output Capacity Range per heater:

Hand Fired Coal	- 278,000 to 1,365,000 Btu per hr.
Stoker Fired Coal	- 278,000 to 2,074,000 Btu per hr.
Oil or Gas	- 278,000 to 3,580,000 Btu per hr.

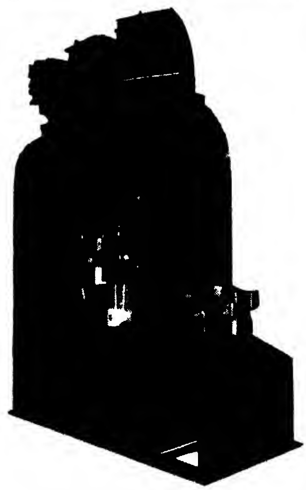
Two or more heaters may be placed in one setting to provide any desired output capacity.
Special models for coal, stoker coal, oil and gas; New Convertible Heater available which is convertible from one fuel to another. (See picture top of page.)



UNIT HEATER TYPE FORCED WARM AIR

For Industrial Buildings, Warehouses, Factories, etc. For oil, gas or stoker fired coal. Use for heating and ventilating, or for tempering outside air supplied to replace air exhausted. Each unit is complete heating plant. Induced Draft Fan optional. Output Capacity Range per Unit:

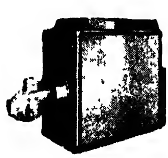
Stoker Fired Coal	- 669,000 to 2,028,000 Btu per hr.
Oil or Gas	- 440,000 to 3,580,000 Btu per hr.



DOMESTIC HEATING EQUIPMENT

June-Aire Gas Fired

June-Aire Oil Fired



June-Aire Oil Fired

June-Aire Vertical Gas Fired

BRYANT HEATER COMPANY

17825 St. Clair Avenue - - - Cleveland, Ohio

Let the pup be furnaceman

Engineering, Sales and Installation Information on Bryant Equipment available through Bryant Distributors, Dealers and Gas Companies in principal cities.

Forced Warm-Air Furnace (Vertical Type)

Steam and Water Boilers

Bryant Gas designed boilers include tubular cast iron sections, ribbed lower tubes, large steam liberating areas, all heating surfaces readily accessible for cleaning. Insulated metal covers and Bryant gas controls. Complete range of *AGA* inputs from 67,500 to 3,996,000 Btu/hr for steam and hot water heating systems, volume water heating and industrial process.

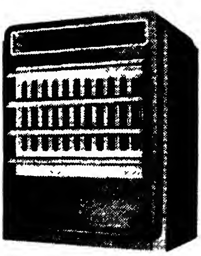
Bryant Forced Warm-Air Gas-Fired Furnaces complete with blowers, humidifiers and air filters are compactly designed for both small and medium sized housing, and for offices or industrial use. Models are available with heating sections made of either tubular cast iron or 12-gage steel. All are *A.G.A.* approved with Bryant automatic controls.

Vertical Type—suitable for almost all installations; especially for utility rooms and closets where floor space is limited. Overall height approximately five feet. Capacities range from 45,000 to 145,000 Btu/hr input.

Horizontal Type—low in height, best suited for low-ceiling installations. Available in sizes 60,000 to 375,000 Btu/hr input.

Forced Warm-Air Furnace (Horizontal Type)

Bryant suspended type Gas-Fired Unit Heaters available in nine sizes ranging from 65,000 to 255,000 Btu/hr *AGA* inputs. Efficient heat exchanger of vertical tube construction. Available in either cast iron combustion chamber, alloy steel tube or all steel types. Quick, clean, efficient heat for all types of industrial and commercial space. Flexible, automatic control and large volume air circulation produce ideal space heating results.



Suspended Type Unit Heater

Bryant Air Dryers with rotary silica gel drum, is completely automatic in operation and finds application for exact humidity control in industrial processing, comfort air conditioning and the drying and storing of hygroscopic materials. Available in air capacities ranging from 800 to 15,000 cfm. Standard units are reactivated by gas. Indirect units arranged for use with high pressure steam coils or electric strip heaters are available.

Dehumidifier

See your local Bryant Distributor or write for complete details and specifications.

DUO-THERM *Division of Motor Wheel Corporation*

Lansing 3, Michigan

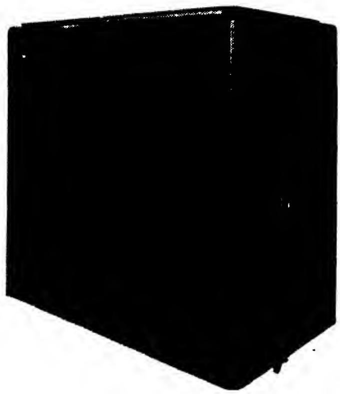
Duo-Therm Automatic Fuel Oil Furnaces are complete package unit cabinet models in baked enamel. They are designed to fit any type home of average size. Can be installed with ease in the basement or in the utility room. Listed as Standard by the *Underwriter's Laboratories* and listed by the *Canadian Standards Association*—for use with No. 1 or No. 2 fuel oil. Write for Engineering Manual on Duo-Therm Oil Burning Furnaces.

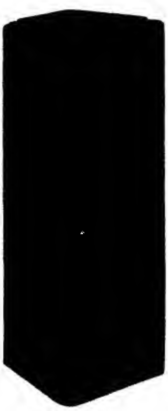
All models listed below tested and rated in accordance with commercial standard CS104, U. S. Department of Commerce at .06 draft an with CS12-48 No. 1 oil.

DELUXE MODEL WINTER AIR CONDITIONER

340-DB-78,000 Btu/hr gross capacity-1,000-1,700 cfm blower output.

350-DB-108,000 Btu/hr gross capacity-1,500-2,600 cfm blower output.





UNDERNEATH BLOWER MODEL WINTER AIR CONDITIONER

340-UB-75,000 Btu/hr gross capacity-1,000-1,700 cfm blower output.

GRAVITY MODEL FURNACE can be manually controlled or be equipped for automatic thermostatic control. Can easily be converted to either DeLuxe or Underneath blower types. Available in 3 sizes as follows:

339-M—73,100 Btu/hr gross capacity.
340-T

349-M—104,000 Btu/hr gross capacity.
350-T

CAMPBELL "WINTER-CHASER" AIR CONDITIONING SYSTEM

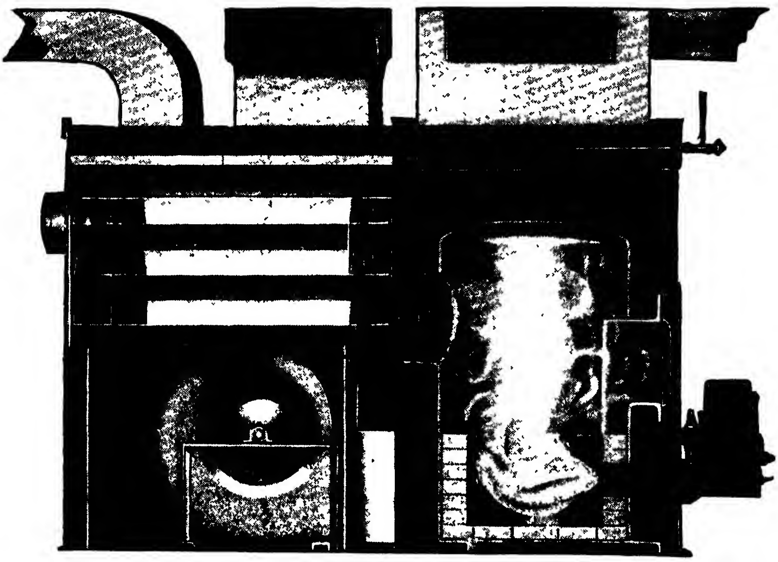
ENGINEERING SERVICE

Our Engineering Department consists of trained graduate engineers backed by 60 years of practical experience. We will be glad to help solve any heating or ventilating problems or help with layouts and specifications for churches, schools, garages or any large building.

GUARANTEE

If the duct system is designed or approved by our Engineering Department, and heater and blower are furnished by us and are according to our ratings, we will guarantee any heater for ten years against repairs from any cause and will guarantee the heating of all rooms to which warm air is delivered to 70° in the coldest and windiest weather. The motor, humidifier, automatic burner and controls, and other parts made by others carry their manufacturers' guarantee.

GAS OR OIL FIRED

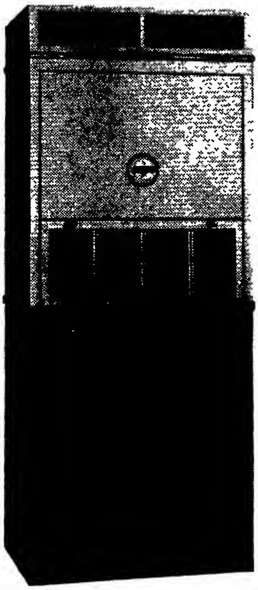


No.	Heat Loss of Bldg., Btu per hr.	Output at Registers, Btu per hr.	Heating Surface, sq. ft.	C.F.M. for 135° Register Temp.	Motor Size, H.P.	Gals. Oil per hr.	Gas or Oil Input, Btu per hr.	No. of Filters and Size, inches	Casing Size, inches L	W	Ht.
1866	100,000	112,000	66	1400	1/6	1	140,000	4-16 x 20	78	41	52
18100	150,000	168,000	100	2100	1/4	1½	210,000	4-20 x 20	86	41	57
27133	200,000	224,000	133	2800	1/4	2	280,000	6-16 x 25	87	50	61
30166	250,000	280,000	166	3500	1/3	2½	350,000	6-16 x 25	92	60	74
30200	300,000	336,000	200	4200	1/3	3	420,000	9-16 x 25	97	60	74
30233	350,000	392,000	233	5000	1/2	3½	490,000	12-20 x 20	104	80	72
30250	400,000	448,000	250	5600	1/2	4	560,000	12-20 x 20	104	80	72
For churches, schools, or buildings where rapid temperature raising is necessary											
133D	200,000	290,000	133	3700	1/3	3	365,000	6-16 x 25	87	50	61
166D	250,000	365,000	166	4600	1/2	3¼	455,000	6-16 x 25	92	60	74
200D	300,000	440,000	200	5500	1/2	4	550,000	9-16 x 25	97	60	74
233D	350,000	510,000	233	6400	3/4	4½	635,000	12-20 x 20	104	80	72
250D	400,000	580,000	250	7300	3/4	5¼	730,000	12-20 x 20	104	80	72
255D	450,000	660,000	255	8300	1	6	820,000	12-20 x 20	104	80	72

Campbell Heaters are guaranteed to deliver full rated capacity.

CHRYSLER AIRTEMP

AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON 1, OHIO



"PACKAGED" AIR CONDITIONERS

AVAILABLE IN 3 and 5 H. P.

COMPLETE—Cools, dehumidifies, filters, circulates the air. Free air discharge or duct distribution. Heating coil for year 'round service, optional.

COMPACT—Everything enclosed in a rust-resistant "Bonderized" cabinet of modern design with chrome hardware and trim. Occupies minimum of floor space—4.7 sq ft for 3 hp and 6.5 sq ft for 5 hp.

NEW AIRFOIL GRILLE—The modern Airtemp grille is fully adjustable for horizontal and vertical air flow. The grille vanes are made of extruded aluminum for streamlining and beauty.

EASILY, QUICKLY INSTALLED—Tested and completely assembled at the factory. Needs only three connections; electric, water and drain.

SEALED RADIAL COMPRESSOR—Quiet, all moving parts balanced to eliminate vibration. Entire compressor assembly suspended from single rubber mounting. Long life because vital moving parts are "super-finished" and pressure lubricated.

FLEXIBLE—"Packaged" Air Conditioners can be installed singly or in multiple to meet almost any requirement. Can be moved easily.

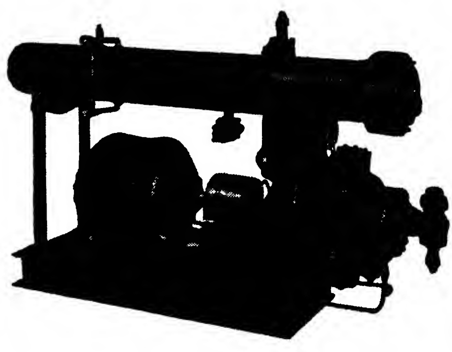
CHRYSLER AIRTEMP RADIAL CONDENSING UNITS

AVAILABLE IN 10 TO 75 HORSEPOWER CAPACITIES

These heavy-duty Radial Condensing Units, for use with Freon, are especially adapted for refrigeration, for industrial processes or air conditioning. Airtemp Radial compressors are direct connected and have force-feed lubrication. The automatic capacity-reduction device gives variable capacity . . . high operating efficiency. Light in weight and economical to operate, these condensing units are shipped ready to run. Vibrationless, they are very easy to install because special foundations are not necessary. Each pressure vessel approved by *Underwriters Laboratories*. Compressor units are also available for use with evaporative condensers.

AUTOMATIC UNLOADER

The automatic cylinder unloading device permits starting the compressor under no load and keeps the compressor automatically adjusted to varying loads with no stopping and starting during operation.



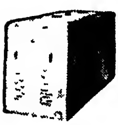
Automatic Capacity Regulation ● Unloaded Starting ● Direct Connected ● Simplified Installation ● Compact Design ● Practically No Vibration ● No Special Foundations Needed ● Interchangeable Parts.

CHRYSLER AIRTEMP
AUTOMATIC HOME HEATING

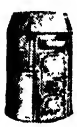
VAPORIZING-OIL-BURNING AUTOMATIC FURNACE . . . Models for forced-air—56,000 Btu output; gravity, 50,000 Btu output. Sure-Draft Fan assures highest overall efficiency. "Bonderized" and insulated jacket. Forced air model approved for closet installation, *Underwriters' Laboratories, Inc.*

OIL-FIRED AUTOMATIC FURNACE . . . Heats, humidifies, filters and circulates the air. Five models, from 70,000 to 160,000 Btu output. "Bonderized" and insulated jacket. Metal combustion chamber, seam-welded firebox of copper-bearing steel; large, slow-speed, rubber mounted fan. Airtemp oil burners on all models. Approved by *Underwriters' Laboratories, Inc.*

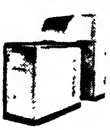
GAS-FIRED AUTOMATIC FURNACE . . . Heats, humidifies, filters and circulates the air. Steel Models 80,000 to 160,000 Btu output. Cast-iron Models 50,000 and 75,000 Btu output. "Bonderized" and insulated jacket. The Airtemp "Silent Flame" Gas Burner starts, stops and operates quietly, has many exclusive features—no popping or flash-backs. Approved, *A.G.A.* Laboratories.



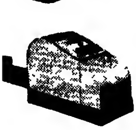
OIL-FIRED STEEL BOILERS . . . Three models—BLF-110 rating 460 sq ft EDR, BLF-165 rating 690 sq ft EDR, BLF-220 rating 920 sq ft EDR, of steam at the boiler header. Combustion chamber of quality chrome steel. Boiler is combination of "Scotch Marine" and Locomotive types. Complete with Airtemp burner and all controls.



COAL-FIRED FURNACE, GRAVITY . . . Furnace body of heavy steel boiler plate—seams electrically welded. Front, heavy gauge pressed steel. Fire brick lining. Sizes 22 in., 24 in., and 27 in.; Btu at bonnet, 108,000, 119,700, and 154,200.



COMBINATION HEATING AND COOLING FOR THE HOME . . . Combination of a 3 hp or 5 hp Chrysler Airtemp "Packaged" Air Conditioner and any of the larger Chrysler Airtemp automatic furnaces. The same blower, filters and ducts of the automatic heating system are employed for cooling in the summer.



STOKERS . . . Domestic, available in both hopper and bin-feed models for anthracite or bituminous coal for burning 15 to 35 lb per hour. Powerful transmission, sectional retort and safety coupling . . . eliminates shear pins.

CONVERSION OIL BURNERS . . . Pressure-atomizing oil burner, 1.35 to 4.5 gallons No. 3 furnace oil per hour. All Airtemp Oil Burners are approved by *Underwriters' Laboratories, Inc.*

THE MEYER FURNACE COMPANY
PEORIA, ILLINOIS

Manufacturers of Heating and Air Conditioning Equipment for Coal, Gas and Oil Burning

MEYER OIL AND GAS FIRED EQUIPMENT

"A" Series Oil Fired

"F" Series Gas Fired

"B" Series Hi-Boy

"K" Series Oil Fired

The MEYER oil-fired air conditioners are available in three types—the "A" Series suitable for basement installation in two sizes with outputs from 110,000 Btu to 165,000 Btu. The "B" Series Hi-Boy suitable for either basement or first floor installation in basementless houses available in two sizes with capacities of 93,500 Btu to 110,000 Btu. The "K" Series oil-fired air conditioners made in two sizes with outputs of 203,000 Btu and 294,000 Btu suitable for larger spaces.

"A" SERIES OIL FIRED AIR CONDITIONERS

Built for Fine Homes. Compact in Size. But Highly Efficient

Furnace Number	Input Btu/Hr	Gph	Output at Bonnet	Cfm	Fan Size	Mtr. Hp	No. & Size of Filters	Vent Diam.	L	W	H	W. A. Plenum	R. A. Plenum
A-100	140,000	1	110,000	1200	10	¼	2 - 20x20x2	6	71	25	48	20x21	20x21
A-150	210,000	1.5	165,000	1800	12	¼	4 - 16x20x2	8	81	32	48	26x28	24x28

"B" SERIES OIL FIRED HI-BOY

This High-Boy Handles Small Homes and Buildings Using A Minimum of Floor Space

B-85	119,000	.85	93,500	1000	10	¼	1 - 25x20x1	6	28	26	65	22x24	
B-100	140,000	1.0	110,000	1200	10	¼	1 - 25x20x1	6	28	26	70	22x24	

"K" SERIES OIL FIRED AIR CONDITIONERS

A Heavy Duty Type Unit That Has A Fine Background of Successful Service

K-175	258,000	2	203,000	2690	15	½	4 - 20x20x2	9	46	79	55	35x38	40x26
K-250	368,000	3	294,000	3890	15	¾	4 - 20x20x2	9	52	82	55	41x41	40x26

The MEYER gas-fired air conditioner as illustrated above is available in five sizes from 110,000 Btu per hour to 495,000 Btu per hour input capacities.

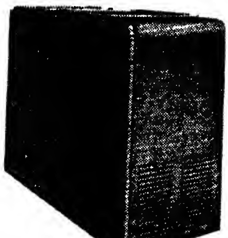
"F" SERIES GAS FIRED AIR CONDITIONERS

These Units are Unusually Compact and Can Be Equipped For Manufactured, Natural, or Liquified Petroleum Gas

Furnace Number	Input Btu/Hr	Output at Bonnet	Cfm	Fan Size	Mtr. Hp	Vent Diam.	No. & Size of Filters	L	W	H	W. A. Plenum	R. A. Plenum
F-10	110,000	88,000	960	10"	¼	5	2 - 20x20x2	65	25	48	20x21	20x21
F-10	165,000	132,000	1440	12"	¼	6	4 - 16x20x2	69	32	48	20x28	20x28
F-20	220,000	176,000	2400	2-10"	⅓	2-5	4 - 20x20x2	65	46	48	20x42	20x42
F-30	330,000	264,000	3600	2-12"	½	2-6	6 - 20x20x2	69	60	48	20x56	20x56
F-45	495,000	396,000	5400	3-12"	¾	3-6	8 - 20x20x2	69	88	48	20x84	20x84

L. J. Mueller Furnace Co. - Milwaukee 7, Wis.

HEATING MUELLER Climatrol REG. U. S. PAT. OFF. **AIR CONDITIONING**



Type "102"

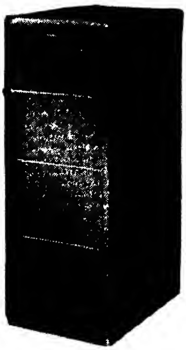
This gas-fired winter air conditioner is of steel sectional construction, with the blower compartment at the rear of the unit. It is available in two sizes with *A.G.A.* input ratings of 180,000 and 225,000 Btu per hour.

Type "107"

Gas-fired cast iron winter air conditioner for basement or utility rooms. Furnished in three sizes with *A.G.A.* input ratings of 60,000, 90,000 and 120,000 Btu per hour. Also available for gravity operation as Type 106.

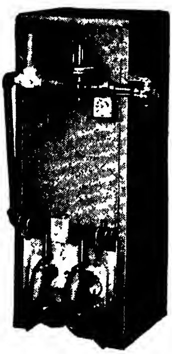
Type "109"

Gas-fired winter air conditioner for basement or utility room installation. Available with *A.G.A.* inputs of 67,500, 100,000 and 135,000 Btu. Convertible to oil firing with either vaporizing or pressure-atomizing burners.



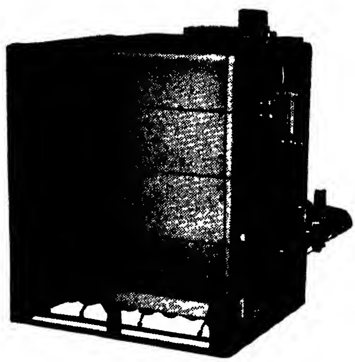
Type "10"

Gas-fired boiler, with enclosed controls, used for residential installation. It has *A.G.A.* ratings of 290 to 2015 sq ft for water and 180 to 1260 sq ft for steam. Approved for LP-Gas.



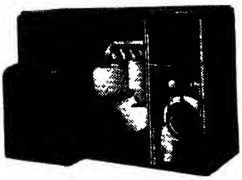
Type "11"

Gas-fired boiler, with exposed controls, used for home installation. It has *A.G.A.* ratings of 290 to 2015 sq ft for water and 180 to 1260 sq ft for steam. Also approved for LP-Gas.



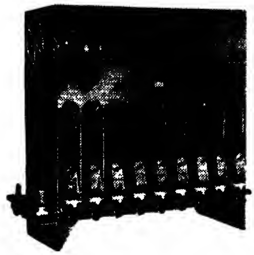
Type "20"

Gas-fired boiler used primarily for large installations. It has *A.G.A.* ratings of 1010 to 20,160 sq ft for hot water and 630 to 12,600 sq ft for steam. Both this boiler and the Types "10" and "11" can be equipped for both direct and indirect water heating applications.



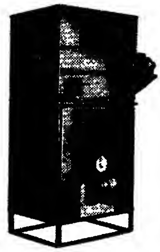
Type "901"

Summer air conditioner available in 3- and 5-ton sizes. Shown installed in Type 105 gas-fired furnace, which has *A.-G.A.* inputs of 100,000 and 150,000 Btu.



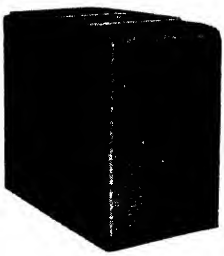
Type "UH"

Gas-fired unit heater for space-heating requirements. Available with *A.G.A.* inputs of 67,000 to 540,000 Btu per hour.



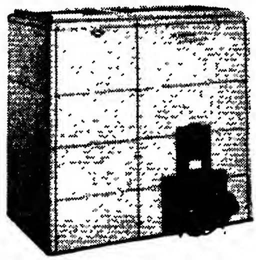
Type "OH-57"

Oil-fired unit heater used primarily for commercial and industrial applications. Available with outputs of 54,000, 80,000, 108,000 and 250,000 Btu per hour.



Type "50"

Oil-fired winter air conditioner with outputs at register of 100,000 to 225,000 Btu, with selection of blower sizes. Fired by pressure-atomizing burner. Welded steel heat exchanger.

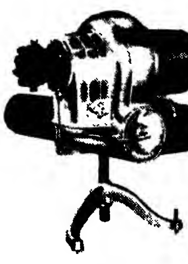


Type "202"

Oil-fired winter air conditioner available with inputs of 100,000 and 150,000 Btu. Unit may be converted to gas. Also available for gravity operation. Fired by vaporizing or pressure burners.

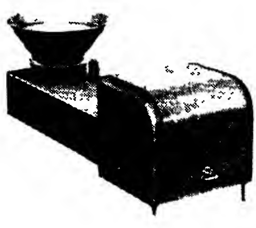
Type "209"

Oil-fired winter air conditioner with 100,000 and 135,000 Btu inputs with pressure burner; 67,500, 100,000 and 135,000 Btu with vaporizing burner. Can be converted to gas firing.



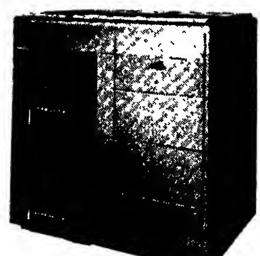
Type "450"

Oil-fired conversion burner of the pressure-atomizing type. Burner is available with inputs of 1 gallon through 5 gallons of oil per hour.



Type "500"

Gas-fired conversion burner available with *A.G.A.*-listed maximum inputs of 125,000, 175,000, 225,000 and 275,000 Btu per hour.



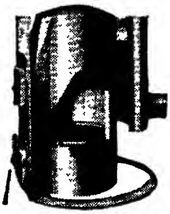
Type "702"

Steel coal furnaces manufactured in 20, 22, 24, 27 and 30 in. sizes. Type "FB" cast iron furnaces made in 20, 22, 24, 27 and 30-in. sizes.

The Waterman-Waterbury Co.

Minneapolis 13, Minnesota

Manufacturers of WATERBURY Coal, Oil and Gas Fired Furnaces and Air Conditioning Equipment



WATERBURY GAS-TITE FURNACE 700 Series

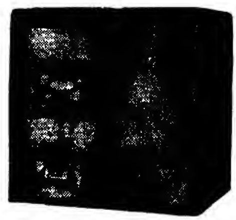
The welded steel furnace body keeps dust, smoke and fumes out of the air stream. A large combustion chamber and radiator with long fire travel insure efficiency and economy.

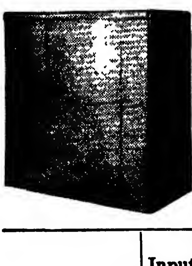
Also furnished in a complete home Air Conditioner.

Size Furnace	Diameter: Furnace Body	Diameter: Pipe Casing	Height: Furnace Body	Height: Pipe Casing & Hood	Steel Thickness: Body	Steel Thickness: Head	Steel Thickness: Radiator	Diam. Smoke Pipe	Approx. Shipping Wt Lbs Furnace less Casing	Output Ratings: Sq In. Leader Pipe	Output Ratings: Btu at Bonnet
720	20"	39"	49"	64"	10 Ga.	10 Ga.	12 Ga.	8"	531	482	87,529
722	22"	41"	49"	64"	10 Ga.	7 Ga.	12 Ga.	8"	603	505	91,730
724	24"	44"	49"	64"	7 Ga.	7 Ga.	12 Ga.	9"	730	541	98,056
727	27"	46"	55"	70"	7 Ga.	¼"	12 Ga.	9"	866	672	121,856

Gastite Air Conditioner

Size	Width	Length	Height	Size Outlet Opening	Size Inlet Opening	No. of Filters	Size Filter	Blower C. F. M. Range	BTU Output
720-10	36"	50"	52"	28x32	16x32	2	16x20	375 to 1050	87,963
722-10	38"	54"	52"	28x34	20x34	2	16x25	375 to 1050	94,011
724-10	40"	58"	52"	30x36	22x36	2	20x25	375 to 1050	100,193
724-12	40"	58"	52"	30x36	22x36	2	20x25	645 to 1677	100,193
727-15	44"	61"	59"	26x36	22x36	2	20x25	922 to 2397	142,570





WATERBURY GAS FIRED AIR CONDITIONER

The latest achievement of Waterbury's engineers, designed specifically for gas. Completely automatic—filtered, humidified forced air—enclosed in a compact, baked enamel casing. Can also be furnished for Propane or Butane.

Size	Input Rating BTU Per Hr.	BTU Output	Blower C. F. M. Range	Width	Length	Height Over-all	Size Outlet Opening	Size Return Inlet Opening	No. of Filters	Size Each Filter	Flue Pipe Diameter	Size Gas Connection Required
6413A—9B	90000	72000	400 to 800	28"	59"	48"	23x24	16x24	2	16x25	5"	¾"
6413A—9HBB	90000	72000	400 to 800	28"	28"	67"	24x24	15x22	2	16x25	5"	¾"
6415A—10B	120000	97000	600 to 1200	28"	61"	53"	23x24	18x24	2	16x25	6"	¾"
6418A—12B	150000	120000	645 to 1677	34"	71"	53"	30x30	22x30	2	20x25	6"	1"
6426A—15B	180000	144000	1000 to 2800	55"	60"	50"	24x51	28x36	4	16x20	2-5 to 7	1"
6430A—18B	240000	192000	1232 to 3204	55"	60"	58"	24x51	28x36	4	16x20	2-6 to 8	1¼"

MASTER BLOWERTROL

This control is now standard equipment on all Waterbury units with a ½ hp or smaller blower motor. It gradually increases and decreases blower speed. With this device you have almost continuous air circulation during the heating season. Those units equipped with Blowertrol have the letter "B" following the unit number.

The Waterman-Waterbury Co.

Minneapolis 13, Minnesota

WATERBURY 6300 Series OIL FIRED AIR CONDITIONER

Completely automatic oil heat with forced, filtered, humidified air, all enclosed in one compact casing finished in baked enamel.

Size	Input Rating Gal Oil per Hr	Output Capacity Btu per Hr at Bonnet	Width Casing	Length Casing	Height Overall	Size Outlet Opening	Size Return Inlet Opening	Distance between Opening	No. of Filters	Size each Filter	Diam. Smoke Pipe
6313A-9B	¾ gal	72000	28″	59″	48″	23x24	24x16	2″	2	16x25	6″
6315A-10B	1 gal	97000	28″	61″	53″	24x23	24x18	2″	2	16x25	7″
6313A-9HBB*	¾ gal	72000	28″	28″	67″	23x24	22x15	..	2	16x25	6″
6315A-10HBB*	1 gal	97000	30″	30″	76″	26x26	22x15	..	2	16x25	7″

* *HB indicates a Hi-boy model*

WATERBURY GAS GRAVITY FURNACE

Size Furnace	Input Rating Btu per Hr	Output Rating Sq In. Warm Air Pipe	Width In.	Length In.	Height Overall In.	Size Outlet Opening	**Rec. Flue Pipe Diam.	Size Gas Connection Required In.
6413A-G	90000	67500	28″	28″	48″	24x24	5″	¾″
6415A-G	120000	90000	30″	30″	53″	26x26	6″	¾″
6418A-G	140000	105000	40″	40″	53″	36x36	6″	1″

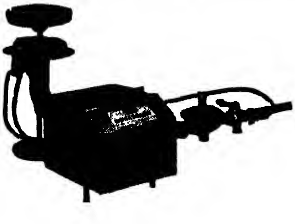
SEAMLESS OIL FIRED FURNACE BODY

Specially engineered for oil firing only, it may be used as a gravity circulating pipe furnace. It is the heating element of the Comfortrol Oil Fired Air Conditioners.

Specifications Comfortrol Oil Fired Air Conditioners

Size	Input Rating Gal per Hr	Btu Output	Normal Cfm Range	Width Less Front Hood‡	Length Over-all	Height	Size Outlet	Size Inlet	Filters No.	Filters Size	Smoke Pipe
1322-12B	1.5	164,177	645 to 1677	44″	72″	64″	36x36	18 x24	2 2	16x25 16x25	8
1322-15B	1.5	164,177	922 to 2397	44″	74″	64″	36x36	25⅛x34⅛	4	16x20	8
1324-15B	2.0	218,903	922 to 2397	48″	78″	64″	40x40	25⅛x34⅛	4	16x20	8
1324-18B	2.0	218,903	1232 to 3204	48″	81″	64″	40x40	40 x36	4	20x25	8
1327-18B	2.5	273,609	1232 to 3204	52″	85″	64″	44x44	40 x36	4	20x25	9
1327-21	2.5	273,609	1450 to 3770	52″	85″	64″	44x44	38 x44	6	16x25	9
1330-21	3.0	328,354	1450 to 3770	56″	89″	70″	48x48	38 x44	6	16x25	9
1333-21	3.5	383,080	1450 to 3770	60″	104″	72″	52x52	46 x48	6	20x25	9
1336-24	4.0	437,805	1968 to 5118	66″	110″	81″	58x58	46 x48	6	20x25	9

‡ Burner hood all sizes is 24 in. wide, 10 in. deep.



GAS CONVERSION BURNER

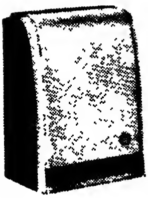
A real gas burner, the same as used in the Gas Air Conditioner. Can also be furnished for Propane or Butane.

Number	Maximum Btu Input	Maximum Cubic Feet of Mixed Gas	Maximum Cubic Feet of Manuf'd Gas	Minimum Fire Pot Size
G-100	165,000	208	300	18 Inch

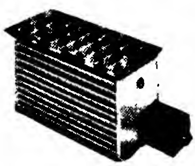
Rheem Manufacturing Co.

570 Lexington Ave., **New York 22, N. Y.**

9 Plants in U. S. A.—foreign affiliated companies in Brisbane, Melbourne, Sydney, Rio de Janeiro, Singapore, Hamilton, Canada, and Zaan Dam, Holland.

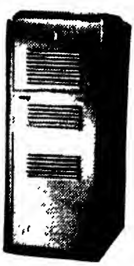


CONSOLE HEATER—modern warm air heating appliance for small homes. Handsome walnut finished cabinet blends with furnishings. Available with manual or automatic controls, for operation with any type of gas including LP. 25,000, 35,000 and 50,000 Btu's. Oil-fired models also available. Thermostat optional.



FLOOR FURNACE—compact warm air furnace for small homes. Can easily be installed beneath floor. Works on any type of gas fuel. Also available in dual-wall models. 25,000, 35,000 and 50,000 Btu's. Oil-fired model also available.

GRAVITY FURNACE—economical warm air furnace for central heating. Has efficient burner unit and combustion chamber. For any type of gas including LP. 75,000, 187,000 Btu's. Similar model for coal also available.



WINTER AIR CONDITIONER—fully automatic warm air furnace, with blower-filter unit and humidifier enclosed in a deluxe casing. Highboy and Lowboy models. Operates on any type of gas. 60,000 to 350,000 Btu's. Coal and oil-fired models also available.

NOTE: Complete specification data will be rushed upon request.

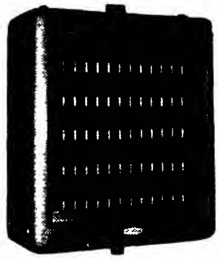
Airtherm Manufacturing Company

728 S. Spring Ave.

St. Louis 10, Mo.

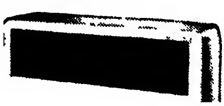
AIRTHERM Gas or Oil Fired Space Heaters

A complete factory heating unit. Capacities from 650,000 to 1,950,000 Btu per hour. For detailed information, write for Bulletin 801-A.



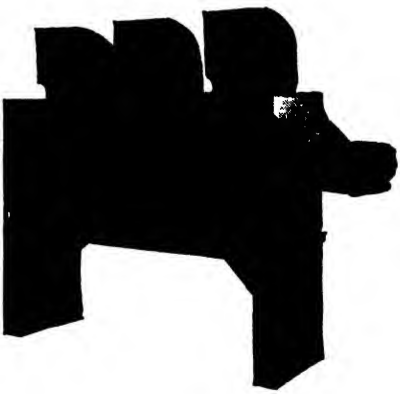
AIRTHERM Steam Unit Heaters

Airtherm Horizontal Propeller Type Steam Unit Heaters are available in capacities from 27,000 to 270,000 Btu. Vertical discharge models also available. Write for Bulletins 1206 and 1207.



AIRTHERM Convectors

Airtherm Convectors are available in three cabinet styles—Type F, free standing or partially recessed; Type W, wall cabinet; and Type S, sloping top wall cabinet—in a complete range of sizes. Write for Bulletin 701.



Airtherm Blower Fan Type Unit Heater available in capacities from 222,000 to 827,000 Btu. Floor, vertical or horizontal models. Write for Bulletin 401.

Campbell Heating Company

3121 Dean Ave., Des Moines, Iowa

DIRECT FIRED SPACE HEATERS

Oil, Gas, Stoker, Hand Fired or Combination Gas and Oil Fired

Heater Guaranteed for 10 Years from Any Cause, Blower, Motor, Burner and Controls, for One Year Against Defects.

The Campbell Direct Fired Space Heaters are designed to be located in the room to be heated but they can be connected to a duct system for heating any type of building and are quiet enough to be used in churches, schools, etc

They are designed to be shipped as a complete unit ready to connect to fuel line, electric power and flue but they can be shipped in sections for assembling in a basement room. The steel heater is then assembled and welded on the job.

Illustration Shows the Oil Fired Unit

Furnished also for Gas, Stoker or Hand Firing

Unit Number	Btu (1) Output Capacity	70° Return Air—No Ducts: Blower CFM (2)	Motor HP (3)	Heating Surface Sq. Ft.	Overall Dimensions inches Not incl. burner: W	L	Ht.	Stack Connection Dia. In. (4)	Firing Rates: Oil GPH	1000 Btu GAS C.F.H.	Stoker Lbs/Hr	Approx. Shipping Weight
U8075	725,000	8,850	2-½ HP	280	75	80	114	12	7.0	900	80	5,000
U8100	893,000	10,900	2-½ HP	320	75	93	118	14	8.5	1,120	100	6,000
U8125	1,080,000	13,200	2-¾ HP	360	75	105	120	16	10.5	1,350	125	7,000
U8150	1,275,000	15,600	2-1 HP	440	75	118	120	16	12.0	1,600	150	8,000
U8175	1,440,000	17,500	2-1 HP	480	75	130	120	18	14.0	1,800	175	10,000
U8200	1,725,000	21,100	2-1 HP	600	94	137	136	20	16.5	2,150	200	11,000
U8250	2,160,000	26,400	2-1½HP	750	94	157	136	20	20.5	2,700	250	13,000

(1) Heat Emission per square foot of heating surface is 3000 Btu per hour or less. If heater is not located in room to be heated an allowance for radiation losses should be made.

(2) Rated air volume produces 75 deg air temperature rise through the heater. This air volume can be varied to suit other conditions.

(3) Motor size can be increased to provide for any duct system.

(4) An induced draft blower can be furnished if a stack or chimney is not available. The gas passages of the heater are adequate so that an induced draft blower is not ordinarily necessary.

For data on other Campbell Products see pages 1052, 1053.

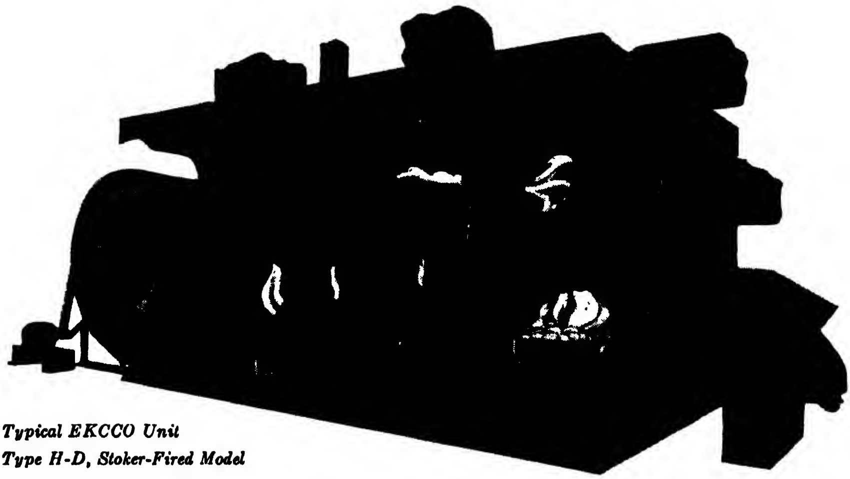
EASTERN REPRESENTATIVE: O. C. ADAMS, 41 EAST 42ND ST., NEW YORK 17, N.Y. MUrray Hill 7-4044

E. K. Campbell Heating Company

Kansas City 3, Missouri

HEAVY DUTY FURNACE FAN EQUIPMENT

Standard Units Up To 8,000,000 BTU/HR.



Typical EKCCO Unit
Type H-D, Stoker-Fired Model

- Made in Units from 500,000 Btu/hr to 8,000,000 Btu/hr.
- Available for any fuel or in "All-Fuels" model.
- Operating efficiencies up to 84 per cent with resultant fuel economy.
- Counter-Flow heat transfer.
- Drive thru principle exclusively (no suction applications).
- Low Internal resistance to flue gases.
- Fiberglas insulated casing.
- Baked enamel exterior finish.
- Extra Heavy welded steel construction throughout.
- Available in any required arrangement of duct outlets.
- Extreme flexibility of equipment arrangements.
- Balanced job design—blower and furnace sized separately.
- Sold only on an engineered basis to fit job requirements.

Used in thousands of large buildings over country, the E. K. Campbell Heating Co's Type H-D Furnace-Fan system is particularly suited for buildings containing large spaces, such as industrial plants, churches, schools, hangars, etc. High quality equipment, designed to last, giving unusually low cost on a year service basis.

The E. K. Campbell Heating Company guarantees RESULTS as well as its equipment.

Inquiries invited regarding LARGE SPACE HEATING PROBLEMS.

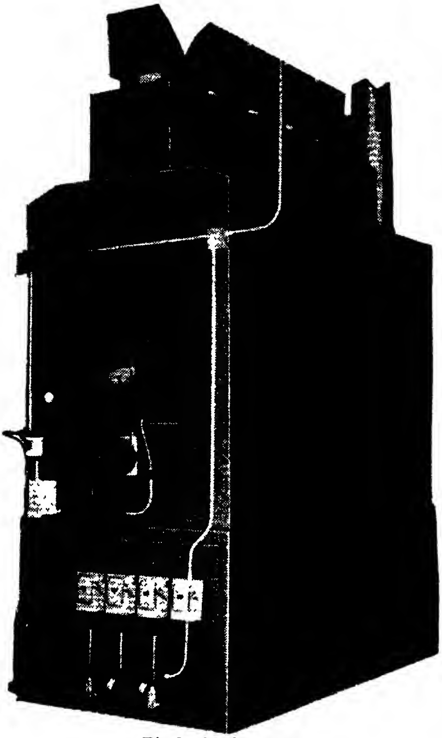
Manufacturing Engineers since 1910

Chicago Steel Furnace Co.

9326 S. Anthony Ave., **Chicago 17, Ill.**

DIRECT-FIRED SPACE HEATERS

Standard Sizes Up to 1,500,000 B.T.U.



Fired with Gas or Oil

HEAT EXCHANGER—Designed and constructed in a near tear-drop shape which permits the air to flow with a minimum of friction. This design affords a maximum of both strength and area in its crown sheet, and reduces power consumption to the very minimum.

COMBUSTION CHAMBER—Fabricated from four way $\frac{3}{16}$ and $\frac{1}{4}$ in. floor plate. The chamber is of the correct size and shape to provide proper volumetric content for the complete combustion of oil or gas.

BLOWERS—Three standard up-blast discharge, multi-vane Blowers, each individually powered, are provided to supply the correct amount of air to properly cool the Heat Exchanger. Individually-driven Blowers are used because of the great flexibility of operation obtainable. Uniform bonnet temperature may be enjoyed by the simple expediency of increasing or reducing the amount of air driven around any particular section of the Heat Exchanger.

A Modern, Economical and Flexible Method of Heating Large Areas

TECHNICAL AND RATING DATA—Larger Sizes Available

SERIES "A"—MODEL NUMBERS		350-A	500-A	750-A	1000-A
Heat Delivery	Btu. per Hr.	350,000	500,000	750,000	1,000,000
Heating Surface	Sq. In.	26,152	34,152	46,330	59,124
Combustion Space	Cu. In	82,264	119,416	182,232	243,524
Oil Firing Rate	G. P. H.	2.5—4.0	4.0—6.0	6.0—9.0	9.0—12.0
Air Delivery	C. F. M.	3,000 to 4,200	6,000 to 7,800	8,200 to 9,600	12,500 to 15,000
Blower Motor	3 Req'd (ea.)	¼ H. P.	⅜ H. P.	½ H. P.	1.0 H. P.
Dimensions					
Height	overall	8'—0"	9'—6"	9'—10"	10'—6"
Width	overall	3'—0"	3'—6"	4'—2"	4'—8"
Length	overall	6'—10"	7'—10"	8'—4"	10'—0"
Cold Air Inlet	2 each	6'-10" x 2'-0"	7'-10" x 2'-3"	8'-4" x 2'-7"	10'-0" x 3'-0"
Warm Air Outlet		6'-0" x 1'-0"	7'-0" x 1'-8"	8'-0" x 1'-8"	10'-0" x 2'-0"
Flue Gas Outlet	Diam	8"	10"	12"	14"

"TERRITORIES NOW BEING ASSIGNED"

Lee Engineering Company

Seaboard Trust Bldg., 95 River St., **Hoboken, N. J.**

LEE DIRECT WARM AIR HEATING

The Lee System of warm air heating generally costs less to install than steam or hot water; utilizes fuel with a high degree of efficiency; distributes the heat exactly where needed; responds promptly without lag; requires little or no maintenance; and needs no licensed attendant. Heaters for use with the Lee System are made in the four types illustrated and described briefly below.

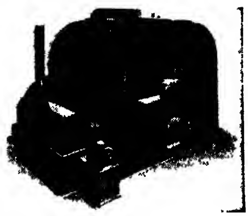
BRICK-SET TUBULAR HEATER

For use with central heating system in connection with duct distribution. Single heater capacities from 2,800,000 Btu per hour to 8,000,000 Btu per hour. Two heaters, installed as a battery serving as one unit, provide capacities over 10,000,000 Btu per hour.

Brick Set Tubular Heater

STEEL ENCASED TUBULAR HEATER

For use with central heating systems in connection with duct distribution over a capacity range of from 2,000,000 Btu per hour to 6,000,000 Btu per hour. Heater may be installed in heated area without enclosure, requires no foundation, and may be moved from one location to another by taking unit apart and reassembling.



Steel Encased Tubular Heater

TUBULAR UNIT HEATER

For use either as a central system in connection with duct distribution or with adjustable outlet nozzles as a unit heater. Capacity range from 2,000,000 Btu to 6,000,000 Btu per hour. In sizes up to 4,000,000 Btu heater is shipped as a completely assembled unit with all but mechanical equipment, refractory lining and controls in place. Heaters require no foundation and are equipped with crane hooks so that they may be moved from one location to another.

Tubular Unit Heater

SHELL UNIT HEATER

For use with or without distributing duct system. Heaters have capacity range of from 300,000 Btu to 2,000,000 Btu per hour. Available for gas, oil or combination gas-oil firing. All units are shipped completely assembled, wired and ready for operation. Units furnished with either refractory lined or stainless steel combustion chambers. Heaters may be floor mounted or suspended in any position. Also available in both hand and stoker fired models.

Shell Unit Heater

For further information write for Catalog HV-49.

DRAVO CORPORATION

HEATING SECTION

Dravo Bldg., Fifth and Liberty Avenues

PITTSBURGH 22, PA.

DRAVO *Counterflo* DIRECT-FIRED HEATERS

Gas-Fired Dravo "Counterflo" Direct Fired Heater

Used in the following and many other industries and commercial operations

Automobile Manufacturers	**Municipalities, Government**
Aircraft Mfrs. and Airports	**Non-Ferrous Industries**
Construction Industry	**Paint**
Communications	**Paper and Paper Products**
Cement, Rock Products	**Plastics and Insulation**
Chemical Plants	**Printing and Publishing**
Electrical Products	**Rubber**
Farms	**Railroads**
Foundries	**Shipbuilding**
Food Products	**Schools, Colleges, Churches**
Glass, Ceramics, Refractories	**Steel Producers**
Institutions	**Transportation**
Metal Working Industries	**Textile**
Mercantile	**Woodworking**

5 FUNCTION HEATING with just one unit

DRAVO *Counterflo* Heaters are direct-fired space heaters designed for providing comfortable warmth for large open areas of buildings, but provide four other functional uses as well—namely; year 'round ventilating, tempering make-up air, process drying and heat curing. Obviously not all five functions of a DRAVO Heater are required in all plants, but every one of these functional applications can be provided by the DRAVO Heater.

DRAVO *Counterflo* Heaters are self-contained, requiring only fuel pipe, power line and vent stack for installation. Thermostatically controlled, they require no regular attendant. Structural changes are unnecessary for installation. They occupy little floor space, and can be wall-mounted or suspended from roof trusses. Because their stainless steel combustion chamber eliminates refractory linings, they can be mounted in either a horizontal or vertical position.

DRAVO Heaters are available for firing with gas, oil or coal. Units burning non-solid fuels can be converted from one fuel to another very quickly. In addition, coal-fired units can be converted to gas or oil-firing.

Descriptive Bulletin HVA-523 and specification sheets are available on request. Write the Heating Section, DRAVO CORPORATION, Room 812, Dravo Bldg., Pittsburgh 22, Pa.

DRAVO CORPORATION

Pittsburgh • New York
Cleveland • Chicago
Philadelphia • Atlanta
Detroit • Boston
Sales Representatives in Principal Cities
Manufactured and sold in Canada by Marine Industries, Ltd., Sorel, Quebec

Features of the

DRAVO *Counterflo* Heater include:

- Stainless Steel Combustion Chamber
- *Counterflo* Combustion and Heat Transfer
- 80–85 per cent Sustained Efficiency
- Effective Comfort-Zone Recirculation
- Minimum Heat Loss Through Roof
- Simplicity of Operation
- Flexibility of Application
- Adaptability for Wide Range of Fuels

Unit capacities of DRAVO *Counterflo* Heaters range from 400,000 to 2,000,000 Btu per hour output. The following tables show the external dimensions and weights of the various units as well as the capacities of each size heater.

Cutaway view of Dravo Counterflo Heater showing the stainless steel combustion chamber and the extended flame travel within the chamber which results in efficiency of 80 to 85 per cent burning gas or oil.

ENGINEERING DATA

Heater Number	Btu Output Capacity	Air at 70 Deg. Free Delivery		Fan Motor HP	No. Discharge Nozzles	Approx. Shipping Weight
		Approx. CFM	Temp. Rise Deg. F			
40	400,000	4500	82	1½	3	1400
50	500,000	5500	84	2	3	1500
75	750,000	8500	82	3	4	2500
100	1,000,000	11000	84	5	4	2600
125	1,250,000	14000	83	7½	4	3400
150	1,500,000	17000	82	10	4	3600
175	1,750,000	19000	85	15	4	4200
200	2,000,000	22000	84	20	4	4400

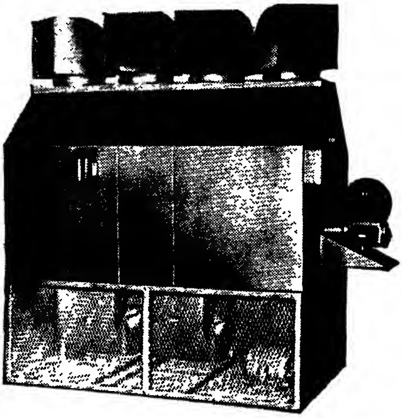
Heater Number	FUEL CONSUMPTION—Minimum of 80 per cent Heater Efficiency Assumed			Approximate Overall Dimensions		
	Light Oil GPH 140,000 Btu	Heavy Oil GPH 148,000 Btu Oil	Gas CFH 1000 Btu Gas	Width Ft.—In.	Length Ft.—In.	Height to top of Nozzles Ft.—In.
40	3.5	3.5	500	2—6	5—1	8—3
50	4.5	4.3	625	2—6	5—1	8—3
75	6.7	6.5	940	3—7	7—5	9—11
100	8.9	8.6	1250	3—7	7—5	9—11
125	11.1	10.8	1560	4—2	9—1	11—2
150	13.4	13.0	1875	4—2	9—1	11—2
175	15.6	15.0	2190	4—8	9—8	12—9
200	17.8	17.2	2500	4—8	9—8	12—9

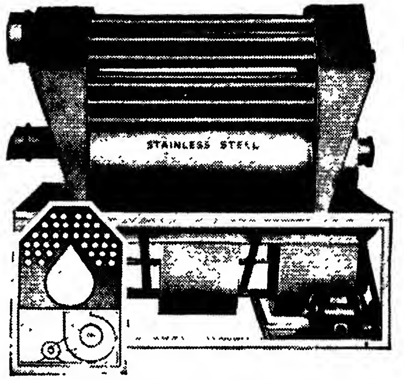
National Heater Company

2182 Cleora Avenue, St. Paul 4, Minnesota

NATIONAL CHAMPION

STEEL S-P-A-C-E HEATERS





GAS AND OIL FIRED UNITS

National Champion Heaters are available in both factory floor set units, as illustrated, and in quiet operating, resilient mounted models for convenient connection to duct distribution system. Skeleton view illustrates compactness of heat exchanger and blower section mounted on sturdy angle-iron base frame. Lifting eyes are provided for crane handling. Streamlined firebox of rust and heat resisting Stainless Steel requires no refractory and affords life far beyond that possible with mild carbon steel. Tear Drop combustion chamber design and convector tube arrangement assure complete efficient air impingement on all heating surface at a minimum of resistance. Induced draft blowers can be supplied when suitable chimney or stack is not available.

ENGINEERING RATING CHART—TEAR DROP DIRECT FIRED HEATERS

Heater Number	B.T.U. Output Capacity	C.F.M. at 5/8″ S.P.	Sq. Ft. Heating Surface	Blowers D W D I	Maximum Firing Rate: #1-2-3 Light Oil G.P.H.	Maximum Firing Rate: Gas C.F.H.	Blower Motor: H.P.	Blower Motor: R.P.M.	Diameter Smoke Outlet
T.D. 25	250,000	3,600	120	14″ Twin	2.5	330	¾	1750	10″
T.D. 40	400,000	5,400	135	14″ Twin	4.0	550	1	1750	10″
T.D. 50	500,000	6,600	150	14″ Triple	5.0	680	1½	1750	12″
T.D. 70	750,000	8,800	175	14″ Triple	7.0	940	2	1750	12″
T.D. 80	800,000	10,200	200	16″ Triple	7.5	1100	3	1750	12″
T.D. 100	1,000,000	12,500	250	16″ Triple	9.5	1400	5	1750	12″
T.D. 125	1,250,000	15,300	375	18″ Triple	11.5	1610	5	1750	14″
T.D. 150	1,500,000	19,400	425	18″ Triple	14.0	2040	7½	1750	14″

Heater Number	Tubes: No.	Tubes: Dia.	Approximate Shipping Weight	Warm Air Plenum Opening	Overall Dimensions Inches: Width	Overall Dimensions Inches: Length	Overall Dimensions Inches: Height	Gauge Material: Tubes	Gauge Material: Headers	Gauge Material: Stainless Steel Comb. Chamber	Gauge Material: Casing	Gauge Material: Discharge Heads
T.D. 25	23	4″	1,300	14 x 60	32	60	81	14	12	14	20	20
T.D. 40	23	4″	1,350	14 x 60	32	60	81	14	12	14	20	20
T.D. 50	23	4″	1,780	14 x 80	32	80	81	14	12	14	20	20
T.D. 70	23	4″	1,855	14 x 80	32	80	81	14	12	14	20	20
T.D. 80	36	4″	2,110	18 x 80	48	80	81	14	10	14	20	20
T.D. 100	36	4″	2,200	18 x 80	48	80	81	14	10	14	20	20
T.D. 125	48	4″	3,000	20 x 100	54	100	103	14	3/16	14	20	20
T.D. 150	48	4″	3,250	20 x 100	54	100	103	14	3/16	14	20	20

Arthur A. Olson & Company

Broad and Court Streets **Canfield, Ohio**

Manufacturers of Direct Fired Heaters

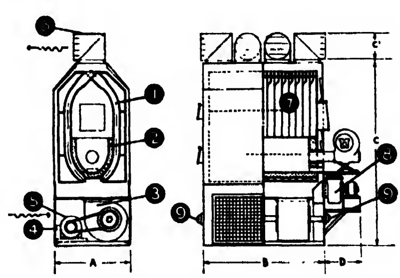
A completely automatic direct-fired unit heater. Olson heaters operate on gas, oil, dual gas-oil or coal. These units may be hung, suspended horizontal, or inverted. All heaters can be adapted to filtering, humidifying and air conditioning. Use of standard accessories and easy access to bearings and other vital parts simplify maintenance.

Also process and make-up air applications.

Engineering Data—Gas, Oil, or Dual Gas-Oil Heaters.

Model Number	B.T.U./Hr. Output Capacity	CFM Blowing Fans	GENERAL DIMENSIONS					No. of Blowing Fans	Motor H.P. Blowing Fans	Stack Dia.	No. of Nozzle Outlets
			A	B	C	C1	D				
U-300	300,000	3,500	2′-11″	4′-1″	6′-6″	1′-0″	1′-3″	2	1	8″	3
U-400	400,000	4,500	2′-11″	4′-1″	6′-6″	1′-0″	1′-3″	2	1½	8″	3
U-500	500,000	6,000	3′-1″	4′-6″	7′-3″	1′-3″	1′-9″	2	2	8″	3
U-600	600,000	7,000	3′-1″	5′-2″	7′-3″	1′-3″	1′-9″	2	3	8″	3
U-750	750,000	8,500	3′-4″	5′-10″	7′-10″	1′-3″	1′-9″	2	3	10″	4
U-1000	1,000,000	11,000	3′-8″	5′-10″	8′-9″	1′-3″	1′-9″	2	5	10″	4
U-1250	1,250,000	14,000	3′-8″	7′-10″	8′-9″	1′-3″	1′-9″	3	7½	10″	5
U-1500	1,500,000	17,000	4′-4″	7′-10″	9′-9″	1′-7″	1′-9″	2	10	12″	4
U-1750	1,750,000	20,000	4′-4″	9′-7″	9′-9″	1′-7″	2′-0″	3	10	12″	5
U-2000	2,000,000	22,000	4′-4″	9′-7″	10′-3″	1′-7″	2′-0″	3	15	12″	5

1. Standard 10 Gage Boiler Tube Heat Transfer Surface.
2. Refractory or Stainless Steel Combustion Chamber.
3. Heavy Duty Fans and Shaft.
4. Inlet Screens or Frame for Filter Box.
5. Motor Accessible from Front.
6. Adjustable, Deflecting Outlets.
7. Four Pass Gas Travel—Counter Current.
8. Separate Induced Draft Fan.
9. Bearings Outside Heater at End of Shaft.



Write for complete data on stoker fired units and large central systems.

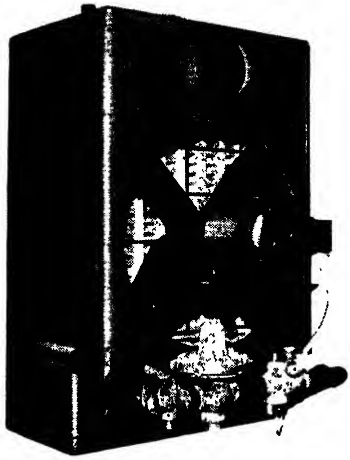
Automatic Gas Equipment Company

Brushton and Thomas St., **Pittsburgh 21, Pa.**

Manufacturers of Pittsburgh Gas Unit Heaters

Cast iron is considered to be the best material to withstand the corrosive effects of the products from combustion of gases. For this reason, both the heat exchanger and combustion chamber in a Pittsburgh Gas Unit Heater are made of cast iron. Furthermore, they are cast in one piece and the extended heating surface fins on the heat exchanger are cast integral.

Pittsburgh Gas Unit Heaters have been designed to consume exactly the right amount of air to support complete combustion but without permitting an excess of air to lower heating efficiency. This is accomplished by means of a built in draft hood which absorbs all excessive chimney action and thereby conserves heat. The heater does not depend upon forced draft from the fan for either the primary or secondary air supply. For this reason there is no possibility of variation in the air supply to the burners resulting from changes in fan speed or louver adjustment. By the use of adjustable horizontal louvers, warm air can be directed to any desired level.



Cut-away view of heater, showing the cast iron heat exchanger in place.

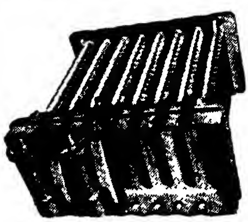
Safety Features

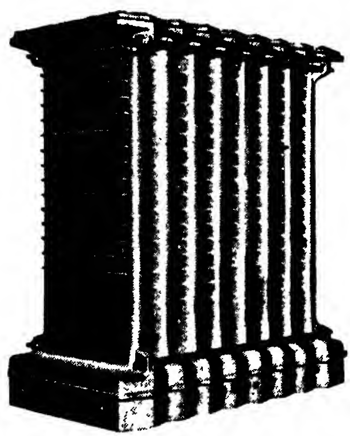
A tested and proved safety pilot is used on these heaters to automatically turn off the gas if the pilot light goes out or if it burns too low to insure positive ignition. The draft diverter is absolute protection against any possible down drafts through the chimney. Write for folder containing complete details, including installation measurements.

Approved by *American Gas Association.*

(Right) Heat exchanger and combustion chamber.

(Below) Bottom view showing burner assembly (several burners have been removed).





SIZES AND CAPACITIES

Unit No.	Input B.T.U. Per Hour	Output—AGA B.T.U. Per Hour	Sq. Ft. E.D.R.	Air Del. C.F.M.	Motor H.P.	Speed R.P.M.	Approx. Wts.
215 C	215,000	172,000	744	3500	[illegible]	1140	525
175 C	175,000	140,000	605	2900	[illegible]	1140	475
160 C	160,000	128,000	553	2600	[illegible]	860	450
140 C	140,000	112,000	484	2320	1/15	860	400
110 C	110,000	88,000	381	1820	1/20	860	350
85 C	85,000	68,000	294	1350	1/30	1000	300

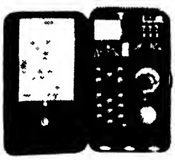
COMBUSTION CONTROL CORPORATION

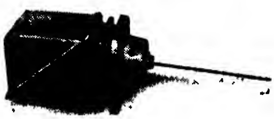
Flame Failure Safeguards **fireye** ***For Oil and Gas Flames***

77 Broadway, **Cambridge 42, Mass.**
10447 Sangamon St., Chicago—765 Cordilleras Ave., San Carlos, San Francisco
District Offices in all Principal Cities

FIREYE PHOTOELECTRIC FLAME FAILURE SAFEGUARD AND PROGRAMMING CONTROL FOR OIL BURNERS


Type 45PH5


Type 24PJ8


Type 45JP1

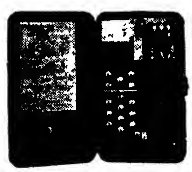
Complete operating and starting protection for industrial and commercial oil burners with Flame Rod protection of gas pilot. Type 24PJ8 automatically starts burner and programs sequence of gas pilot, ignition, burner motor, oil valves, providing scavenging period, fuel valve delay, post ignition time. Flame Rod, Type 45JP1, monitors gas pilot flame, preventing opening of oil valve unless gas pilot is established. Scanner Type 45PH5 takes over monitoring of oil flame after pilot is established. Failure of either gas flame during ignition or main oil flame during normal operation results in immediate shutdown of burner system.

Fireye equipment is available in combinations providing one or all of these safeguard functions, depending on requirements of installation.

FIREYE ELECTRONIC ROD FLAME FAILURE SAFEGUARD AND CONTROL FOR GAS BURNERS

Operating protection for industrial and commercial gas burners. Flame Rod Type 45JQ1 constantly monitors gas pilot flame after its manual or electric ignition. Main gas valve cannot open until Flame Rod indicates pilot flame is established. Pilot flame failure after opening of main fuel valve is instantly signalled by Flame Rod to Electronic Control Type 24QJ5, which immediately shuts off Burner.

Fireye equipment is designed with completely fail-safe characteristics. Any circuit element failure results in system shutdown. A built-in low-voltage interlock completely checks system, internal and external to control, on each burner recycle.


Type 24QJ5


Type 45JQ1

PHOTOSWITCH SMOKE DETECTORS FOR AIR CONDITIONING DUCT SYSTEMS


Type A28E


Type L30R

Photoelectronically monitors duct systems detecting the presence of even small amounts of smoke. At the first sign of smoke, Photoswitch Type A28E automatically turns off blowers, closes automatic louvres, and signals the maintenance department. Recommended for use in theatres, stores, hotels, and other locations where the presence of smoke is a hazard to property or a possible cause of panic.

UNIT DEHUMIDIFIERS FOR DYNAMIC AS WELL AS STATIC DEHUMIDIFICATION

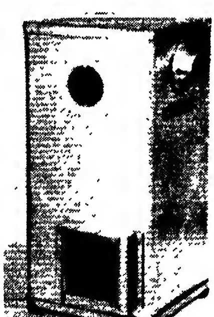
DAPCO AIR DRYER—MOD. SOR-8 Solid Adsorption Desiccant Dehumidifier for spaces up to 5000 cu ft. Applied for preservation of materials in storage and for basement and recreation room drying, as well as plaster and paint drying.

Dapco Air Dryer SOR-8 Also U. S. Navy Package Dehumidifier

UNIT DATA:

Dimensions: 15 in. x 13 in. x 18¾ in. high
Air Flow: 23 cfm
Capacity: 8 lb of water/day at 73F dew point.
5 lb of water/day at 41F dew point.
Power Economy: 0.65–0.90 kwhr/lb of water removed.
Electrical Characteristics 110 volts, A.C.,60 cycles. Available for other currents on request.
Connected load on reactivation 850 watts. On adsorption 50 watts.
Operation: Intermittent drying and reactivation. Completely automatic.
Controls: Unit may be started either manually or operated by humidistat control.

DESOMATIC—MOD. A. This is a small dynamic unit of similar design to DAPCO AIR DRYER—SOR-8, but intended for smaller spaces of 100–500 cu ft. The unit will, however, reduce the relative humidity in the air about 10 per cent in relatively tight spaces of up to 2000 cu ft volume, which often is ample to prevent condensation.



Desomatic Mod. A.

UNIT DATA:

Dimensions: 7½ in. x 7½ in. x 15 in. high
Air Flow: 12 ½ cfm
Capacity: 2½ lb of water/day at 73F dew point.
Power Economy: About 1 kwhr/lb of water removed.
Electrical Characteristics: 110 volts, A.C., 60 cycles. Available for other currents on request. Connected load on reactivation 325 watts. On adsorption 25 watts.
Operation: Intermittent drying and reactivation. Completely automatic.
Controls: Unit may be started either manually or operated by humidistat control.

DYNAMIC UNITS FOR SPECIAL APPLICATIONS SUCH AS DRYING OF COMPRESSED AIR LINES, ELECTRICAL CABLE SPLICING OR TELEVISION TOWERS AVAILABLE ON REQUEST.

DESICAN—STATIC DEHUMIDIFIERS. **Desicans** are available in five sizes for dehumidification of confined spaces of from 2–80 cu ft. The desiccant is enclosed in an aluminum container with a perforated metal screen. **Desicans** have indicator eyes to inform about the degree of saturation of the desiccant. Desicans can be regenerated by baking in oven at 300F. The **Desicans** are equipped with a dust screen and may therefore be used for precision instruments or in food containers as the desiccant does not contaminate food.

Dry Air Products Corporation engineers will assist you with any problem you may have in connection with preservation of materials from moisture damage by means of solid desiccants.

ELECTROMODE CORPORATION

45 Crouch St. Rochester 3, N.Y.

FOR HOME • INDUSTRY • FARM

ELECTROMODE *All Electric* HEATERS

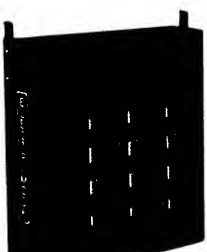
BILT-IN-WALL Heaters for Homes and Offices

These heaters fan-circulate warm air by the Down-Flo principle for greatest heating efficiency. They're quick and easy to install for no ductwork is required. Heating unit is embedded in a finned aluminum casting for complete safety. Available in capacities from 1500 to 4000 watts and with manual, wall thermostat or built-in thermostat control. All have thermal safety switches and silver gray enamel finish.

BILT-IN-WALL SMALL-ROOM Heater

This 115-volt AC, 1320-watt heater was specially designed for baths, small rooms, trailers, etc. It employs the famous Electromode cast aluminum safety heating element with thermal cut-off. Installation for manual or thermostat control is simple and quick. Two-way switch permits use of fan without heat. Available in white baked-on enamel finish or in chrome.

UNIT HEATERS for Auxiliary Warmth



These fan-circulating units are used as auxiliaries to a main heating system, or where electric rates permit, as main heat sources. They require no plumbing or ductwork—only circuit wiring. Like all Electromodes, they employ the completely safe cast aluminum heating element that has no exposed hot or glowing wires, that gives high thermal conductivity and resists corrosion. A safety cut-off prevents overheating. Thermostat control available on all models.

Suspension Type (left, above) for wall or ceiling mounting in factories, stores, offices. 10,000 to 60,000 watts.

Combination Portable and Suspension Type (left, below). Plug in where needed, or mount permanently. 1500 to 7500 watts.

Heaters for Special Applications—Blower units; explosion-proof heaters; blast heaters for air ducts; mobile units on casters; electric home furnaces; portables; heavy-duty farm heaters. Engineering help is at your disposal. For more information see your supplier or write Dept. HVG-19, Electromode Corporation.

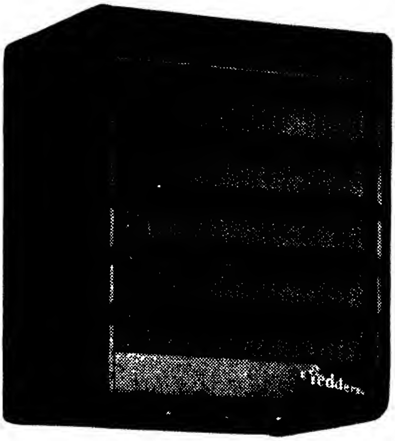
Electromodes are approved by *Underwriters' Laboratories* and are fully guaranteed.

Fedders-Quigan Corporation

57 Tonawanda Street
Buffalo 7, N. Y.

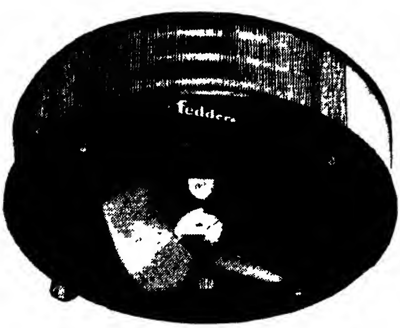
Heat Transfer Specialists since 1896

Manufacturers of Convector-Radiators, Unit Heaters, Railroad Car Convectors, Unit Coolers, Refrigeration Coils, Air-cooled Fin and Tube Condensers, Clip-on Thermometers, Room Air Conditioners, Automotive Radiators, Car Heater Cores, Electric Water Coolers.



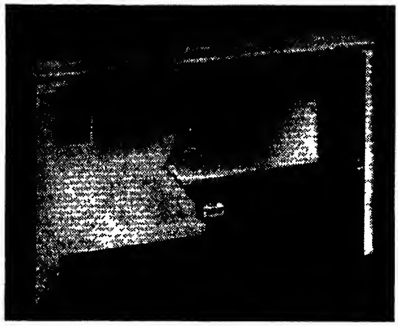
Fedders Series 15 Horizontal Unit Heaters

FEDDERS SERIES 15 HORIZONTAL UNIT HEATERS have capacities of 100 to 1,000 EDR. Handsome, rugged cabinets, with latest type broad blade fans provide large air volume, quiet operation and high efficiency. Write for Bulletin 15C-4.



Fedders Series 16 Downblow Unit Heaters

FEDDERS SERIES 16 DOWNBLOW UNIT HEATERS have capacities of 155 to 2,050 EDR. Designed for applications necessitating clearance for material handling equipment, tall buildings and other conditions which require piping to be kept out of the way. High velocity air stream delivers heat into working zone. Write for Bulletin 16C-1.



Fedders Type F Convector-Radiators

FEDDERS TYPE F CONVECTOR-RADIATORS

Designed for installation in living rooms, bedrooms, offices, institutions and other locations. They can be used with steam, and forced or gravity hot-water systems.

Cabinets can be used as a free-standing unit or partially recessed into the wall. Naturally induced circulation of air entering through archway at bottom and actively flowing out through louvered grille at top assures more uniform temperatures and maximum comfort at all levels.

FEATURES INCLUDE

1. Directional louvers in front panel direct the heated air actively into the room, assuring more uniform temperatures from floor to ceiling.
2. Cabinet finished in oven baked neutral ground coat.
3. Front panel easily removed.
4. Light weight, easily handled and completely packaged for simplified stocking and delivery.
5. Heating element with copper tubes and aluminum fins.

Write for catalog giving complete data on 4, 6, 8 and 10 in. depths, lengths from 20 in. to 64 in. and 18, 20, 24 and 32 in. heights.

Grinnell Company, Inc.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence 1, R. I.**

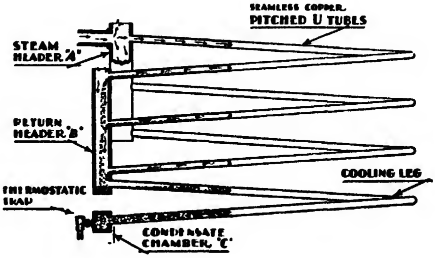
National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 1218–1219

THERMOLIER

REG. U. S. PAT. OFF.

THE GRINNELL UNIT HEATER



Patented Construction of Internal Cooling Leg

In addition to its patented Internal Cooling Leg Thermolier has many other desirable features including its Single Header U-Tube construction which compensates for expansion and contraction strains. Radiation is from brass-finned seamless copper U-tubes rolled into a cast iron tube sheet.

Steam circulation and the removal of condensation in Thermolier are distinctly different than is usual in unit heaters. The actual cooling effect of this construction is equal to a run of more than 100 ft of ordinary, exterior cooling leg piping.

Steam is delivered into Chamber "A" of the header and circulates from there through the pitched U tubes, carrying its condensation with it into Chamber "B." By partitioning off the lower tube or tubes at the bottom of the Steam Supply Chamber "A" these tubes carry all condensation from Chamber "B" into Drain Chamber "C." In passage of this condensation through these tubes, the air from the fan is rapidly carrying off heat just as it does in the rest of the unit. The result is that these two bottom tubes form an efficient internal cooling leg, integral with the unit.

Thermolier is available in 10 Models. Catalog will be sent on request to Grinnell Company, Inc. 277 West Exchange Street, Providence 1, R. I., or to any branch office in principal cities listed on our page 1218.

CAPACITIES

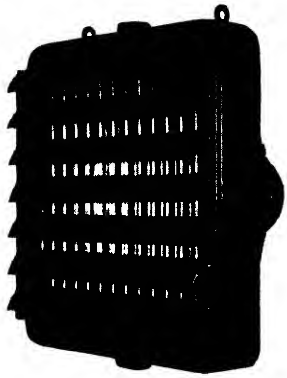
60 F Entering Air Temperature—2 Lb Steam Pressure

Model Nos.	Btu per Hour	Equivalent Direct Radiation	Model Nos.	Btu per Hour	Equivalent Direct Radiation
D21	35,600	148	D57	101,300	422
D31	48,700	203	D66	128,700	536
D37	62,200	259	D71	151,700	632
D41	71,000	295	D91	196,000	817
D44	84,100	350	D111	275,300	1147

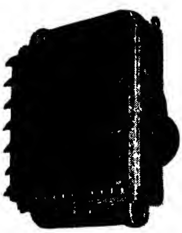
ILG Electric Ventilating Co.

2876 North Crawford Ave., **Chicago 41, Ill.**

Offices in More Than 40 Principal Cities



Horizontal Type Unit Heaters—have ILG-built Self-Cooled Motor which counteracts coil heat—never "slow roasts." Graduated, 2-piece, cast iron header gives "balanced" steam distribution. Brass orifice bushings expand tubes uniformly in header plate. Copper fins are *pressed* into round copper tubes for permanent union—no brazing, soldering, or welding. Bottom header "floats" to permit expansion and contraction of coil independent of casing. Tested and Rated according to codes of *I.U.H.A.* and A.S.H.V.E. Ratings certified by *I.U.H.A.*—"One-Name-Plate" Guarantee. Wide range of sizes and capacities.

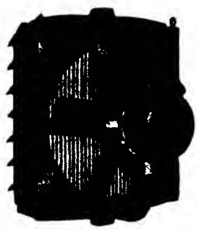


Low-Ceiling Type

For vertical mounting in buildings where headroom is at a premium. Side inlets and outlets at top and bottom assure extremely compact installation.

Vertical Type

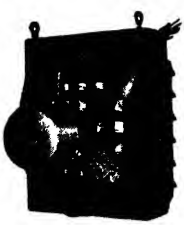
Recommended for installations with extremely high or extremely low ceilings. Air diffusers or deflectors available to direct flow of heater air.



Textile Type

More tubes, no fins—for textile mills and applications where lint or other material normally adheres to fin surfaces and clogs up coil.

ILG Electric Unit Heaters



STANDARD TYPE

For instant, clean, safe, dependable heating. Coil is of black heat type which operates below 400 degrees. Protected against excessive temperature rise by patented automatic thermal cut-out and magnetic starter. Sizes 5 to 15 KW.

TYPE "HT"

For installations requiring a small volume of heat. Exceptionally efficient. Suitable for constant duty. Non-overheating black heat type coil with individually interchangeable elements. Sizes 1½ to 4 KW.

For ILG Propeller and Centrifugal Fans, see page 1163

Jaden

MANUFACTURING COMPANY

HASTINGS **NEBRASKA**

SINCE 1914

New and Improved Self Contained Air Conditioning Units—for use with cold water.

Capacities From 1 Ton to 20 Tons with Comparable Capacities for Heating.

Models for Cooling Include Floor Type for Individual Rooms, Suspended Units and Central Plants for Duct Systems.

Centrifugal Unit Heaters, Package and Utility Blowers for Heating and Ventilating

● **Central Plants**—capacities from 3 tons to 24 tons. Air delivery from 1100 cfm to 10,000 cfm. All central plants are assembled in sections to simplify installation. All models available with steam coils for combination heating and cooling.

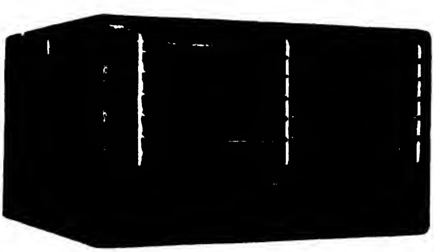
● **Suspended Models**—capacities from 1 ton to 4 tons. Air delivery 585 cfm to 2200 cfm. Especially adaptable to individual rooms where duct system is undesirable. Completely self-contained with motor, coils, blower and filters housed in sturdy steel cabinet with attractive featherweave finishes.

● **Floor Type Models**—capacities from 1 ton to 3 tons. Designed for use in individual rooms. All models self-contained with motor, blower coil and filters housed in well designed metal cabinets attractively finished or available in base coat for finishing in special colors.

● **Package Blowers**—capacities 1000 cfm to 4000 cfm. All models equipped with filters and available with or without fan controls. Cabinets of extra heavy steel construction with featherweave finish or with base coat for finishing in special colors.

● **Utility Blowers** both single—twin mounted double inlet type—capacity from 800 cfm to 10,000 cfm. Available in any discharge arrangement and with or without motors. Single blowers available in "base type" which includes heavy angle frame for special cabinet construction when necessary.

● **Unit Heaters**—capacities from 76,800 to 155,000 Btu/hr. std. rating, air delivery 1165 cfm to 2100 cfm. Centrifugal type for use with steam or forced circulated hot water. Unusually quiet in operation—highly efficient and easily adaptable to duct installation. Finished in attractive featherweave or base coat for special colors.



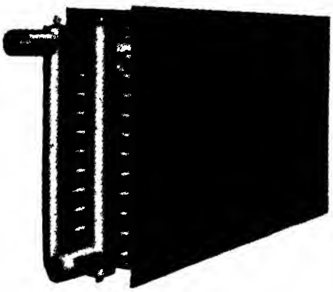
WRITE FOR CURRENT SPECIFICATION CATALOG

KENNARD Corporation

1819 So. Hanley Rd., St. Louis 17, Mo.

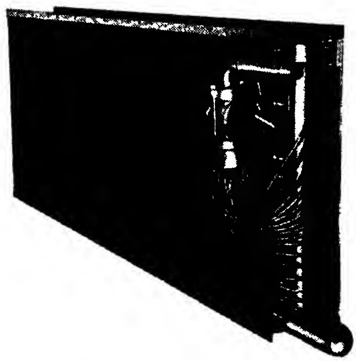
Manufacturers of
HEAT TRANSFER EQUIPMENT

AIR CONDITIONING BLOWER UNITS—HEATING AND COOLING COILS — EVAPORATIVE CONDENSERS — COOLING TOWERS — INDUSTRIAL COOLERS — UNIT COOLERS — UNIT HEATERS



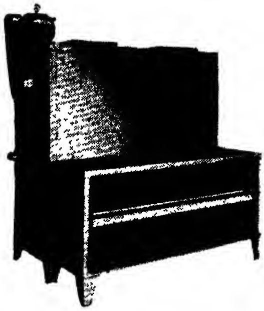
WATER COOLING AND HEATING COILS

For cooling with cold water and for heating with hot water.



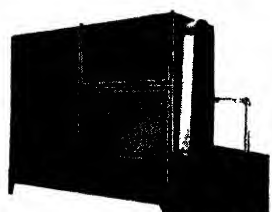
DIRECT EXPANSION COILS

For freon. A complete range of sizes to meet all conditions and capacities.



EVAPORATIVE CONDENSERS

3 to 50 tons. All refrigerants. All prime surface coils. Indoor or outdoor units.



BLOWER UNITS

Ceiling and floor type air-conditioning units. 1 to 50 tons nominal capacities in 12 sizes. 400 to 15,000 cfm. Various arrangements of discharge, filter box and motor drive.

INDUSTRIAL COOLERS
(Not Illustrated)

1000 to 15,500 cfm. Floor type. Coils up to 12 rows in depth. All refrigerants.

UNIT HEATERS

Horizontal propellor fan type. Steam ratings 26,500 to 268,300 Btu's.

D. J. Murray Manufacturing Co.

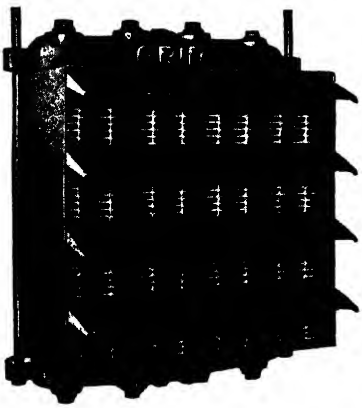
Wausau, Wisconsin

Offices in Principal Cities

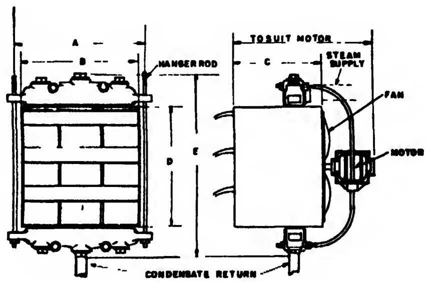
T. M. REG.
U. S. PAT. OFF.

MANUFACTURERS OF THE GRID UNIT AND GRID BLAST COILS

Designed and tested to operate with steam or hot water systems—for steam pressures from 2 lbs to 250 lbs. Engineered along the same lines as the standard GRID Unit which had aluminum heating sections and has been on the market since 1929.



One piece construction "fin" heating sections of high test cast iron—no soldered, brazed, welded or expanded connections. Patented.



Overall dimensions for installation of Cast Iron GRID Unit Heater

CI (CAST IRON) SERIES GRID UNIT HEATER DATA

Model No.	Dimensions					Motor		Vol. Fan CFM	Capacities 5 PSI Steam 60°F Air		Pipe Size		Support Rod Dia.	Approx. Ship. Weight Lbs.
	A	B	C	D	E	HP	RPM		Btu/Hr	Final Temp. °F	Supply	Return		
CI-1000	11⅝	13⅝	12¼	9¾	15⅛	1/25	1550	572	29,080	106	1¼	1¼	⅜	150
CI-1200	14⅞	16⅞	12¾	12¾	18⅛	1/15	1550	798	45,450	112	1¼	1¼	⅜	210
CI-1500	17½	15⅝	11½	16	23⅞	1/8	1750	1500	76,500	107	1½	1¼	½	280
CI-1520	17½	15⅝	11½	21	28⅝	1/8	1750	1700	101,500	114	1½	1¼	½	390
CI-2000	22⅛	20 9/16	11⅞	21 1/16	28⅝	1/6	1150	2600	143,000	110	2	1¼	½	490
CI-2025	22⅛	20 9/16	11⅞	25⅞	35¼	1/6	1150	2875	173,640	115	2	1¼	½	520
CI-2500	27½	25⅞	13	25⅞	35¼	1/2	1150	4350	224,000	107	2	1¼	⅝	700
CI-2504	27½	25⅞	13	25⅞	35¼	1/4	1150	3300	206,000	117	2	1¼	⅝	660
CI-2530	27½	25⅞	13	31	40½	1/2	1150	4650	275,800	114	2	1¼	⅝	900
CI-3000	32⅝	31	13	31	40½	1/2	850	6300	332,000	108	2½	1¼	⅝	1020
CI-3000	32⅝	31	13	31	40½	1½	1150	8000	380,000	103	2½	1¼	⅝	1070

NO ELECTROLYSIS TO CAUSE CORROSION

Low maintenance expense.
More air changes per hour.
Positive "directed" heat.
No leaks—no breakdowns.

Lower outlet temperature.
Larger air volume.
No soldered, brazed or expanded joints.
Open design that keeps units clean.

Send for complete catalog information

Send for information on Blast coils and radiation.

McQuay, Inc.

1602 Broadway, N.E., **Minneapolis 13, Minn.**

MANUFACTURERS OF AIR CONDITIONING EQUIPMENT

Sales Offices in all Principal Cities

PROVEN McQuay PRODUCTS

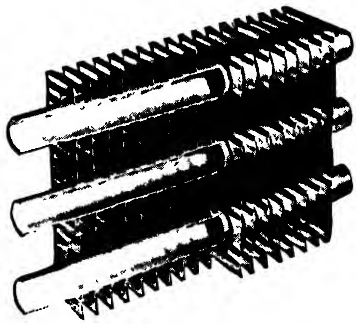
- **Air Conditioners**
- **Air Conditioning Coils**
- **Blast Heating Coils**
- **Refrigeration Coils**
- **Unit Heaters**
- **Unit Coolers**
- **Comfort Coolers**
- **Blower Coolers (Suspended & Floor Type)**
- **Ice Cube Makers**
- **Icy-Flo Accumulators**
- **Zeropak Low Temp. Units**

THE EXCLUSIVE McQUAY RIPPLE-FIN RIPPLE-TUBE COIL ASSEMBLY

McQuay hydraulically expanded Ripple-Fin Ripple-Tube Coils effect permanent contact between tubes and the entire surface of the fin collars—that's the advantage of the hydraulic pressure method—to create the lasting mechanical bond without the use of any "low conductivity" bonding material.

This significant advantage is a typical example of how a seemingly small detail in engineering design plays an important part in making superior products.

McQuay construction means higher flexible strength with less air friction and cleaner operation. To provide greater flexibility, all headers are of non-ferrous tubes, elipted to compensate for any unequal expansions and contractions.



RIPPLE-FIN COIL

All secondary surface is of the aluminum ripple-fin continuous plate type, to give extra strength, and to provide heat transfer surface that remains clean for a longer period, thereby giving greater efficiency; all tubes are electro-tin plated for further protection and longer life.

The new standardized design provides 11 header sizes and 19 tube lengths, plus other intermediate header sizes and many other tube lengths. McQuay thus provides greater flexibility for sizing jobs.

This careful attention to detail makes McQuay performance possible and establishes their preference among users. McQuay coils are available in a wider variety of styles and sizes, both standard and special coils for steam, hot water, cold water, brine, direct expansion and other applications.

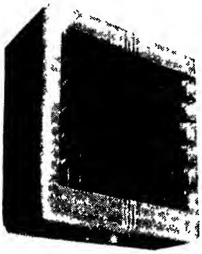
COMBINATION COOLING COIL

WATER COIL

STEAM BLAST COIL

MORE THAN 1,000,000 COIL TYPES AND SIZES

McQUAY manufactures a complete line of Standard Coils for the Industry.
Coils for Heating—1 to 10 rows deep using low or high pressure steam or hot water.
Jet-Tube (Non-Freeze steam inner tube) type coils 1 and 2 rows deep.
Cleanable Tube—(Removable plug) type coils 1 to 12 rows deep.
Water Coils for Cooling—1 to 12 rows deep.
Direct Expansion Coils—for cooling 1 to 10 rows deep.
Refrigeration Coils—all types and sizes.
Special Coils—of various materials furnished on order for special applications.



HORIZONTAL UNIT HEATER

DOWN FLOW UNIT HEATER

HORIZONTAL UNIT HEATERS

Restyled in 1947 to provide up to the minute eye appeal. This renewed line of 12 sizes completely covers the range from 23,000 to 300,000 Btu. All sizes carried in stock. Write for catalog 320.

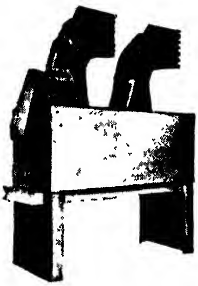
DOWN FLOW UNIT HEATERS

Available with various air discharge arrangements for low, medium, or high suspension, these unit heaters fill a need on almost every job. Carried in stock in 16 sizes from 32,000 to 500,000 Btu. Write for catalog 756.

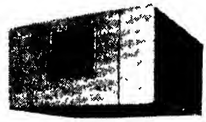
SMALL AIR CONDITIONERS

Cold Water and Freon Types For Small Commercial Applications

Choice of recirculation of indoor air, entire intake of outside air, or a combination of both. Cold water or brine used in one type; Freon or methyl chloride in another. Modern construction assures quiet operation. 2, 3, 5, and 7½ ton sizes. Write for catalog 83A.



BLOWER TYPE UNIT HEATER



AIR CONDITIONER (YEAR-ROUND)

BLOWER TYPE UNIT HEATERS

Made in 8 basic sizes covering the entire range from 750 cfm and 24,000 Btu to 27,000 cfm and 1,659,000 Btu. Provision for handling external static; various outlet diffusers; also available with internal face and bypass dampers.

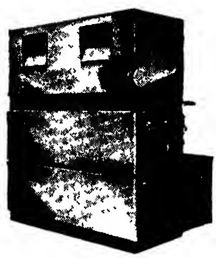
For long life, efficiency, and for eye appeal specify McQuay Unit Heaters. Write for catalog No. 341.

LARGE CENTRAL SYSTEM AIR CONDITIONERS

For Large Industrial and Commercial Applications

Horizontal & vertical types, cools, dehumidifies, filters, and circulates air in summer, heats, humidifies, filters and circulates air in winter. Extreme flexibility and accessibility "built-in." Cooling capacities from 2 to 114 tons in both Suspended and Floor Type. Write for catalog 501.

COMFORT COOLER



AIR CONDITIONER (YEAR-ROUND)

COMFORT COOLERS

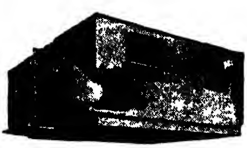
For Small Commercial Applications

Made in two types—one for use with water or brine; another for Freon or methyl chloride. Five sizes (1, 1½, 2, 3, and 5 ton models) in each type—all with variable-speed motors. Write for catalog 81A.

McQUAY

Icy-Flo Accumulators

The new practical "Storage-Battery" for refrigeration effect is now available for handling heavy loads of short duration. Ideal for churches, lodges, mortuaries, noon cafeterias, and many industrial applications. Write for catalog 106.



AIR CONDITIONER (YEAR-ROUND)

ACCUMULATOR

Modine Manufacturing Company

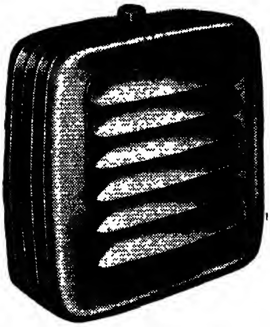
Heating and Air Conditioning Division

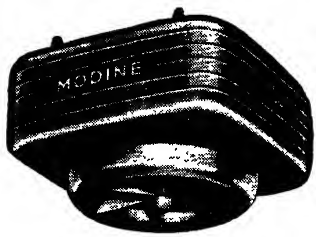
General Offices: 1515 Dekoven Ave., **Racine, Wis.**

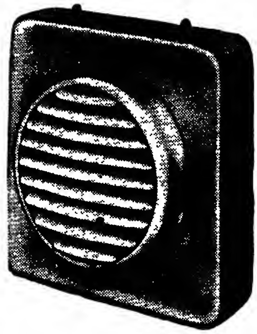
Factories at Racine, Wis., and LaPorte, Ind.

Branches in all Principal Cities

NEW MODINE UNIT HEATER LINE







Horizontal Type
23 Models—for general industrial and commercial applications. Patented Modine center tappings permit direct suspension from pipe.

Vertical Type
16 Models—designed for overhead installation, up near ceilings of high bays to clear plant equipment, or at low levels as in stores and offices.

New Power-Throw
8 Models—for specialized industrial applications. A new type of draw-through horizontal delivery unit heater with scouring jet action.

Three Distinct Types—47 Basic Capacities for Industrial and Commercial Unit Heater Applications

Modine's beautiful, new, integrated line of unit heaters offers new versatility in steam and hot water unit heater application. The three coordinated types may be used individually, or, where requirements indicate, in combination with each other. Thus, it is possible to meet varying heat, air delivery, mounting height, and location demands with the combined action of different unit heaters.

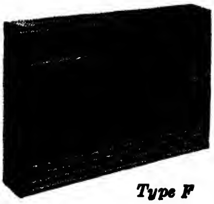
Condensers: Pure copper or copper alloy from inlet to outlet for maximum resistance to internal and external corrosion. Pure copper fins are metallically bonded to round, seamless, heavy-gauge red brass tubes for permanent contact and to insure uninterrupted heat conduction from primary to secondary surface. Modine-patented expansion bend permits tubes to expand or contract individually as temperature requires. All steam and condensate carrying passages are brazed into an integral pressure-resisting unit.

Bonderized Casing: Attractive beige-gray casing with chrome trim protected from rust by Modine Parker Bonderizing.

Quiet Operation: Scientifically sound-silenced for quiet performance. Casing interiors are acoustically-insulated to muffle noises from within. Velocity generator eliminates air rush "peaks," subdues external air-rush noise.

Efficient Motors: Nationally known makes of continuous-duty, totally enclosed fan type. Rubber mounted to prevent vibration noise from being transmitted to casings.

Safety Fan Guard: Staunch, steel safeguard protects against danger of unshielded fan.



Type F

Convector Radiation (Illus.)

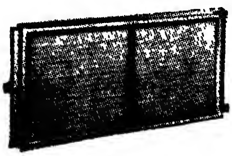
Whether it's a modern apartment, home, school, hospital or office—here is heating equipment styled to match the grace and beauty of modern interior design. Check these features: (1) *New Dual-Purpose Damper,* (2) *Snap-in Lower Grille,* (3) *5-Second Removable Front,* (4) *Convenient Air Venting,* (5) *Versatile Enclosure Design,* (6) *High Capacity Copper Heating Unit.*

For Institutional Use, Modine has combined the tested convection heating ideas with the "specials" most often requested by architects to provide a line of heavy-duty institutional convectors. Available at only a slight additional cost. Modine convectors come in four standard enclosure styles and three institutional models, in a total of 160 modular sizes. All units are Parker-Bonderized for finish protection.

WIDE NEW LINE OF HEATING AND COOLING COILS

The new Modine line of Heating and Cooling Coils now gives you over 3,600 coils to choose from in matching your specific performance and size requirements for heat transfer or cooling surface. But that's not all! Newly engineered and designed, Modine coils permit use of smaller duct sizes for increased convenience and efficiency! Available in eight major types:

1. Standard Heating Coil (Illus.)

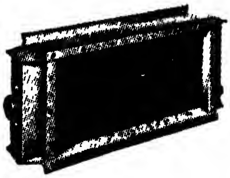


For all normal heating, ventilating, air conditioning and drying applications where steam is heating medium. 595 sizes and models.

2. Non-Freeze Heating Coil (Illus.)

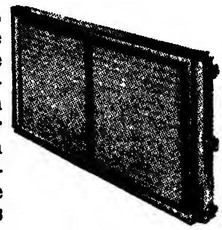
Incorporates steam distributing tubes for uniform face temperature and resistance to freezing. Use where temperature is controlled by modulating steam supply . . . even with below-freezing entering air temperatures. 510 sizes and models.

3. Standard Booster Heating Coil (Illus.)



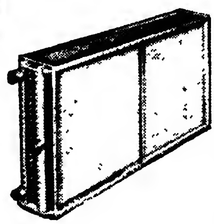
For use where small volumes of air are handled. Ideal for controlling temperatures in branch ducts. Face areas as small as ½ square foot in standard and non-freeze types. 46 sizes and models. Also available in non-freeze type.

4. Hot Water Heating Coil (Illus.)



A serpentine coil for use on hot water. Exclusive Modine feature permits counter-flow installation regardless of air flow direction with complete air venting and drainage provision. 85 sizes and models.

5. Cleanable Water Cooling Coil (Illus.)

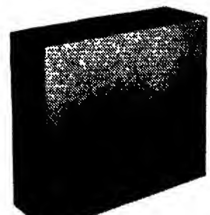


Recommended where occasional tube cleaning is needed. Easy access to each tube is provided by individual pipe plugs along full length of headers. 680 sizes and types. Also available in Standard type.

6. Direct Expansion Coils (Illus.)

For use with Freon. All coils come with multi-circuit distributors for uniform distribution of flash gas and liquid to each circuit under varying load conditions. More than 1000 sizes and types.

Cabinet Unit Heaters (Illus.)



Designed for heating offices, lobbies, corridors, etc. . . . wherever quick, positive distribution of heat combined with quiet operation and gentle air movement are desirable. Has considerably greater output than convector of equivalent size. Employs copper heating coil for use on steam or hot water system. Mounted on wall or ceiling. Capacities: 105 EDR and 310 and 450 EDR.

The Herman Nelson Corporation

General Offices and Factories at Moline, Illinois

Branch Offices and Product-Application Engineers in the Following Cities:

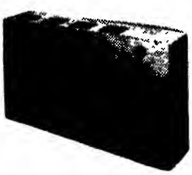
Atlanta, Ga.	Detroit, Mich.	Milwaukee, Wis.	Portland, Ore.
Baltimore, Md.	Duluth, Minn.	Minneapolis, Minn.	Richmond, Va.
Boston, Mass.	Grand Rapids, Mich.	Missoula, Mont.	Saginaw, Mich.
Buffalo, N. Y.	Greensboro, N. C.	Moline, Ill.	St. Louis, Mo.
Cape Elizabeth, Me.	Houston, Tex.	Nashville, Tenn.	Salt Lake City, Utah
Chicago, Ill.	Indianapolis, Ind.	New Orleans, La.	San Antonio, Tex.
Cincinnati, O.	Jackson, Miss.	New York, N. Y.	San Francisco, Calif.
Cleveland, O.	Kansas City, Mo.	Oklahoma City, Okla.	Seattle, Wash.
Columbus, O.	Los Angeles, Calif.	Omaha, Neb.	Spokane, Wash.
Dallas, Tex.	Louisville, Ky.	Philadelphia, Pa.	Springfield, Mass.
Denver, Colo.	Memphis, Tenn.	Phoenix, Ariz.	Syracuse, N. Y.
Des Moines, Iowa	Miami, Fla.	Pittsburgh, Pa.	Washington, D. C.

HERMAN NELSON HORIZONTAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS

Designed for ceiling suspension, these unit heaters project warm air downward, in an angular direction. Copper heating element for use with steam or hot water, incorporates patented stay tube which maintains proper relationship between headers without increasing strain on loops thus prolonging life of unit. A wide variety of models, sizes and arrangements.

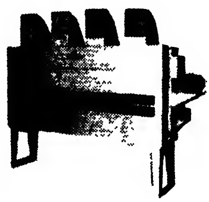
HERMAN NELSON VERTICAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS

For high ceiling installations. Discharge air vertically downward, or at an angle to vertical in various directions. Long life copper heating element for use with steam or hot water incorporates patented stay tube. Units available with either high or low velocity discharge, each with a wide range of capacities.



HERMAN NELSON DE LUXE UNIT HEATERS

Efficient, economical, compact, quiet and attractive, these Unit Heaters provide the ideal method for heating offices, showrooms, corridors, markets, stores, etc. Copper heating element incorporates patented stay tube. Units may be placed on floor, wall or suspended from ceiling. Eighteen models, sizes and arrangements.



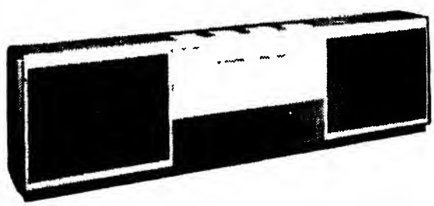
HERMAN NELSON CENTRIFUGAL-FAN TYPE UNIT HEATERS

The Herman Nelson Centrifugal-Fan Type Unit Heater can be applied to solve a multitude of heating and ventilating problems. With 1890 combinations of models, sizes and speeds available, there is a unit to fit the average requirements of commercial and industrial buildings of all types.

HERMAN NELSON UNIT VENTILATORS

The Herman Nelson Unit Ventilator incorporates all of the features of design and construction which make possible the efficient and economical maintenance of desirable schoolroom air conditions. It is quiet, economical, attractive and is designed to permit maintenance of uniform temperatures at all times through gradual throttling of the steam supply.

Integral design of cabinet permits this unit ventilator to be used by itself or as a section of a group including utility cabinets and convectors.



Herman Nelson Unit Heaters and Unit Ventilators are tested and rated in accordance with the Standard Test Code adopted jointly by the *Industrial Unit Heater Association* and the American Society of Heating and Ventilating Engineers

THE HERMAN NELSON CORPORATION

General Offices and Factories at Moline, Illinois

HERMAN NELSON DIRECT DRIVE PROPELLER FANS

Provide most economical form of quality ventilation obtainable for industrial buildings of all types. 7 standard sizes available with wheel diameters from 14 to 36 in. and capacities from 655 to 12,400 cfm. There are 7 high powered models to operate against static resistance of $\frac{3}{8}$ in., with wheel diameters from 14 to 36 in. and capacities from 1200 to 14,600 cfm. Also three models especially adapted to small store and office applications. Standard Models available with two speed motors.

HERMAN NELSON BELT DRIVE PROPELLER FANS

For public and commercial building installations where slow speed, quiet operation are required. Twelve sizes of the standard model with wheel diameters from 24 in. to 54 in. Also six sizes of the High Powered model with the same wheel diameters. Capacities: 5650 to 36,150 cfm. Due to quiet operation of Herman Nelson Belt Drive Propeller Fans, use of two speed motor is unnecessary.

HERMAN NELSON DIRECT DRIVE UNIT BLOWERS

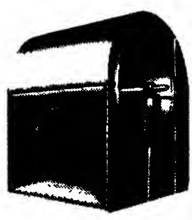
Designed for many applications, such as fume hoods, toilet ventilation, chemical laboratories, industrial processing and drying problems. Compact, direct connected, motor driven units have universal discharge and mount on floor, wall or ceiling. Available in four sizes with 9 speed combinations. Wheel diameters from $6\frac{1}{8}$ in. to 11 in. and capacities from 360 to 2265 cfm.

HERMAN NELSON BELT DRIVE UNIT BLOWERS

Fully self-contained unit including motor, drives and housing; slow speed or non-over-loading type wheels available; adjustable motor pedestal with vibration dampers; universal discharge; nine sizes with 70 drive combinations. Available with any rotation and discharge. Wheel diameters from 11 in. to 30 in. and capacities from 980 to 16,892 cfm.

HERMAN NELSON CENTRIFUGAL FANS

Herman Nelson Centrifugal Fans are designed and constructed for smooth, efficient, long-life operation on any system requiring the use of a Class I or II centrifugal fan. These fans are available in either slow speed or non-overloading type; 17 wheel diameters from $12\frac{1}{4}$ to 73 in.; single or double width; 8 arrangements for direct or belt drive; and any rotation or discharge.



Model BN

Model AN

The Complete Line of Herman Nelson Propeller and Centrifugal Fans is tested and rated in accordance with the Standard Test Code adopted jointly by the *National Association of Fan Manufacturers* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

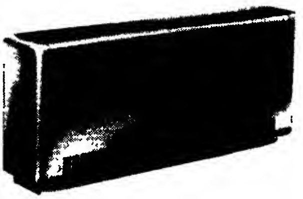
John J. Nesbitt, Inc.

Philadelphia 36, Pa.

Manufacturers of

THE NESBITT SYNCRETIZER Heating and Ventilating Unit, and
THE NESBITT PACKAGE, schoolroom ensemble consisting of the Syncretizer, Convector and storage units, sold by John J. Nesbitt, Inc., and American Blower Corporation;
NESBITT HEATING SURFACE with Dual Steam-distributing Tubes,
NESBITT SERIES H HEATING SURFACE, and
NESBITT SERIES W COOLING SURFACE,
sold by leading manufacturers of fan-system apparatus;
NESBITT CONVECTORS, sold by plumbing and heating wholesalers;
WEBSTER-NESBITT UNIT HEATERS (See page 1335),
distributed in U. S. A. by Warren Webster & Company.

NESBITT SYNCRETIZER—Series 500



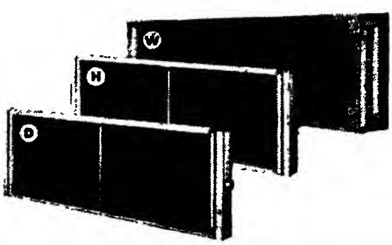
Semi-recessed Model

For heating and ventilating schoolrooms, offices, etc. where the continuous introduction of outdoor air is desired. Incorporates exclusive Nesbitt features: *Comfort Control* provides maximum comfort; *Air Volume Stabilizer* prevents excessive quantities of outdoor air from entering the unit, saves fuel; *Uniform Air Discharge Temperatures* assured by use of Nesbitt Dual Steam-Distributing Tubes inside the Syncretizer radiator; *Directed-Flow Adjustable Outlet* permits the direction of the discharge air to be varied to suit individual classroom requirement. For engineering data, Pub. 261. For data on The Nesbitt Package, school-room ensemble consisting of the Syncretizer, Convector, and storage units. Pub. 258

Nesbitt Series B Thermovent

For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Publication No. 227-2

NESBITT SURFACE



SERIES W. Continuous tube or cleanable tube water surface for air-cooling and dehumidifying with cold water, or air heating with hot water. Copper tubes and plate-type aluminum fins. Wide range of sizes in three types; Type WD sections have exclusive drainability feature and the surface pitched in the casing. Positive drainage insured protecting against winter freeze-ups. Pub. 246. Type WB sections for booster-heating or air-cooling applications with relatively small air volumes and drainability feature unnecessary. Type WC sections employ standard Nesbitt Series W Surface cores, pitched in the casing. Designed for use where tubes require periodic cleaning. Cast iron headers with removable cover plates allow access to tubes. Single or double serpentine circuits in a range of sizes. Pub. 255

SERIES H. General blast coil surface for heating, ventilating, air conditioning and drying in both high- and low-pressure steam systems. Copper tubes and aluminum fins. Seven surface types, full range of sizes. See publication 248.

SERIES D. Heating surface with Steam-Distributing Tubes, ideal for preheating outdoor air. Freeze-proof plus. Uniform discharge temperatures; precise controllability with modulating valves. Copper tubes and headers, aluminum fins. Available in Type DS having a single steam supply, and Type DD having a steam supply at both ends of the section. Publication 247.

SERIES E. For air cooling and dehumidifying with direct expanded refrigerant. Constructed of copper tubes and flat plate-type aluminum or copper fins. Available in a wide range of sizes. Pub. 259

NESBITT CONVECTORS

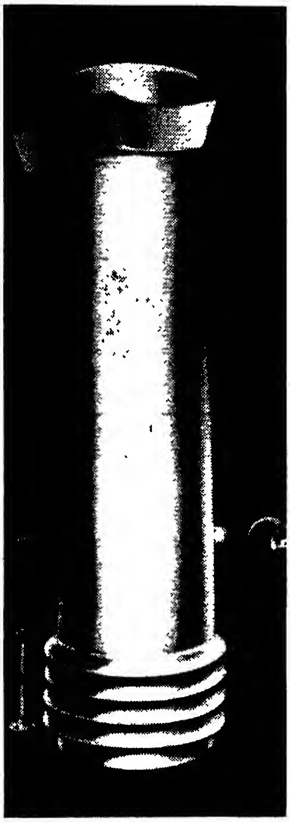
Designed for steam or hot water heating of residences, apartments. Available in 20 stock sizes. Send for Publication 252.

THERMOBLOC

DIVISION

Prat-Daniel Corporation

88 Water Street **East Port Chester, Conn.**



THERMOBLOC

DIRECT-FIRED PACKAGED · UNIT HEATERS

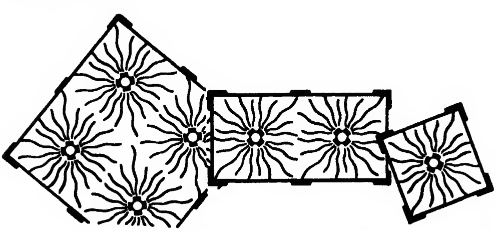
THERMOBLOC is an entirely new, revolutionary, draft free, packaged unit heater. Unique design of outlets permits heated air discharge through a full 360 degree area. Outlet vanes allow horizontal flow control.

THERMOBLOC is shipped ready for installation, with only oil or gas and power line connection necessary to put it into operation. Where larger areas are to be heated, or more heat is required, more than one unit may be installed, either separately or in parallel.

THERMOBLOC affords adequate heating facilities for an entire plant or for additional heating source where needed. For summer, cool air may be drawn from the floor level and circulated at working level.

THERMOBLOC 550 Available from stock. Size: 30 in. diameter, 10 ft high. Output: 550,000 Btu's. Efficiency: 82–86 per cent with #3 oil or gas.

THERMOBLOC 300—This new unit is available immediately from stock. Size: 30 in. diameter, 7 ft high. Output: 300,000 Btu's. Efficiency: 82–86 per cent with #3 oil or gas.



One or more units may be installed as shown at left where greater heating capacity is required.

Information on other applications will be sent on request.

The TRANE Company

2021 Cameron Avenue, La Crosse, Wisconsin

In Canada: Trane Company of Canada, Ltd., Toronto, Ontario

COMPLETE LINE OF HEATING, COOLING, AIR CONDITIONING AND AIR HANDLING EQUIPMENT

Over 75 Trane Sales Offices

Albany, N. Y.
Albuquerque, N. M.
Allentown, Pa.
Amarillo, Texas
Appleton, Wis.
Atlanta, Ga.
Aurora, Ill.
Baltimore, Md.
Billings, Mont.
Birmingham, Ala.
Boston, Mass.
Buffalo, N. Y.
Canton, Ohio
Chattanooga, Tenn.
Chicago, Ill.
Cincincinnati, Ohio
Clarksburg, W. Va.
Clarksville, Tenn.
Cleveland, Ohio
Columbus, Ohio
Dallas, Texas
Davenport, Iowa
Dayton, Ohio
Denver, Colo.
Des Moines, Iowa
Detroit, Mich.
Erie, Pa.
Flint, Mich.
Gainesville, Fla.
Grand Rapids, Mich.
Greensboro, N. C.
Greenville, S. C.
Harrisburg, Pa.
Houston, Texas
Indianapolis, Ind.
Jackson, Miss.
Kansas City, Kans.
Knoxville, Tenn.
La Crosse, Wis.
Lake Charles, La.
Los Angeles, Calif.
Louisville, Ky.
Memphis, Tenn.
Milwaukee, Wis.
Missoula, Mont.
Newark, N. J.
New Orleans, La.
New York, N. Y.
Oklahoma City, Okla.
Omaha, Nebr.
Philadelphia, Pa.
Phoenix, Arizona
Pittsburgh, Pa.
Portland, Maine
Portland, Oregon
Providence, R. I.
Richmond, Va.
Roanoke, Va.
Rochester, N. Y.
St. Louis, Mo.
St. Paul, Minn.
Salt Lake City, Utah
San Antonio, Texas
San Francisco, Calif.
Seattle, Wash.
Sioux City, Iowa
South Bend, Ind.
Spokane, Wash.
Syracuse, N. Y.
Toledo, Ohio
Trumbull, Conn.
Washington, D. C.
West Hartford, Conn.
Wichita, Kans.
Wilkes-Barre, Pa.
Wilmington, Del.
Worcester, Mass.

A COMPLETE LINE

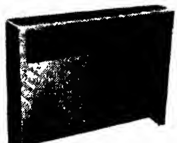
The Trane Company fabricates a complete line of heating, cooling, air conditioning and air handling equipment. Long years of experience with practical knowledge gained from close field contact have developed products for every requirement.

Trane Systems—So comprehensive is the Trane Line that any number of complete heating and air conditioning systems can be designed in which all the major parts are made by Trane. *Examples:* Trane Custom-Air System of Air Conditioning for multiple-room buildings; Central and unit systems of air conditioning for comfort and processes; Steam and hot water heating systems.

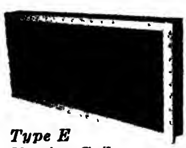
Trane Convector-Radiators—The modern succes-cessor to the old-fashioned radiator, the Trane Convector-radiator is a compact, light-weight, easy-to-install unit. Available for either steam or hot water heating system.

Currently easier to obtain and install than ever, new Type A Units are available from stock throughout the nation.

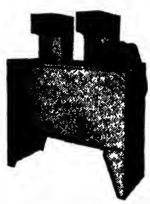
Trane Coils—There are Trane Extended Surface Coils for every heating or cooling, comfort or process application, in all types and sizes. Types include coils for steam, hot water or booster heating, direct expansion or water cooling.



Convector-



Type E Heating Coil



Torridor Unit Heater

Propeller Type Unit Heater

Trane Blower Type Unit Heaters—Better known as Torridors, Trane Blower Type Unit Heaters are available for large space or ductwork applications. Ideal for heating large spaces, exposed areas requiring a blanket of heat, and for process applications.

Trane Propeller Type Unit Heaters—Sides, top and bottom of one-piece wrap-around construction. Grille and fan shroud are attached to complete the unit. Body construction gives greater support to coil and motor—giving cleaner, and handsomer appearance. Coil is protected against expansion and contraction by newly developed floating coil features. Capacities from 20,000 through 352,000 Btu.

Trane Projection Heaters—A Trane development, the Projection Heater taps the usually wasted heat reservoir at the ceiling, bringing it down where it is needed. Available with any of several grille, louver or diffuser arrangements. Exceptionally efficient is the completely adjustable Cloverleaf Diffuser. With this arrangement, part of the air

can be projected downward while a part of the stream can be deflected almost horizontally.

Trane Unit Ventilators—To meet every requirement of ventilation in school classrooms and similar installations. Modernistic cabinets house unit ventilator mechanisms and heating elements. So wide a line is presented that each unit is virtually tailormade for the installation.

Trane Literature—Trane publishes the Trane Air Conditioning Manual ($5.00), unbiased textbook for the engineering profession, and the Trane Refrigeration Manual ($1.50), a reference for the servicing and installing of all types of refrigeration systems. These are non-profit texts, printed to help solve the problems of air conditioning and refrigeration for the engineer, craftsman and layman.

Trane Climate Changers—Trane Climate Changer, a unit type air conditioner, is designed for summer, winter, or year 'round air conditioning, commercial and industrial application, comfort or process installations. Available in various coil combinations with or without humidification equipment.

Trane Refrigeration Equipment—Outstanding in the refrigeration field is the Trane Turbo-Vacuum Compressor, a completely self-contained hermetically sealed centrifugal type water chiller. Constant operation with a minimum of maintenance is assured by the scientific simplicity of this machine.

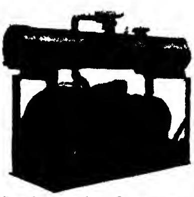
Trane also furnishes a complete line of Reciprocating Compressor and Condensing Units with capacities ranging from 3 to 100 tons. Also available are Trane Self-Contained Air Conditioners for shop and office spaces.

Trane Centrifugal Fans—Recommended for all types of heating, cooling, and air handling applications. In direct or belt-driven units, single or double widths, and all standard discharges in both backward and forward curved blade construction. Capacities 200 to 330,000 cfm.

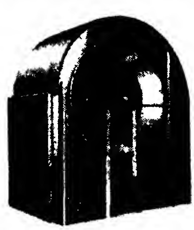
Projection Heater

Climate Changer

Turbo Vacuum Compressor



Reciprocating Compressor



Centrifugal Fan

Lifetime Valve

Trane Steam Heating Specialties—There are over fifty valves, traps, vents, strainers, all allied specialties in the Trane Line. Among them are the famous Trane Hermetic Valve with the Lifetime Diaphragm that absolutely prevents steam leakage around the stem, and the Thermostatic Radiator Trap which together provide an ideal combination for convector radiators.

Trane Hot Water Heating Specialties—Included among Trane Hot Water Heating Specialties are the Trane Circulator, Flo Valves, and Fittings. They combine with Trane Convectors or Unit Heaters to provide an ideal Warm Water Heating System for a great variety of applications.

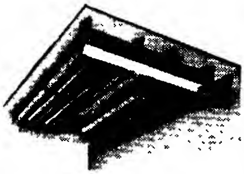
Other Trane Equipment—The complete Trane Line also includes—1. Trane Roof Ventilators—supply and exhaust; 2. Trane Condensation and Centrifugal Pumps; 3. Trane Dry Type Water Chillers; 4. Trane Evaporative Condensers to condense refrigerants in the air conditioning system with a minimum use of water; 5. Trane Cooling Towers; 6. Trane Force-Flo Heater for quiet heat and neat appearance; 7. Trane Railroad and Bus Air Conditioning Equipment of all kinds; 8. Trane Shell and Tube Heat Exchangers for cooling and heating vapors or liquids in a closed system; 9. Evaporative Coolers for cooling fluids in a closed system; 10. Transformer Oil Coolers; 11. Air Washers.

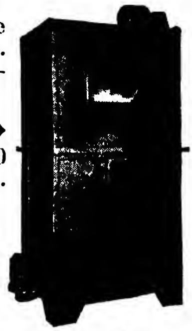
Write today for Trane Condensed Catalog PB290 which describes completely all of the products listed here as well as providing sufficient data for their selection.

Refrigeration Economics Co., Inc.

1231 Tuscarawas St. E., **Canton 2, Ohio**

RECOY PRODUCTS



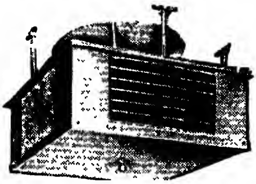


← C. T. COILS

Continuous-tube down-draft fin-coils are still unsurpassed for meat coolers. Others available for practically any application.

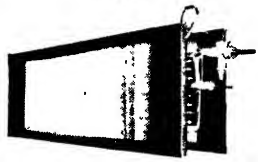
EVAPORATIVE CONDENSERS →

Evaporative condensers from 2 to 100 tons. Brine spray cooling to 25 tons.



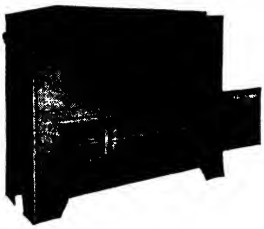
← CEILING DIFFUSER

Ceiling diffusers distribute the cooled air across the ceiling, so the blast does not strike the products stored or occupants.



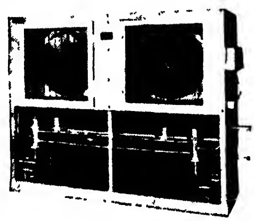
C. F. COILS →

Continuous fin coils for unit coolers, blast heaters, air conditioning and condensers.



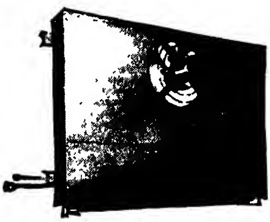
← AIR CONDITIONING

Air conditioning units of ceiling or floor type in all capacities, for cooling, heating, or both.



AUTOMATIC DEFROST → UNITS

Complete, ready for electric, liquid, suction, and hot gas connections. One coil working always, both 98 per cent of time.

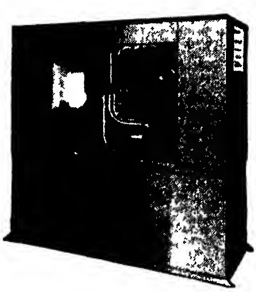


← WALL UNITS

Recoy "All Seasons" wall units provide a damper for deflecting the cold air down along the wall or out horizontally into the room, thus providing proper air circulation for "All Seasons."

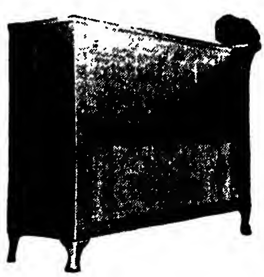
SHELL CONDENSER →

Shell and tube, also shell and fin coil condensers. Both types have tubes arranged for cleaning with tube cleaner.



← WATER COOLING

Self contained complete ice water and brine coolers complete with high and low sides, circulating pumps, controls, and insulation. 1½ to 50 hp.



FLOOR UNITS →

Floor units with cooling surface exposed to view have a definite advantage over those with coils hidden. Design permits water defrosting.

Factories:
NEWARK, N. J.

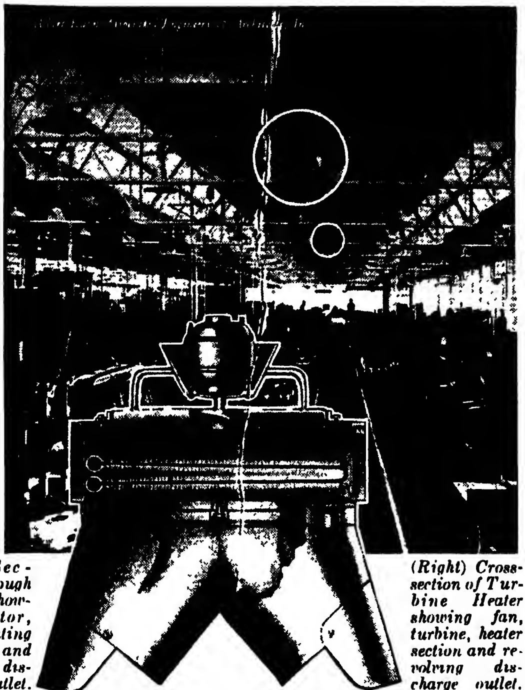
L. J. Wing Mfg. Co.

59 Seventh Avenue, New York 11, N. Y.

Canadian Factory:
MONTREAL

Branch Offices in **Principal Cities**

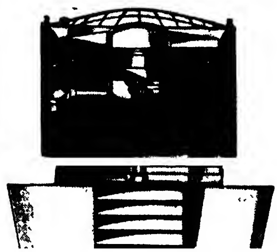
WING REVOLVING UNIT HEATERS



Cross-Section through heater showing motor, fan, heating section and revolving discharge outlet. Design No. 8.

(Right) Cross-section of Turbine Heater showing fan, turbine, heater section and revolving discharge outlet. Design No. 5.

Motor Driven Heater



Turbine Driven Heater

This innovation in the method of distributing heat produces a sensation in heating comfort never before attained—a sensation of fresh, live, invigorating air.

The fact that the outlets revolve assures uniform and thorough distribution of comfortably warmed air throughout the entire working area, without drafts, hot spots or cold spots.

Such an unprecedented high efficiency in distributing heat is the result of 25 years of constant study by Wing engineers to improve on the Floodlight System of heating pioneered by WING in 1921. This method projects the heated air vertically downward by means of light-weight, ceiling-suspended unit heaters.

It has needed only this latest refinement of slowly revolving discharge outlets to bring that method to perfection.

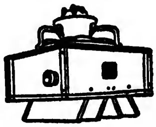
The WING Revolving Discharge type supplements the WING line of standard fixed discharge outlets, illustrated and described on the following page.
Bulletin HR-5.

The latest type of WING Unit Heater - with Revolving Discharge Outlets—is the WING TURBINE DRIVEN UNIT HEATER in which the steam used to drive the fan (instead of an electric motor) also supplies the heater section.

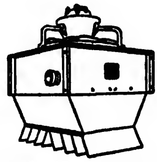
The Wing Turbine Revolving Unit Heater employs the new Wing Allsteel Steam Turbine which operates at *any* pressure. Condensate from the heater is never at a temperature exceeding 170 F. The entire unit is so designed that there is no back pressure on the turbine, assuring against leaks without the use of power-absorbing, troublesome packing. This also eliminates the need for traps and extra piping.

As in the motor-driven heater, the revolving discharge outlet distributes the heat continuously in constantly changing directions.

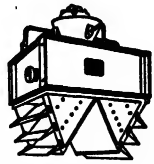
Type "HC" Fixed Discharge Design No. 7



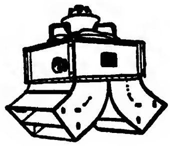
Design No. 1



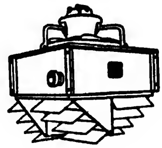
Design No. 1-HV



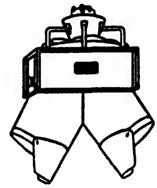
Design No. 3



Design No. 4



Design No. 6



Design No. 8

WING FIXED DISCHARGE UNIT HEATERS

The first light-weight, ceiling-suspended, unit heater. Eight different designs of outlets meet the requirements of every type, size and height of building or occupany. Located near ceiling or roof, the accumulation of hot air in the upper spaces, with the accompanying costly waste of heat, is prevented. They project the air, comfortably warmed, downward to the working area. *Bulletin HR-5*

DOOR HEATERS GARAGE HEATERS

WING developed this vertical cone-discharge heater in 1921 and today it is still applicable for heating the inrush of cold air at large doorways and for garage heating. Often cuts heating costs in half. *Bulletin HR-5.*

FOR LOW CEILINGS

Type "LC"

In this type of WING Unit Heater the position of fan and motor are reversed to meet conditions of ceiling or roof height, form and shape of building, coverage, etc. *Bulletin HR-5.*

WING UTILITY UNIT HEATERS

A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. This is the latest refinement of the original horizontal light-weight heater which was developed by WING. *Bulletin U-6.*

WING GAS FIRED UNIT HEATERS

For natural or manufactured gas. Combines gas burners, heat exchanger and combustion chamber with motor driven Wing fan and discharge outlets. The revolving discharge outlet distributes the heat continuously in constantly changing directions. *Bulletin GH-1.*

FEATHERFIN HEATER SECTIONS

For heating or cooling air for any purpose by steam, hot or cold water or refrigerant. The heating element is extremely light and, for equal heat transfer, offers little resistance to air flow. Available for any desired final air temperature. *Bulletin HS-4.*

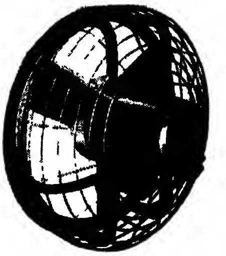
VARIABLE TEMPERATURE SECTIONS

Invaluable in supplying fresh air of varying temperatures for space heating or process work. Close control of the delivered air temperature. Positively will not freeze. Manual or automatic control. *Bulletin HS-3.*

WING INDUSTRIAL FOG ELIMINATORS

Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required. *Bulletin FE-12.*

WINGFOIL SAFETY VENTILATING FANS



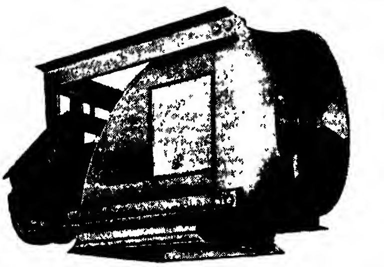
An axial flow fan that will deliver air against static pressure, quietly and efficiently.

Moves the air forward in straight lines with minimum eddy. Capacities to 100,000 cfm. *Bulletin F-9.*

WING FEATHERFIN PROCESS HEATING UNITS

For manufacturing processes such as drying, aging, etc., requiring the recirculation of the heated air. Motor or turbine located outside air current. *Bulletin P-2.*

WINGFOIL DUCT FANS



For economically moving air wherever ducts are used. It combines the efficient WINGFOIL AXIAL FLOW Fan with a housing which places the motor entirely outside the air duct. Motor and drive remain cool and clean and are easily accessible.

The powerful WINGFOIL Fan delivers high air volume with low power consumption against any pressures for which duct systems should be designed. V-belt or direct drive.

Light, compact and easy to install. *Bulletin F-9.*

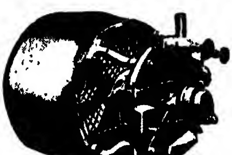
WING SYSTEM OF CONTROLLED COMBUSTION

For low pressure heating boilers and small power boilers. Increases capacity and permits use of lowest cost fuel. Includes Type EM Blower equipped with fully enclosed dustproof motor with speed regulating rheostat and automatic control. Eliminates necessity of frequent firing, allowing intervals as great as 24 hours even in zero weather. *Bulletin M-96.*

Installation of Wing System of Controlled Combustion in a large school

WING TURBINE-DRIVEN BLOWERS

Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels. The exhaust steam, free from oil, can be used for heating or processes. *Bulletin T-98.*



WING DRAFT INDUCERS

Installed in breeching or flue, or on chimney top; provide positive, exact draft regardless of weather conditions or inadequate chimney or breeching construction. Suitable for coal, oil, or gas-fired boilers; industrial furnaces and kilns. *Bulletin I-10.*

Chimney-Top Installation

WING MOTOR-DRIVEN BLOWERS

Type COM for static pressures over 5 in. and volumes up to 35,000 cfm. Type EMD for moderate static pressures up to 5 in. Both blowers have fully-enclosed dustproof constant speed motor and built-in adjustable control vanes. Type COM has double-staged axial flow fan; Type EMD, single stage fan. Extremely compact; discharge can be vertical, horizontal or inclined. *Bulletin SW-1.*

Type COM

Type EMD

Young Radiator Co.

Dept. 179, Racine, Wis.

Sales and Engineering Offices in Principal Cities

YOUNG

HEAT TRANSFER PRODUCTS

Convectors ● Unit Heaters ● Heating Coils ● Cooling Coils ● Evaporators ● Air Conditioning Units ● Gas, Gasoline, Diesel Engine Cooling Radiators ● Jacket Water Coolers ● Heat Exchangers ● Intercoolers ● Condensers ● Evaporative Coolers ● Oil Coolers ● Gas Coolers ● Atmospheric Cooling and Condensing Units ● Supercharger Intercoolers ● Aircraft Heat Transfer Equipment.

'STREAMAIRE" UNITS

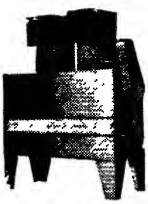
"Streamaire" units are a development of Young's quarter century of experience in building heat transfer products. Personalized, "on-the-job" engineering service by field men—backed by modern research and manufacturing facilities—assures the practical, economical installation as required in your plans.

Type "SH" unit heaters for horizontal air discharge.

Available with capacities from 19,000 to 325,000 Btu per hour.

Type "V" or Vertiflow unit heaters for vertical air discharge.

Capacities from 52,600 to 552,000 Btu per hour.



Type "BH" blower unit heaters for floor, wall or ceiling mounting.

Capacities from 109,400 to 1,047,000 Btu per hour.

Type "W" water coils for cooling or heating with central plant systems.

Five widths, 11 to 35 in.; 2 to 8 rows of tubes; many lengths.

Type "E" evaporator coils for direct expansion cooling systems using Freon or Methyl chloride.

Four widths—one to six rows of tubes; many lengths.

"Streamaire" Convectors Standardized Types—circulate rather than radiate heat. Used with steam or hot water systems.

Cabinets blend with architecture and room furnishings.

"YAC" vertical or horizontal type air conditioning units provide "weather to order" year-round. A constant, quiet flow of evenly distributed clean, healthfully-fresh air at fingertips.

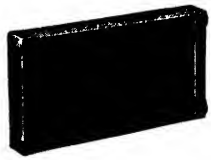
Also available in units for winter or summer conditioning only. Capacities from 400 to 16,625 cfm.

Types "B" and "A" blast coils for central plant heating and air conditioning systems.

Steam distributing tube type available.

Type "C" commercial heat transfer coils for use in factory built air conditioning units.

Steam distributing tube type available. One, two and three rows of tubes.



Revere Copper and Brass Incorporated

Executive Office: 230 Park Avenue, **New York 17, N. Y.**

MILLS—Baltimore, Md., New Bedford, Mass., Rome, N. Y., Detroit, Mich., Chicago, Ill.

SALES OFFICES- Boston, Mass., Providence, R. I., Philadelphia, Pa., Atlanta, Ga., New York, N. Y., Pittsburgh, Pa., Cleveland, Ohio, Cincinnati, Ohio, Grand Rapids, Mich., Milwaukee, Wis., St. Louis, Mo., Indianapolis, Ind., Minneapolis, Minn., Dallas, Texas, Seattle, Wash., San Francisco, Calif., Los Angeles, Calif., Hartford, Conn., Dayton, Ohio, Houston, Texas

REVERE PIPE AND TUBE OF COPPER AND COPPER ALLOYS

For Heating, Air Conditioning, Plumbing, Fuel Lines, Compressed Air Lines, etc.

Revere Copper Water Tube, Types K, L, and M meets Federal and *ASTM* specifications.

Types K, and L furnished in hard and soft tempers.

Types M, 1¼ in. and above, furnished in hard temper only.

Revere Type K Copper Water Tube in hard temper is not too hard to be bent with a hand bender.

Type K soft temper tube is recommended for underground water service or fuel lines.

Hot water lines of Copper Water Tube lose very little heat to ambient air, hence save fuel.

Revere Red-Brass Pipe (Gov't Grade A) or Copper Pipe (both SPS) are recommended for piping systems where threaded connections are required.

For Radiant Heating

Revere Copper Water Tube, furnished in 60 ft. coils is easily bent to form sinuous coils for heating panels.

Long, one-piece lengths of copper tube reduce the number of couplings or joints required.

Small diameters of copper tube require less thickness of embeddment in plaster. **Revere Dryseal Copper Tube** is dehydrated and sealed. It is commonly used for Refrigeration and Air Conditioning Systems, fuel lines, compressed air lines, and general service work.

Furnished in dead soft temper, it is easily bent and flared.

For Condensers and Heat Exchangers

Revere Cupro-nickel condenser tube has definitely been found superior for condensers, after coolers, and similar heat exchangers.

Similar tubes of Revere Admiralty Metal are widely used.

Revere Seamless Copper Tube is commonly used for finned tube coils.

For Industrial Piping and the Process Industry

Revere produces a wide range of pipe and tube made of copper and copper alloys for industrial use where high resistance to corrosion is required.

Solicitations for assistance in selecting piping material best suited to specific conditions are welcome.

Silver-brazed Joints

Revere Red-Brass Pipe or Copper Pipe is recommended where silver-brazed joints are required with standard pipe sizes.

Revere Copper Water Tube and standard soldered type fittings can also be silver-brazed satisfactorily and generally at less cost than heavier pipe.

Technical Advisory Service

Revere maintains a staff of technical men to assist engineers, designers, and contractors in the selection of suitable Revere products for various applications. Their services are available without obligation.

Technical Literature

Literature relating to many fields of application for **Revere** pipe and tube products are available upon request.

Two booklets on Radiant Panel Heating cover the subject of design procedure and in the form of a non-technical and unbiased discussion for lay readers.

Revere Copper Water Tube

STANDARD DIMENSIONS AND WEIGHTS

Size In In.	Type K			Type L		Type M	
	O.D in In.	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft
¼	.375	.032	.134	.030	.126		
⅜	.500	.049	.269	.035	.198		
½	.625	.049	.344	.040	.285		
⅝	.750	.049	.418	.042	.362		
¾	.875	.065	.641	.045	.455		
1	1.125	.065	.839	.050	.655		
1¼	1.375	.065	1.04	.055	.884		
1½	1.625	.072	1.36	.060	1.14		
2	2.125	.083	2.06	.070	1.75		
2½	2.625	.095	2.93	.080	2.48	0.65	2.03
3	3.125	.109	4.00	.090	3.33	.072	2.68
3½	3.625	.120	5.12	.100	4.29	.083	3.58
4	4.125	.134	6.51	.110	5.38	.095	4.66
5	5.125	.160	9.67	.125	7.61	.109	6.66
6	6.125	.192	13.9	.140	10.2	.122	8.92

The American Brass Company

General Offices: Waterbury 88, Conn.

District Offices in Principal Cities

ANACONDA
from mine to consumer

IN CANADA: ANACONDA AMERICAN BRASS LIMITED, NEW TORONTO, ONTARIO

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe. These advantages may briefly be summarized as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a lower resistance to flow. In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Anaconda Solder Fittings are compact. They can be installed in restricted space where the use of a wrench would be impossible.

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping. Anaconda Copper Tubes can be installed with a minimum of cutting in the structure—although holes should be large enough to permit movement of tubes due to expansion and contraction.

Temper and Thicknesses—Anaconda Copper Tubes are made in both hard and soft temper and in standard wall thicknesses.

They meet the requirements for these types of tubes in Federal Specification WW-T-799a and *A.S.T.M.* Specification B88. Type K, the heaviest, is recommended for heating lines and general piping.

Accuracy of Dimensions—Anaconda Deoxidized Copper Water Tubes are all finished to the close size tolerances required by the *A.S.T.M.* and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

Anaconda Copper Tubes, in standard sizes are furnished soft in 60-ft coils; also hard and soft in 20-ft straight lengths.

The American Brass Company

REFRIGERATION TUBING

Anaconda Dehydrated Copper Refrigeration Tubes are manufactured in accordance with *A.S.T.M.* Specification B68, in all standard sizes up to and including ¾ in. O.D., in 50-foot coils. Other lengths are made to special order. These tubes are manufactured under exceptionally clean mill conditions and close technical control to assure clean, smooth inside surfaces, unusual accuracy in size and shape, and uniform softness. The tubes are cup-sealed immediately after annealing and dehydrating.

ANACONDA "85" RED BRASS PIPE

Anaconda "85" Red Brass Pipe, in standard pipe sizes, is considered the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines.

Anaconda "85" Red Brass Pipe contains 85 per cent copper and conforms to Government specifications for Grade "A" water pipe. The mark "Anaconda 85" is stamped in the metal at one-foot intervals throughout each length.

EVERDUR*

Everdur Metal is the original copper-silicon alloy group. It is manufactured by The American Brass Company in five standard compositions and in practically all commercial forms.

This high strength engineering metal is resistant to a wide range of corroding agents. Because of a versatile combination of useful properties, Everdur has become standard as a materal for equipment in many fields of engineering and industry.

In addition to their non-rusting properties and high strength, Everdur alloys possess many qualities not usually found in metals of this character. They are unusually resistant to general atmospheric conditions and other normally corrosive factors. Everdur alloys have excellent machining and working characteristics and can be fabricated into a variety of forms and shapes. Everdur alloys are available for oxy-acetylene or carbon arc welding.

* "Everdur" is a trademark of The American Brass Company registered at the U. S. Patent Office.

CORROSION RESISTANCE

The corrosion resistance of Everdur is equivalent to that of pure copper and in some cases, slightly superior.

However, like copper and all copper alloys, Everdur is not equally resistant to all corroding agents, nor to the same corroding agents under all conditions. As with copper, the resistance to corrosion may be substantially reduced in some instances by the presence of oxidizing agents. Nevertheless, Everdur does offer excellent resistance to the corrosive action of many solutions and atmospheres.

Everdur Tanks—Everdur copper-silicon alloy is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to large storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes including annealed tank plates which have physical properties as given in *A.S.T.M.* Specification B96.

Minimum specification requirements for hot rolled-and-annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 per cent elongation under load) 18,000 psi.; Elongation, 40 per cent in 2 inches.

Welds made with annealed Everdur tank plates meet the requirements for U68 and U69 construction in the *A.S.M.E.* Code for Unfired Pressure Vessels.

For additional data and names of fabricators address our nearest office or agency.

EVERDUR FOR AIR CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may be substituted for steel and fabricated by substantially the same methods and with much the same equipment as steel.

Everdur metal has been used with marked success for fans and blowers, ducts, humidifiers, cast and wrought parts of other equipment items subject to corrosive influences.

EVERDUR LITERATURE

Descriptive literature containing much pertinent tabular data will be sent on request.

Mueller Brass Co.

Port Huron, Mich.

Branch Offices and Representatives in Principal Cities

Albany, N. Y.
Atlanta, Ga.
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Cleveland, Ohio
St. Louis, Mo.
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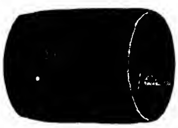
N. S. Pittsburgh, Pa.
San Francisco, Calif.
Tacoma, Wash.
Washington, D. C.

Canadian Sales and Manufacturer

Mueller Ltd., Sarnia, Canada

PRODUCTS—STREAMLINE Copper Pipe and Seamless Tubes; STREAMLINE Hard Copper Pipe and Solder Fittings; Valves, Flared and STREAMLINE Solder Fittings for Mechanical Refrigeration; Forgings of Brass, Bronze and Copper; Castings of Brass and Bronze; Rod; Screw Machine Products; Fabricated Parts and Special Nickel and Chromium Plated Parts; Machined Formed Tubes.



Coupling Copper to Copper

Copper to Outside I.P.S.

45 Deg Elbow

Streamline Copper Pipe and Fittings for heating, plumbing, air conditioning and industrial use are made by the Streamline Pipe and Fittings Division, Mueller Brass Co., Port Huron, Mich.

The **Streamline** Solder Fitting is the original solder type fitting, introduced and manufactured by the Mueller Brass Co. of Port Huron, Mich. It incorporates many advantageous features and has proved to be the revolutionary advance of the age in the development of piping systems for plumbing and heating and for many industrial uses.

The **Streamline** Solder Fitting is not connected either by threading or flaring, but by soldering. The outside surface of the copper pipe and the inner surface of the **Streamline** fitting are cleaned with sandcloth, and solder flux is then applied to the cleaned surfaces to eliminate oxidation when the assembled joint is heated. The joint is then sufficiently heated with a blow or acetylene torch and the soldering operation is performed by feeding wire or stick solder through the feed hole in the fitting.

The **Streamline** Cast Bronze Solder Fitting alone has the solder feed hole. When solder is introduced through the feed hole into the pre-heated joint, it is distributed evenly and thoroughly between the bonding surfaces, traveling inward to the joint and outward to the edge of the fitting, where it appears as a continuous solder ring around the full circumference of the pipe. This ring and feed hole, completely filled with solder, constitute positive, visible proof to the operator, that the joint is permanently leak-proof.

We also manufacture Wrought Copper Fittings in all popular demand sizes and a complete line of Compression Stops, Globe and Check Valves. Fittings are supplied with or without solder feed holes.

Streamline Pipe and Fittings Division

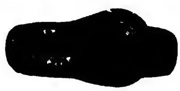
MUELLER BRASS CO.

Port Huron, Mich.

Patents 1770852; 1776502

Tee

90 Deg Elbow



Tee
Copper to Inside I.P.S.

Crosses

The solder may be fed from any position, whether the feed hole is located at the top, side or bottom. Owing to the never failing phenomena of capillarity, the solder will flow up, down or laterally with equal facility.

Streamline Copper Pipe is a seamless cold drawn copper tubing conforming to *A.S.T.M.* B88. It is made in sizes ¼ in. to 12 in. and Types K, L, and M, of which Type K is the heaviest. The intermedite weight, Type L, is the preferred weight for plumbing, heating and refrigeration. Type M is manufactured only in sizes 2½ in. and larger.

For most purposes hard drawn pipe is used, though Types K and L can be furnished annealed when bending is required. Annealed Type "K" in sizes up to 2 in. is widely used for underground water "services."

Streamline Solder Fittings are furnished in all sizes to 6 in. inclusive. They are of the same thickness as Navy and MSS Fittings for 125 steam or 175 lb non-shock water pressure.

All fittings over 6 in. are flanged and may be had with either *A.S.A.* or riveted pipe standard flanges.

Mating flanges are soldered to the pipe—*A.S.A.* 125 standard flanges are available from 1 in. up.

During the last fifteen years architects and engineers have used **Streamline** Copper Pipe and Fittings successfully in every type of building construction and in thousands of installations throughout the United States and Canada.

In addition to its rust and vibration-proof qualities and long life, **Streamline** has many other advantages such as the reduction in size of pipe lines and radiator connections from those nominally used, a neat, compact installation requiring a minimum of space and important advantages in industrial and drainage applications. There is a **Streamline** product for every piping requirement.

Write Mueller Brass Co., Port Huron, Michigan, for complete information and catalog.

Chicago Metal Hose Corporation

Maywood, Illinois

PLANTS

MAYWOOD, ELGIN, and ROCK FALLS, ILLINOIS

District Offices

Atlanta Boston Chicago Cleveland Detroit Ft. Worth Los Angeles New York Philadelphia Pittsburgh San Francisco

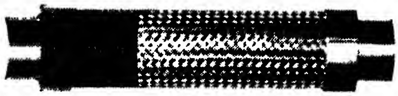
Distributing Outlets in Principal Cities

In Canada: Canadian Metal Hose Company, Ltd., Brampton, Ontario

REX Super Service Vibra-Sorbers Noise and Vibration Control

Rex Vibra-Sorbers control vibration and reduce noise in refrigeration and air conditioning machinery. All-metal construction is liquid- and gas-tight, withstands high pressures, does not age, and has high corrosion resistance.

Available in copper bearing alloy for use with Freon or Methyl Chloride; or steel for Ammonia systems.



SIZES: $\frac{3}{16}$ in. to 4 in. inside diameters.

BURST PRESSURES: 1,000 psi to 3,700 psi.

LENGTHS: Standard stock lengths. Special lengths available on order.

COUPLINGS: Stock units with male or female sweat fittings; also available with male pipe thread fittings.

REX-TUBE

Flexible Metal Hose

FOR

Diesel engine exhaust lines

Refrigeration tubing armor

Air blower ducting

Ventilating ducts

Control wire casing

Wiring conduit

Suction hose

General utility hose

Rex-Tube Convoluted Flexible Metal Hose Types have three basic formation patterns: square-locked, ball-bearing (or double-groove), and fully interlocked. Made of stainless steel, brass, steel, aluminum, bronze and other alloys. Packless and packed types. Sizes range from $\frac{3}{16}$ in. to 12 in. inside diameters, and lengths to suit requirements.

REX-WELD

Flexible Metal Hose

FOR

Steam Hose

Reciprocating flexible connections

Refrigerant loading, unloading and charging

Oil burner connections

Pressure lubricating lines

Conducting searching gases and liquids

Diesel engine exhaust lines

Misalignment correction

Rex-Weld hose types are manufactured from uniform wall tubing by a special CMH corrugation-forming process. Metals used are steel, bronze, and other alloys. Rex-Weld sizes range from $\frac{3}{16}$ in. to 12 in. inside diameter; with lengths and couplings to fit specific requirements. Especially designed for use under high temperatures and pressures, and where corrosive action is present.

FLEXIBLE METAL HOSE FOR EVERY USE

PACKLESS

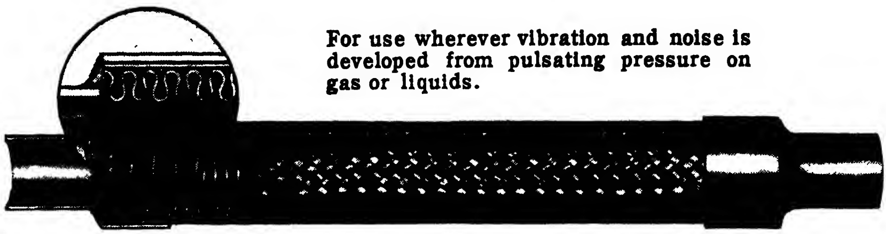
METAL PRODUCTS CORPORATION

31 Winthrop Avenue, **New Rochelle, New York**

PACKLESS VIBRATION ABSORBERS

for the Air Conditioning and Refrigeration Industries

For use wherever vibration and noise is developed from pulsating pressure on gas or liquids.



Packless Vibration Absorbers are furnished with standard copper tube "sweat fittings" thoroughly welded on at both ends. Plain, straight end-extensions are provided to slip over any standard piping on air conditioning and refrigerating lines.

Packless units, covered, with high-tensile bronze braid, afford an extra margin of safety in absorbing vibration and resultant noise.

These units can be furnished in various sizes and lengths for any commercial, industrial, or domestic refrigerating or air conditioning unit. Through years of experience, Packless correlates the proper relationship between hose length and hose diameter to assure maximum efficiency.

SPECIFICATIONS

Female Copper Tubing Ends To
Solder Over Standard Copper Water Tube

Flexible Hose I D.	I.D.	O.D.	Wall	A Length	Water Tube Size (Nominal)	Packless Part No.	B Fexible Hose Length	C Overall Length With Female Ends
3/16"	.255	.319	.032	1/2"	1/8"	VAF-1	6"	7"
1/4"	.255	.319	032	1/2"	1/8"	VAF-2	6 1/2"	7 1/2"
5/16"	.380	.444	.032	5/8"	1/4"	VAF-3	7"	8 1/4"
3/8"	.505	.603	049	3/4"	3/8"	VAF-4	7 1/2"	9"
1/2"	.630	.728	049	7/8"	1/2"	VAF-5	8"	9 3/4"
5/8"	755	.853	049	1"	5/8"	VAF-6	8"	10"
3/4"	.755	.853	.049	1 1/8"	5/8"	VAF-7	9"	11 1/4"
3/4"	.880	1.010	.065	1 1/4"	3/4"	VAF-8	9"	11 1/2"
1"	1.130	1.260	065	1 1/2"	1"	VAF-9	10"	13"
1 1/4"	1.380	1.510	065	1 5/8"	1 1/4"	VAF-10	11 1/2"	14 3/4"
1 1/2"	1.630	1.774	.072	2"	1 1/2"	VAF-11	13"	17"
2"	2.130	2.296	.083	2 1/2"	2"	VAF-12	15"	20"
2 1/2"	2.630	2.820	.095	3"	2 1/2"	VAF-13	18"	24"
3"*	3.130	3.348	.109	3 1/2"	3"	VAF-14	20"	27"

* 3 1/2" I D. and larger—data given upon request.

Packless can furnish proved and pretested Vibration Absorbers to fit most normal requirements. Special requirements of diameter, length, or strength, involving either regular or specialized manufacturing procedure, can also be supplied on request. Packless Engineers are available to aid in solving difficult applications.

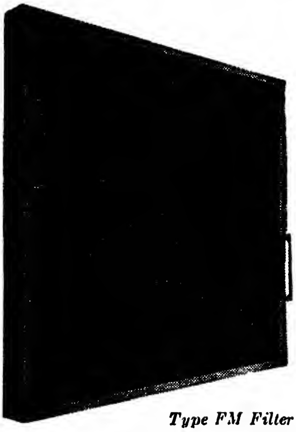
Air Devices, Inc.

Air Diffusers ● Exhausters ● Air Filters
Filter Holding Frames ● Hot Water Generators

17 East 42nd St.
New York 17, N. Y.

Agents in All Principal Cities

AGITAIR AIR FILTERS



Type FM Filter

HIGH VELOCITY ● ALL METAL PERMANENT ● CLEANABLE

Designed along entirely new air filtering principles, the high velocity Agitair Type FM permanent, cleanable air filter assures an amazingly high dust arresting efficiency and dust-holding capacity coupled with sustained low resistance to air flow. This permits the Type FM to remain in service from two to three times as long as ordinary 2 in. permanent, cleanable filters.

Although these new filters do not have to be cleaned as often, particular attention has been paid to their design to make cleaning easier and more thorough. They can be restored to top efficiency easily and quickly. Ruggedly constructed to withstand the mechanical abuse of cleaning. Panels and frames are accurately designed to prevent leakage around the filters.

HOW IT WORKS

High turbulence of many finely divided air streams is the keynote of Type FM air filter's new design. The media divides the air into countless fine streams and throws those streams into violent cyclonic turbulence. Each little "cyclone" centrifuges its dirt particles against countless viscous-coated "wiping surfaces" which virtually scrub the air clean by catching and holding the dirt. There is no straining action, hence no clogging.

HIGH VELOCITY

The Agitair FM Filter is designed to perform at highest efficiency at an approach velocity of 432 fpm—or 1200 cfm through a 20 x 20 in. filter panel. The efficiency of the FM is higher than conventional filters when operating at the lower design velocity of 288 fpm.

1/3 LESS SPACE REQUIRED

The ability of the FM to filter, with greater efficiency, 50 per cent more air at the high velocity of 432 fpm reduces the number of filter panels required. Now TWO FM's will do the work of THREE ordinary filters . . . ⅓ less space required . . . fewer units to be installed . . . fewer units to be serviced . . . overall installation and maintenance costs reduced to a minimum.

HIGHER EFFICIENCY

At the recommended velocities of other leading all metal viscous type filters, the Agitair FM has a higher dust arresting efficiency, which increases as velocities are stepped up.

LOWER RESISTANCE

The sustained low resistance of the Agitair FM means sustained peak volume of air for longer periods of time . . . no loss in air volume . . . no danger of unloading . . . clean filtered air at all times.

HIGHER DUST HOLDING CAPACITY

Employing a new formula for air filtration the new Agitair FM holds more dirt, from two to six times as much as ordinary 2 in. permanent, cleanable filters. No early clogging of air passages . . . less frequent servicing . . . lower maintenance cost.

LONGER SERVICEABLE LIFE

The Agitair FM Filter with its greater dust holding capacity, stays in service for longer periods of time; gives efficient performance for months instead of weeks or weeks instead of days.

Less frequent servicing—and rugged construction combine to give the Agitair FM Filter a much longer serviceable life.

TWO TYPES OF HOLDING FRAMES

Individual type: Designed and constructed for easy handling in single unit installations, and to facilitate "on the job" assembly, into a multiple unit bank.

Pre-Fabricated Type: Delivered completely knocked down for easy assembly, this Agitair holding frame has been especially designed for installations requiring an unusually large number of filter panels and where cramped and unusual conditions place a limitation on available space.

GREASE FILTERS

An efficient, all-metal grease catching, grease holding media. Available in all sizes.

Air Filter Corporation

108G North Water St. **Milwaukee 2, Wis.**

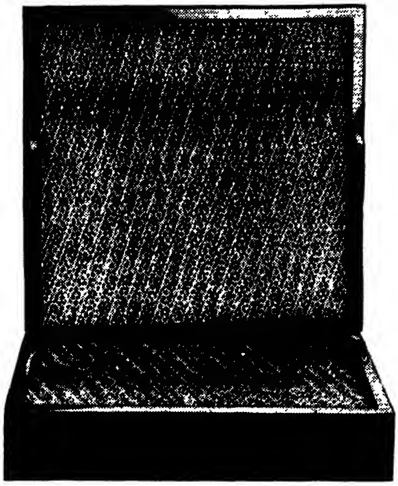
Canadian Representative
DOUGLAS ENGINEERING CO., LTD. MONTREAL

Airsan

(formerly Aircor)

AIR FILTERS **GREASE FILTERS**

Permanent-Cleanable



AIRSAN AIR FILTERS

Industrial Domestic Commercial

Airsan's expanded metal face plate acts as a lint arrestor to provide easier cleaning and servicing. It distributes air evenly over entire filtering area providing high filtering efficiency and dust-holding capacity with low resistance. Media is viscous type, permanent, cleanable, and is constructed of multiple layers of galvanized wire mesh to give maximum air resistance. All Airsan Filters have full bronze welded corners, galvanized steel frames and drain slots for quicker, easier cleaning.

Airsan Air Filters are available in standard 1 in. and 2 in. thickness—Bulletin 1245A. Also HEAVY DUTY filters for industrial and special applications in 2 in. to 4 in. thickness—Bulletin 146.

ENGINEERING DATA

Initial Resistance		*Rated Efficiency*
Type F_1 (1 in. thick)	.060 in. w.g at 288 fpm	98.5%
Type F_2 (2 in. thick)	.065 in. w.g at 288 fpm	98.5%
Type D_2 (2 in. thick)	.09 in. w.g. at 288 fpm	98.5%
Type D_4 (4 in. thick)	.10 in. w.g. at 288 fpm	98.5%

AIRSAN HOLDING FRAMES

Made of heavy gauge metal complete with fireproof felt seal and locking device. Available in straight and V-banks. Pre-fabricated with Airsan slip-groove construction—eliminates felt between filter frames and cuts installation costs. Built to your specifications. Bulletin 601.

AIRSAN GREASE FILTER

Permanent cleanable type **Airsan** Grease Filters specially designed for range canopies, galleys, kitchens. Removes grease at source, reduces fire hazard in exhaust ducts and prolongs life of fans, motors and other mechanical equipment. Assemblies for mounting on ceiling or wall, single or multiple units—includes holding frames, supporting angles and end seals. Bulletin 346.

Initial Resistance: .07 in. w.g. at 216 fpm
Efficiency Rating: 98.5% Stand.
Thickness: 2 in.

Write AIR FILTER Corp. for Complete Bulletins

AIR-MAZE
Corporation

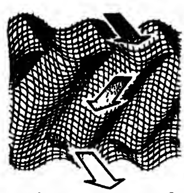
5200 Harvard Ave., **Cleveland, Ohio**

THE FILTER ENGINEERS

Representatives in all principal cities

All Metal Washable Impingement Filters—Air-Maze viscous impingement washable type air filters constructed of alternate layers of crimped galvanized screen cloth of various mesh enclosed in rugged all metal frames are the engineered result of over 20 years experience in air filter design. The viscous impingement filtration principle involves the break-up of the air stream into minute swirling currents by filter media and the resulting impingement of entrained dirt particles on wire baffles previously coated with adhesive.

The viscous impingement principle is scientifically used by Air-Maze in a filter media of progressive density design shown at left which permits lint and large particles to be collected on upstream section of the filter while progressively smaller particles are collected in interior of media resulting in better performance and greater periods between servicing.



The impingement principle is also used by Air-Maze in media design shown at right where layers of screen cloth are crimped and placed to obtain maximum performance with minimum resistance to air flow thus allowing greater air volume to be passed through filter.

Types available—Air-Maze design has provided many types of filter panels for various ventilating applications. Each type is suited best for its application, however, correct selection involves a compromise between ideal and practical considerations. The various types and their applications are shown on page at right.

Efficiency—Efficiency of dirt arrestance of various filter panels varies with filter design, type of dust, amount of dust, method of charging filter with adhesive, etc. Specific information on any type of filter furnished on request.

Resistances—Filters available with initial resistances as low as .045 in. water at face velocity of 300 fpm. Resistance versus velocity curves available on request.

Face Velocities—Air-Maze offers filters designed to operate at peak efficiency at velocities ranging from 100 fpm to 700 fpm. Recommended velocities for each type of filter furnished on request.

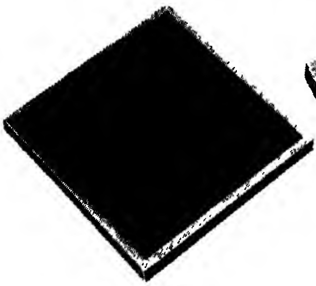
Sizes—Filter panels furnished in accepted standard sizes and in special rectangular sizes if required. Thickness varies with filter type. Most types furnished in both 2 in. and 4 in. thicknesses but available in certain special thicknesses if required.

Holding Frames—Holding Frames, complete with neoprene seals and choice of locking devices, available for use either singly or drilled for assembly into panel bank.

Cleaning and recharging filters easily cleaned by washing in hot water containing commercial grease solvent or with steam. Recharged by immersing in special adhesive or SAE 30–50 oil. Complete instructions available on request.

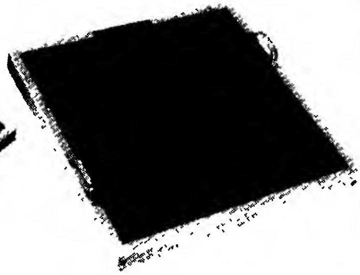
Write factory for name of nearest representative on information on any type of filter. Representatives are in most principal cities. See classified section of your telephone directory.

AIR-MAZE CORPORATION



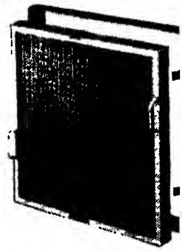
KLEENFLO

Filter for average duty residential and commercial ventilating applications. Available in 1 in., 2 in., and 4 in. thicknesses.



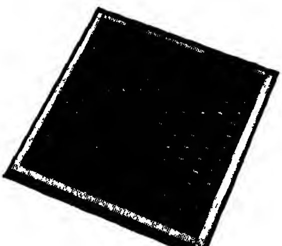
TYPE P-5

High velocity panel for handling a large volume of air at low resistance. Good in commercial installations where space is limited. Available in 2 in. thicknesses only.



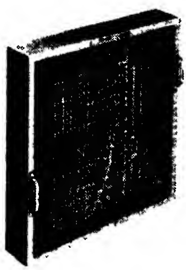
TYPE "A"

Heavy duty industrial filter for ventilating service where dust load is exceptionally high. Incorporates both high efficiency and high dirt holding capacity features. Available in 2 in. and 4 in. thicknesses.



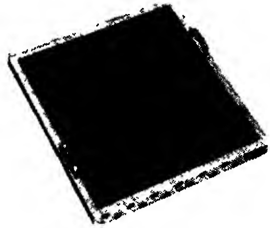
TYPE P-5-R

High velocity panel similar to type P-5 but of heavier constuction necessary for railroad applications. Available in 2 in. thicknesses only.



TYPE 'B"

Average duty industrial filter for fresh air intakes. Has large dirt holding capacity and high efficiency. Available in 2 in. and 4 in. thicknesses.



GREASTOP

Filter designed to prevent grease nuisance and fire hazard in commercial kitchen ventilating systems. Conforms to specifications recommended and approved by National Board of Fire Underwriters. Available in 2 in. thicknesses only.

Air-Maze Corporation manufacture a complete line of viscous impingement type ventilating air filters to meet your requirements within a wide range of limits of pressure drop, velocity, efficiency, weight, dirt holding capacity and price. A few types are illustrated on this page. THESE FILTERS ARE ALL-METAL, WASHABLE, FIRE RETARDENT.

AMERICAN AIR FILTER COMPANY INC.

673 Central Avenue, **Louisville, 8 Ky.**

In Canada: Darling Brothers, Ltd., Montreal, Quebec

PRODUCTS: The American Air Filter Company, Inc. manufactures a complete line of Air Filtering Equipment including Electronic Air Filters, Automatic Self-Cleaning Filters, Viscous Unit Filters, and Dry Process Filters to be used for the removal of dust, soot, smoke, dirt, bacteria and other foreign matter from the air.

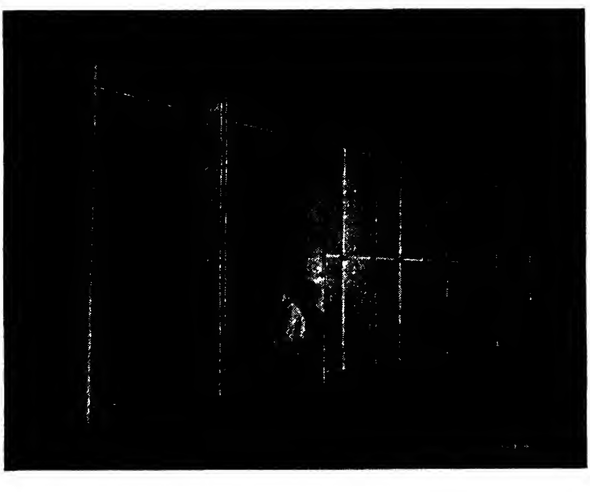
Because air filtration has proved to be the economical, practical and efficient method of cleaning the air, the modern air filter is now considered an absolute necessity in building ventilation or air conditioning, in the maintenance of health and personal efficiency, protection of interiors and furnishings and valuable merchandise and equipment. Shown here are but a few American Air Filter products in most general use.

The American Air Filter Company is recognized as an authority on dust problems and their satisfactory solution. Products offered embody the knowledge accumulated from twenty-six years of intensive research devoted exclusively to the study of dust problems and the development of air cleaning apparatus; the experience gained from designing, building and applying thousands of air filters; are backed by ample technical and financial resources; and may be relied upon as the most modern equipment in their field of service.

THREE TYPES OF ELECTRONIC AIR FILTERS

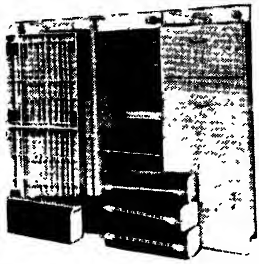
For more than ten years, research and experimentation with electronic air filtration have been in progress by AAF engineers. Today's complete line includes the self cleaning Electro-Matic introduced in 1939, the washable Electro-Cell with removable collector plates, and the Electro-Airmat with the replaceable Airmat paper medium. Here for the first time is high efficiency air cleaning in three types of electronic filters to meet any requirements for super clean air.

Electro-Matic Self-Cleaning Electronic Filter—The Electro-Matic filter is a self cleaning electric precipitator combining the most advanced principles of electronic air cleaning with notable improvements in construction details and method of operation. The self-cleaning feature is an exclusive advantage of the Electro-Matic filter. It eliminates the necessity of shutting down the filter for manual cleaning, minimizes the need for personal attention and permits continuous high-efficiency operation. It also allows the Electro-Matic filter to be built in standardized self-contained sections, easy to install and with all exposed parts of the filter casing electrically grounded for the protection of operating personnel. Send for Bulletin No. 250.

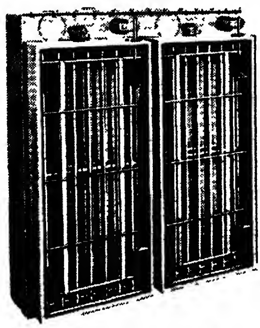


Electro - Cell Electronic Filter—Incorporates all of the new developments of major importance in electronic air filtration using collector plates. Installation has been simplified, performance improved and maintenance advantages provided. Built in vertical sections of two widths—2 ft and 3 ft over-all. Collector plate assemblies are removable; ionizers are hinged and extend the full height of the sections, preventing current loss. A choice of washing collector plate assemblies while in place, or removing assemblies for individual cleaning. Installation is simplified because filter is built in vertical sections rather than assembled of small units. Parts to be lifted and aligned are light in weight and electrical connections simplified. Write for Bulletin No. 252.

Electro-Airmat Electronic Filter—The application of an electrostatic charge to a dielectric filtering material is an exclusive AAF research development which began early in 1934. Airmat paper is composed of a number of plies of porous tissue-like cellulose sheets. When electrically charged the plies tend to separate and each individual fibre becomes a collecting electrode which attracts and holds the dust and smoke particles. Ease and convenience of maintenance is a desired advantage of the Electro-Airmat. Requires neither water nor sewer connections for cleaning, nor spraying with oil to maintain its efficiency. When Airmat paper has accumulated its dust load it is removed and replaced with clean material by means of a mechanical loader. In case of power failure, filter media provides best mechanical air filtration known. Send for Bulletin No. 253.



Electro-CELL Filter



Electro-AIRMAT Filter

American Multi-Duty Self Cleaning Filter

American Multi-Duty Automatic Filter provides outstanding features of performance and design and will accomodate either armored screen panels or die stamped louver panels available in three types. Offers advantage of uniformly constant air supply, fixed operating resistance and automatic operation. Ideal for ventilation and air conditioning service. Available in any size or capacity. Send for Bulletin No. 241-A.

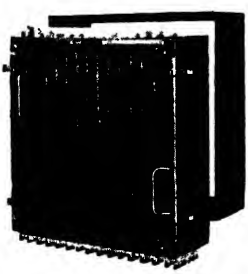
AAF UNIT FILTERS

Throway Air Filter—Throway filters are inexpensive and designed to be discarded after accumulating dust load. Send for Bulletin No. 117-E.

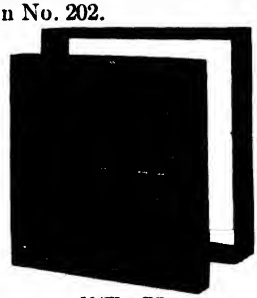
Airmat Type PL-24—Airmat filters use standard Airmat medium, renewable after collecting dust load. Used both for comfort and industrial air conditioning. Available with unit frames to be set up to meet any capacity requirement or space condition. Send for Bulletin No. 230-C.

M/W Filters—The M/W comes in 2 in. and 4 in. thicknesses. Ideally suited to air cleaning problems encountered in general ventilation and commercial air conditioning. Permanent type, washable. Send for Bulletin No. 202.

Throway Air Filter



Airmat Type PL-24 Filter



M/W 2 Filter

W. B. CONNOR ENGINEERING CORP.

114 East 32nd Street,
New York 16, N. Y.

Representatives in
All Principal Cities

In Canada: Douglas Engineering Co., Ltd., Montreal, P. Q.

Air Recovery DOREX **Air Purification**

WHERE TO APPLY AIR RECOVERY

Air Recovery is simply the conversion of foul or stale air to fresh air. It has been used to advantage wherever air is conditioned to enhance comfort, raise production efficiency, extend food preservation or protect product quality. Depending on the source of contamination, Dorex Air Recovery Equipment has been installed to remove odors and other gaseous impurities from intake air, from recirculated air or from exhaust air.

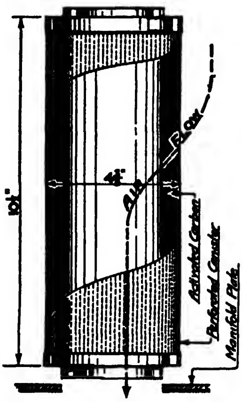


Fig. 1 Dorex Canister

When applied to recirculated air, Dorex Adsorbers reduce the amount of unconditioned outdoor air needed for ventilation and effect savings in installation and operating costs. For example: Given an air conditioning requirement of an area of 20,000 cfm, of which it is assumed 14,000 cfm would be recirculated and 6,000 would be outdoor ventilation before installation of Air Recovery Equipment, it may be possible to cut the amount of unconditioned outdoor air intake to 2,000 cfm by converting 4,000 cfm of used, already conditioned recirculated air to fresh air. Figured for average temperate zones, this 33⅓ per cent load reduction would lower the installation and operating cost substantially because each 1,000 cfm of heated or cooled air that is converted saves: (1) 100,000 Btu of installed heating capacity, (2) 2.6 tons of installed refrigeration, (3) 1,800 kw hours of current per cooling season, (4) 1,500 gallons of fuel oil or 9 tons of coal, and (5) incidental water consumption and maintenance.

In existing systems, the application of Dorex Air Recovery Equipment will enable the system to serve a larger space or satisfy a greater conditioning load without increasing cooling or heating equipment and without consuming more fuel or power.

Activated Carbon Traps Gases and Odors

Activated carbon removes gases and odors by adsorption—a natural phenomenon which takes place when air-borne gases or vapors come in contact with it. An instantaneous condensation occurs and the condensed impurities are held tenaciously until the carbon is forced to give them up in reactivation. For air conditioning purposes, however, the carbon must be especially processed, activated, and impregnated to meet the following specifications: (1) High activity (adsorptive capacity) for a wide range of gases and vapors; (2) High retentivity over an entire range of normal operating conditions; (3) No retentivity for water vapor; (4) Extreme hardness to avoid dusting in handling and in service; (5) High apparent density (in the granu-

ig. 2 Typical Canister Arrangement—side view

lar form) of not less than 0.45; (6) Adaptability to repeated reactivation without appreciable loss in activity or retentivity. In actual use, Dorex activated carbon has removed and retained 95 per cent of all gaseous impurities from the air passed through it and maintained that efficiency from six months to two years, depending on the air contamination.

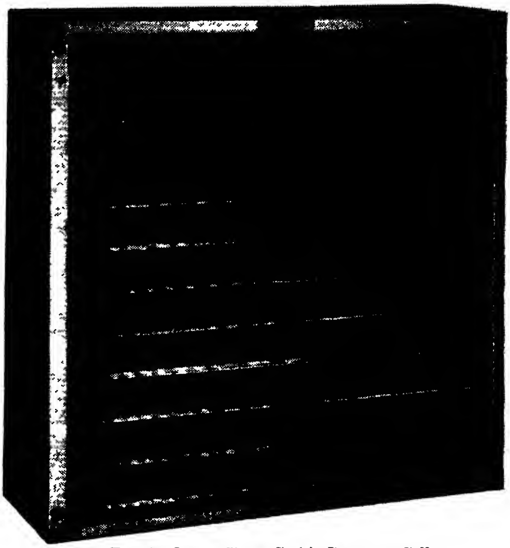


Fig. 3 Dorex Type C Air Recovery Cell

Equipment to Suit Individual Requirements

Dorex Air Recovery Equipment is available in a range of types and sizes to suit individual requirements. Each type is designed to hold the correct amount of activated carbon in a manner to provide a maximum area for decontamination, a minimum of air resistance and uniform air flow through the carbon. The average resistance to air flow ranges only from 0.15 to 0.2 in. wg.

TYPE H - Adaptable to Most Central Systems for Recovering the Freshness of Intake Air and Recirculated Air and for Eliminating Exhaust Nuisances

Type H Equipment—for complete decontamination of all air passed through it - consists of removable, perforated, carbon-filled canisters which are mounted in multiple on one or more supporting manifold plates. Fig. 1 shows a canister and its function; Fig. 2 shows a typical arrangement of canisters as installed. The flexibility of this arrangement makes Type H Equipment readily adaptable to a wide variety of space limitations.

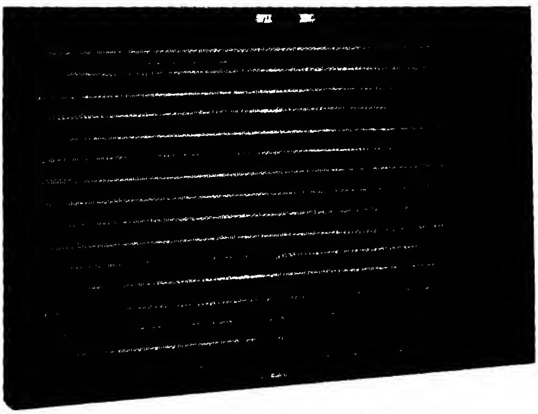


Fig. 4 Dorex G panel

TYPE C—Equipment for Recovering the Freshness of Recirculated Air.

Dorex Type C Air Recovery Cells were developed to meet a need for a large capacity, easily handled and installed air purification unit. Each cell measures only 24 in. x 24 in. x 7 in. deep and completely purifies 1,000 cfm. They require no more engineering than that required for ordinary dust filters and can be mounted right along with them in either flat or "V" arrangement.

TYPE G—for "Package" Conditioners, Unit Heaters, Refrigerated Spaces, Airplane, Bus, Railway Car, and Marine Installations and Other Systems Where Space Is at a Premium.

These compact panels consist of sturdy metal frames, each housing a battery of exposed perforated metal tubes which contain the activated carbon. Standard units of one, two or three tube rows in depth are available in a range of stock sizes for arrangement in air ducts.

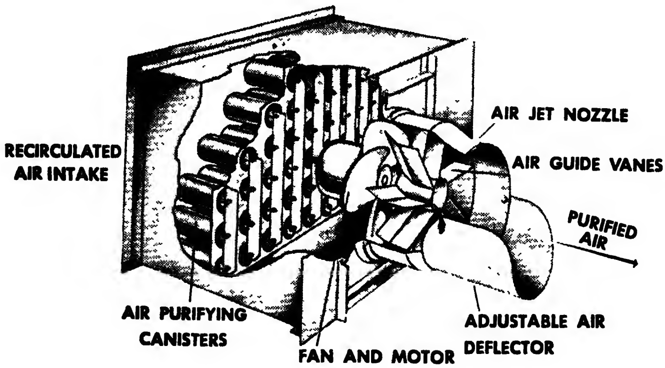


Fig. 5 Dorex Type D Storage Unit

TYPE D—For Extending Storage Life and Preserving Produce Quality in Refrigerated Storage

Type D units are designed to remove ripening gases, disease-causing gases and flavor-impairing odors from the air in cold storages, thus extending storage life and generally preserving produce quality. In apple storages, for instance, they have added 3 to 8 weeks to the keeping time of the fruit. In order to control storage atmosphere adequately and economically, Type D equipment was engineered to the following specifications: (1) Constant purification and recirculation of *all* storeroom air, (2) Thorough mixing of purified air with storage room air, (3) Continuous operation independent of other equipment in the storage space, (4) Flexibility in location, and (5) Self-contained unitary design to eliminate costly duct work or alteration.

Dorex units are portable and can be floor mounted or hung from walls or ceilings. The directional air jet creates an individual pattern for air mixing and distribution and avoids undue air impact on stored produce or fixtures. With the straightening vanes, the amount of "throw" can be adjusted up to a tight jet that reaches 90 feet away from the unit. The large quantity of air thus handled and the high aspiration it creates (five to six times the volume of supply air) results in a very efficient mixing of room and supply air. Dorex Storage Units are built in sizes and capacities to fit any storage space. All parts are either of non-corrosive metal or protected with corrosion-resistant coating.

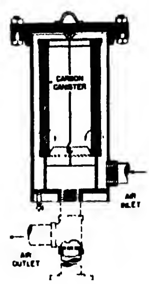


Fig. 6 Dorex Type PL

TYPE PL—For Purifying Compressed Air

The Type PL Dorex Vapor and Gas Adsorber is designed specially to extract oil vapors, fermentation odors and other gaseous impurities from compressed air. It is especially designed to effectively remove air-entrained gaseous odors and impurities not eliminated by commercial filters, separators, after-coolers or receivers.

TYPE A-100-B for Homes, Offices, etc.

Type A-100-B is a self-contained recirculating unit. Attractively designed in enameled wood, it contains a dust filter, four carbon gas adsorbing canisters, circulating fan and motor.

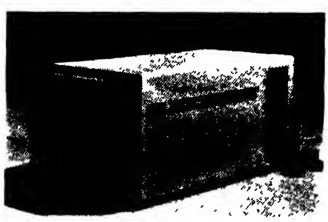


Fig. 7 Type A-100-B

All Dorex equipment is covered by U. S. Patents Nos. 2,214,737; 2,303,331; 2,303,332; 2,303,333; 2,303,334 and others pending; Canadian Patents Nos. 383,986; 429,200; 395,611; 404,855; 410,088; 418,787; 443,235.

Nation-wide Sales and Engineering Service

The W. B. Connor Engineering Corporation maintains a research laboratory, a staff of trained specialists, and district representatives in leading cities. Their services are at the disposal of consulting engineers, architects, air conditioning dealers, and plant engineers. Our staff can assist you in determining whether or not it would be to your advantage to install Dorex in a system you may be designing or improving.

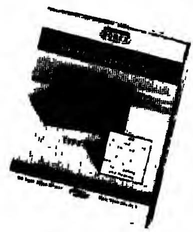
Among Thousands of DOREX Users

American Tel. & Tel. Co.
Anheuser-Busch, Inc.
Boeing Aircraft Co.
Bristol Myers Co.
E. I. du Pont de Nemours & Co., Inc.
General Motors Corp.
Hammermill Paper Co.
Lever Bros.
Monsanto Chemical Co.
Pennsylvania R. R. Co.
Radio Corp. of America
E. R. Squibb & Sons
Union Carbide & Carbon Corp.
Western Electric Mfg. Corp.
Western Union Telegraph Co.

FREE LITERATURE

Shows How to Save Money on Air Conditioning

Bulletin 105A

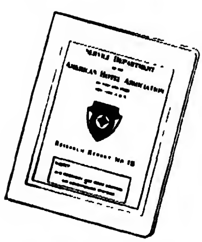


Bulletin 106A

Bulletin 105A on Type H Equipment and **Bulletin 106A** on Type G Equipment are handbooks containing all the detailed drawings, charts and text necessary for the selection and application of Air Recovery Equipment and some typical applications. They also cover pertinent information on ventilation, oxygen requirements and recommended fresh air volumes for offices, stores, apartments, hotels, restaurants, night clubs, theaters, hospitals, and schools.

Bulletin 117

Bulletin 117 contains complete information on Dorex Type C Air Recovery Cells.



York Research Report

Air Recovery and Odor Control in Air Conditioning Systems is an unbiased research report prepared for the American Hotel Association by the York Research Corporation of Connecticut on their investigation into the effect of Air Recovery on air quality and conditioning costs.

New and Complete Textbook for Only $3

Completely revised and brought up to date, the new edition of **Air Conservation Engineering** goes thoroughly into the economics, functions and mechanics of Air Recovery. It can help you figure requirements to a wide variety of uses. Contains technical data needed in designing most applications, including valuable air conditioning tables and charts. Cites actual cases and describes the advantages to Design Engineers, Architects, Plant Engineers and Contractors.

Air Conservation Engineering

For your copies of the FREE literature outlined above or the textbook, Air Conservation Engineering, at $3, please send your request to our Engineering Dept. at 112A East 32nd Street, New York 16, N. Y.

(See pages 1196 and 1197 for data on KNO-DRAFT Adjustable Air Diffusers.)

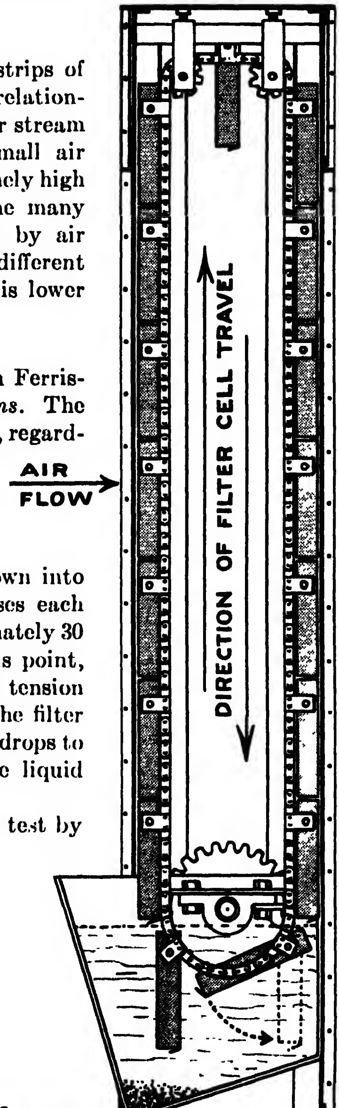
Continental Air Filters, Inc.

2524 Helm Street — Louisville, Kentucky

CONTINENTAL

THE CONTINENTAL AUTOMATIC SELF-CLEANING AIR FILTER

— now being used by many nationally-known companies (*names on request*)—has three entirely unique features: Revolutionary new filter media, Ferris-Wheel action, and positive, effective self cleaning.



THE FILTER MEDIA

The filter media consists of double-corrugated strips of aluminum or other metal, placed in non-nesting relationship to form a honeycomb structure. The main air stream is divided positively into approximately 2300 small air streams per square foot of filter surface. An extremely high efficiency in dust removal is obtained due to the many changes in direction and the turbulence caused by air streams crossing each other while traveling in different directions. The resistance to air flow, however, is lower than in comparable air filters.

FERRIS-WHEEL ACTION

The filter cells, suspended from chains, move in Ferris-Wheel fashion *without changing their relative positions.* The air enters and leaves the same side of each filter cell, regardless of whether it is in the front or back section of the filter curtain. Thus *dirt cannot be blown off the dirty side of a cell, back into the clean-air stream.*

POSITIVE SELF CLEANING

After the individual filter cells have traveled down into the viscous liquid in the tank, a cam action causes each cell to be held in a horizontal position for approximately 30 minutes. The dirty cell is fully submerged at this point, and the 30-minute submergence breaks the surface tension of the liquid which binds the accumulated dust to the filter media. Then, when the cell is released, it quickly drops to a vertical position, and this movement through the liquid washes the dust particles out of the cell.

Cleaning Efficiency: 91.3 per cent, determined in test by *Professor F. B. Rowley, University of Minnesota,* under A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices used in General Ventilation Work.

Resistance:

85% Rated Capacity — 0.17″
95% Rated Capacity — 0.20″
100% Rated Capacity — 0.22″
105% Rated Capacity — 0.23″
115% Rated Capacity — 0.26″

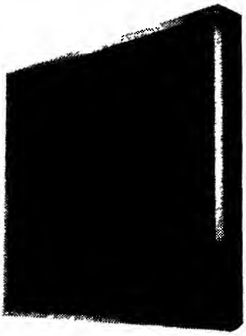
Write for Bulletin No. 201-B, and Suggested Specifications

Farr Company

Manufacturing Engineers

Los Angeles • California

FAR-AIR FILTERS* FOR ALL TYPES OF INDUSTRIAL USE

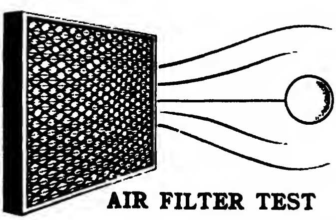


FAR-AIR STANDARD PANEL FILTER

For Ventilation • Grease • Paint

Face velocity in f.p.m.	Capacity in c.f.m. per sq. in. of face area	Starting static pressure	CFM CAPACITIES OF STANDARD 2" #44 FAR-AIR FILTERS			
			16"x20"	16"x25"	20"x20"	20"x25"
346	2.40	.06"	625	795	800	1020
390	2.70	.07"	700	895	900	1145
433	3.00	.09"	780	995	1000	1275
476	3.30	.10"	860	1095	1100	1400
519	3.60	.12"	935	1195	1200	1530
563	3.90	.14"	1015	1290	1300	1655
606	4.20	.16"	1090	1390	1400	1780
650	4.50	.19"	1170	1490	1500	1910
693	4.80	.20"	1250	1590	1600	2035

Based on the recommended net face velocity of 519 fpm, Far-Air Filters deliver 50 per cent more air with the same blower power. This effects a saving in both initial costs and maintenance costs as only ⅔ the filter area need be installed and maintained.



AIR FILTER TEST

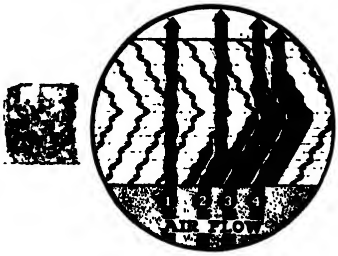
Farr Company has originated a simple procedure for testing air filters in place to determine the comparative efficiency. Write for information regarding this test.

A complete testing laboratory is maintained for all types of research work in the field of air cleaning.

***Trade Mark registered.**

HERRINGBONE-CRIMP DESIGN

Far-Air Filters are permanent, all metal Herringbone-Crimp, zinc electroplated steel wire screen construction. The alternate layers of flat and Herringbone-Crimp fine wire screen results in Low Pressure Drop (0.12 in. W.G. clean at 519 fpm), Greater Air Delivery (1200 cfm at 519 fpm through 20 x 20 x 2 in. unit-permitting equal filter and coil area and eliminating most V-bank installations), Higher Efficiency (improved performance up to velocities as high as 750 fpm), Larger Dirt Holding Capacity (from 30 per cent to over 125 per cent more filtering media per unit), and Easier Cleaning (cold water hosing cleans thoroughly. Far-Air Filters can be cleaned-in-place . . . they do not have to be removed from their holding frames).



PROGRESSIVE LOADING

As entering orifices of Far-Air Filters are loaded, air direction changes, flowing past the front loaded surface and progressively loading the clean screen mesh that remains. Greater free area permits a larger dirt load with lower pressure drop.

For full information about Far-Air Filters write: Farr Company, Dept. HVG, 2615 Southwest Drive, Los Angeles 43, California.

Dollinger Corporation

**Filters for Building Ventilation,
Air Conditioning, Engine Intake, Pipelines and
Many Other Special Applications.**

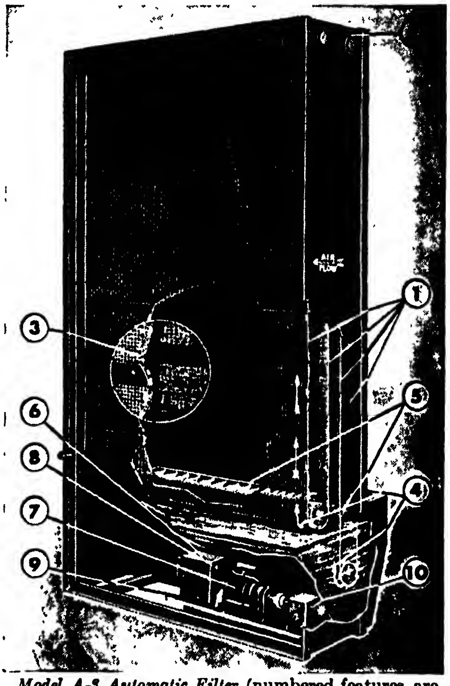
Representatives in Principal Cities

6 Centre Park | Rochester 3, N. Y.

STAYNEW MODEL A-3 AUTOMATIC FILTER

An endless curtain type oil-bath filter for handling large volumes of air at low cost plus exceptionally large dust-handling ability. The efficiency of Staynew Model A-3 is unsurpassed among mechanical self-cleaning filters.

Operation and Features: *Double* filter curtains (1) carried on heavy roller chains driven by sprockets keyed to the shafts of the curtain rollers (2). These rollers float on ball bearings for quiet, frictionless operation. Curtains consist of removable panels (3) made of a single layer of bronze screen cloth to which are attached layers of woven copper mesh.



Model A-3 Automatic Filter (numbered features are referred to in accompanying description)

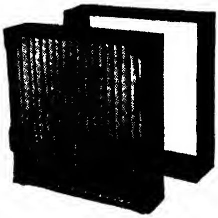
The first of the curtains is the denser, having about twice the impingement surface of the second or rear curtain. This first curtain acts as the filter and travels through the oil reservoir. The second curtain does not enter the reservoir, but acts only as a safeguard against oil entrainment. This design permits a direction of curtain travel such that cleaned panels (4) are always on the filtered air side. Therefore no dust can be carried across the back or return side of the front curtain to be blown off and carried on by the flow of air.

Patented, exclusive Staynew Air Brush Conditioners (5) prevent excessive amounts of oil being carried upward on the curtain panels and being entrained in the air stream.

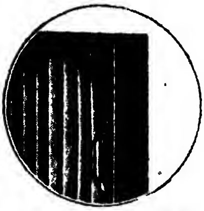
Specifications

Model A-3 Filters are sectional and may be bolted together to obtain any required capacity. Sections come in two widths, 4 ft 3 in. and 2 ft 9 in. Curtain drive and control mechanism (6) arranged either as an integral part of filter unit or for remote mounting, includes a 1/6 hp motor (7) driving through a reduction gear and a momentary contact time switch (8) for testing and checking curtain travel and compressed air control. All are mounted on a common base plate (9) on clean air side of filter. Shear pin (10) is provided for protection of moving parts from accidental damage. The drive motor and the Air Brush Conditioners operate simultaneously for a few seconds at 15 minute intervals.

STAYNEW PANEL TYPE FILTERS

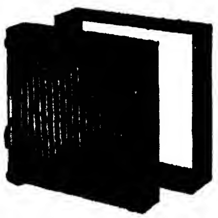

Model WKE Panel and Frame

Model WKE: Dry-type finned panel filter for use in ventilation and air conditioning systems. Extremely large filtering area in relation to overall size. Adaptable to wide variety of filtering media—in fact, almost any medium obtainable in sheet form that can be crimped. Steel mesh on both sides of medium prevents sagging and makes the WKE *fire-resistant* (models available to meet Class I Fire Underwriters approval), and cleanable without possible damage from vacuum cleaning tool or cleaning nozzle. It may also be washed or dry cleaned when and if necessary. There are no cross bars, spacer bars, or other obstructions to interfere with the cleaning operation.


Handles and Latches

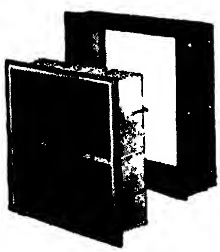
Filter cells are held in rigid box-type supporting frames of heavy gauge metal by spring-loaded cam-type locking latches. Two lifting handles are provided on each cell. Filtering medium supplied already crimped and cut to size. It may be inexpensively replaced in 2 to 5 minutes right at the filter bank—no special tools required.

Frames are drilled so that they can be riveted together to form a flat bank, or by the addition of angle uprights into a "V" or staggered arrangement.


Viscous Model DPV Panel and Frame

Viscous Panel (Model DPV): A permanent type panel for air conditioning systems used in heavy duty industrial service. Filtering media consist of a series of layers of crimped galvanized screen cloth and woven mesh. These media when coated with PD-10 Pingene Filter Oil form an unusually efficient filter. Model DPV filters are cleaned easily with live steam or by washing in a suitable solvent. Spring-loaded locking latches and lifting handles are provided as in Model WKE.

Both Model WKE and DPV cells are furnished in 2 in. and 4 in. depths in various standard sizes.


Standard Panel Insert and Frame

Standard Panel: Dry-type Fin Construction, high filtering efficiency. Heavy steel Panel Insert and Frame. Cleanable. Forty-two square feet of filtering area. Thousands of Staynew Standard Panel Units have given satisfactory service for years. Available in 20 in. x 20 in. x 7 in. size only.

STAYNEW LIQUID FILTER


Model ELS (Sectional View)

Model ELS: Widely used for the filtration of cooling water to prevent clogging of spray nozzles. Exclusive, low-cost SLIP-ON INSERT easy to remove, clean, replace. Radial Fin Construction provides all possible filtering area in smallest possible space. Standard models available to handle up to 1000 gpm.

Representatives in Principal Cities

Complete Information from Factory on Request

FILTERS FOR INTERNAL COMBUSTION ENGINES, COMPRESSORS, PIPE LINES; ALSO DUST COLLECTORS

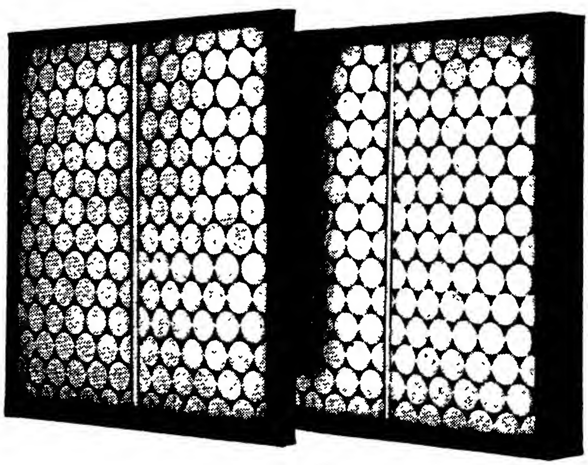
Owens-Corning Fiberglas Corporation

Toledo 1, Ohio

DUST STOP* AIR FILTERS

*T.M. REG. U.S. PAT. OFF.

—a FIBERGLAS* product



Dust-Stop Air Filters are replaceable impingement-type filters for use in all systems in which air is moved mechanically—central heating, ventilating and air-conditioning systems, and mechanically-circulated warm-air furnaces.

Dust-Stops provide high air filtering efficiency. They are constructed of packs of glass fibers (Fiberglas*), coated with an adhesive, faced with a metal grille, and bound on the edges with a fiberboard frame.

The Fiberglas fibers, packed to proper density, form an exceptionally effective medium for air filtration. Being glass, they are inorganic, chemically stable, resistant to heat and corrosive vapors. And being of glass, they do not absorb the nonodorous, nonevaporating adhesive with which they are coated. Each impinged particle of dust is quickly soaked, acting as a wick to carry adhesive to other particles. Thus, the adhesive remains effective until the filter is so heavily loaded with dust that resistance to air flow calls for replacement.

A minimum of manpower and time is required in replacing economical Dust-Stops—and they can be obtained quickly from near-by suppliers. Dust-Stops are made in two standard types: No. 1 (1 in. thick) and No. 2 (2 in. thick). Both are available in several sizes.

*** FIBERGLAS is the trade-mark (Reg. U. S. Pat. Off.) for a variety of products made of or with glass fibers by Owens-Corning Fiberglas Corporation.**

Owens-Corning Fiberglas Corporation Offices

Atlanta, Ga.	Denver, Colo.	Pittsburgh, Pa.
Baltimore, Md.	Detroit, Mich.	Portland, Ore.
Boston, Mass.	Houston, Texas	Richmond, Va.
Buffalo, N. Y.	Kansas City, Mo.	St. Louis, Mo.
Chicago, Ill.	Los Angeles, Calif.	San Francisco, Calif.
Cincinnati, Ohio	Milwaukee, Wis.	Seattle, Wash.
Cleveland, Ohio	Minneapolis, Minn.	Toledo, Ohio
Columbus, Ohio	New York, N. Y.	Tulsa, Okla.
Dallas, Texas	Philadelphia, Pa.	Washington, D. C.

DUST STOP* AIR FILTERS AND FRAMES

*T. M. REG. U.S. PAT. OFF.

TWO TYPES OF FRAMES—

Dust-Stop Air Filters may be installed in banks of either "L" Type or "V" Type Dust-Stop Air Filter Frames. Both types of frames are designed, patented and manufactured by Owens-Corning Fiberglas Corporation for the convenient handling of Dust-Stop Air Filters.

The choice between "L" and "V" Type frames is determined by the frontal area available. The "L" Type frame requires less depth within the duct or plenum chamber but takes a larger face area than the "V" frame for the same cfm capacity. However, it can be set in various arrangements that reduce the required face area by increasing the depth.

The "L" Type frame two filters deep is designed to hold two No. 1 Dust-Stop Air Filters in each cell. The "L" Type four filters deep holds four No. 1 Dust-Stop Air Filters (20 in. x 20 in.). "L" Type frames can be provided in any size from a single unit having one cell at a rated capacity of 800 cfm at a velocity of 300 lineal feet per minute to a unit consisting of 91 cells with a capacity of 72,800 cfm.

The "V" Type frame contains two "cells" each forming one side of the "V." Each cell will hold up to four No. 1 or two No. 2 Dust-Stop Air Filters. "V' Type frames take only the 20 in. x 25 in. filters. The "V" frames are available in any size from a single unit having a capacity of 2000 cfm to a 98 cell unit with a capacity of 98,000 cfm.

All frames are cold rolled steel and are shipped knocked down and crated, complete with all parts and instructions necessary for easy and rapid assembly.

Write for complete information on the application of DUST-STOP Air Filters

"L" Type Frame

"V" Type Frame

Raytheon Manufacturing Company

Waltham 54, Massachusetts

SALES ENGINEERING OFFICES: Atlanta, Boston, Chicago, Cleveland, New Orleans, New York, San Francisco, Seattle, Washington, D. C., and Wilmington, Calif.

RAYTHEON

Excellence in Electronics

RAYTHEON

SERVING THE NATION'S ENGINEERS IN THE FIELD OF

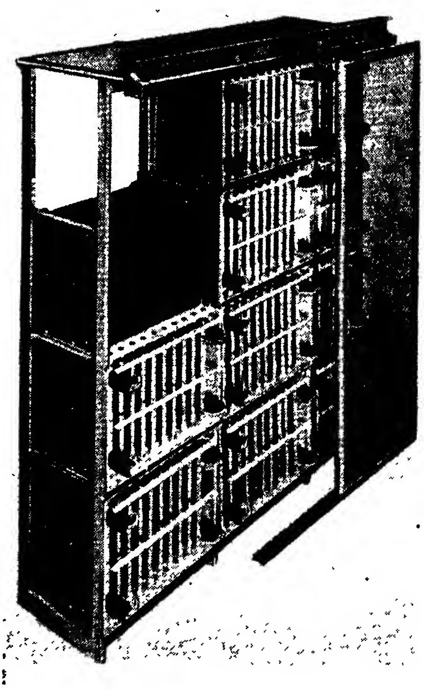
ELECTRONIC AIR CLEANING

Raytheon service in the field of electronic air cleaning means more than the production of efficient equipment for removing dust, soot, smoke, lint, and other air-borne dirt. It begins with the specialized knowledge and assistance offered to architects and engineers in planning the installation . . . extends to contractors who make the actual installation and continues on throughout the years so that the owners may be assured of getting all the benefits that were designed and built into the equipment.

Raytheon Precipitators remove 90 per cent of all air-borne dirt including microscopic particles as fine as 1/250,000-th of an inch, as proved by tests developed by the *National Bureau of Standards*.

CELL-UNIT TYPE Raytheon Industrial Precipitator designed for installation in air circulating, air conditioning, or other air duct systems, can be built up to any required capacity. Washing can be manual (DLP-503), semi-automatic (DLP-521-R), or fully automatic (DLP-549) as desired.

PACKAGE UNITS. Self-contained, for installation with or without duct work for offices, laboratories, commercial establishments, or any other small or light dirt concentration areas. All units contain built-in washing features.



Front view angle of typical 20,000 cfm industrial precipitator showing frame assembly, collector cells in place, ionizer cells, and baffle door. One complete cell and ionizer removed top left and one ionizer removed from cell below to show simplicity of construction and ease of installation. Ionizer slides in from front, and collector cell slides from back, light weight aluminum construction.

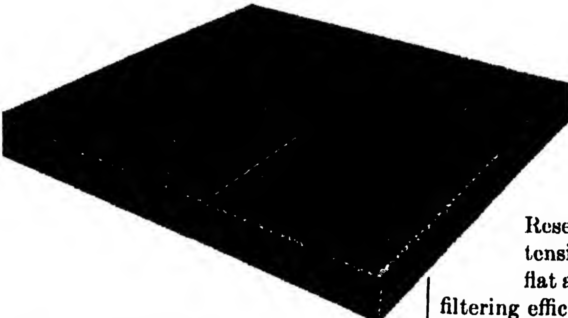
Research Products Corporation

Madison 10, Wisconsin

RESEARCH AIR FILTERS FOR HEATING AND VENTILATING

U. S. Patent 2070073 U. S. Patent 2294478

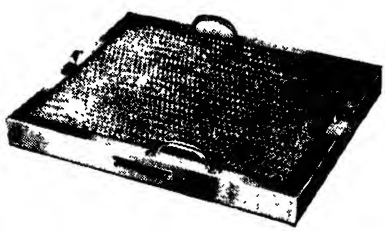
RESEARCH AIR FILTERS



Research Air Filters are widely used wherever air is moved mechanically in blown-air heating and air conditioning systems, in window ventilators and for protection of industrial equipment. Tests prove Research Air Filters have a 93 per cent dust removal efficiency (with 80–20 dust), 99 per cent ragweed pollen removal efficiency. The resistance build-up is very gradual.

Self-Seal Edge The time and labor saving **Self-Seal** edge is an exclusive Research Air Filter feature. The filtering media is a pad constructed of single sheets, cut, expanded and held together to form a honeycomb pattern of thousands of tiny baffles. This unique construction allows filter to be made slightly oversize, fitting snugly into place and eliminating all by-pass of air. When filter becomes dirt clogged, only the filter pad need be replaced.

Research Filter Banks. Research Air Filters are used extensively in filter banks—both flat and V types. For maximum filtering efficiency and minimum service requirements, new speedy **"Snap-in" grids** are used with filter pads, the removal of front grid only being necessary when changing pads.



Washable Alumaloy Filter. This filter has the same efficiency as the disposable type filter. It is about ⅓ lighter than most other washable type filters. Filter is cleaned by agitating in hot soapy water.

Alumaloy "Clean Duct" Grease Filter. All aluminum construction. Light weight. Easily cleaned. Retains high lustre appearance.

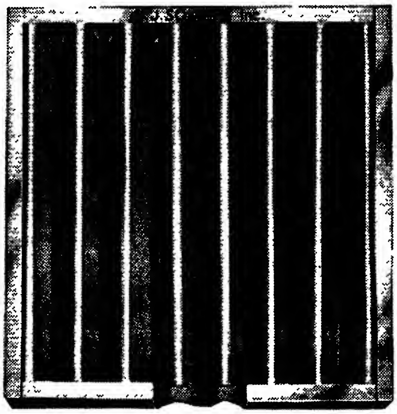
The Research Filter Watchman, a new automatic filter gauge, signals with light when filters are loaded to a maximum restriction to air flow; it may be set for any recommended resistance.

H. J. Somers, Inc.

6063 Wabash Ave., **Detroit 8, Mich.**

Agents in All Principal Cities

SOMERS Heavy Duty Industrial Filter



All Welded Vee Type
Patent No's. 2008800, 2130107

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system.

Consider These Features

- **High rating for dust, soot and bacteria separation.**
- **Require no adhesive, coating or impregnation.**
- **Indestructible in normal service.**
- **Minimum low-pressure drop.**
- **Odorless and non-absorptive.**
- **Fireproof.**
- **Washable.**
- **Permanent—Do not rot nor disintegrate.**
- **All welded zinc-plated 20 ga. steel frame.**
- **Metal protection strip on apex.**
- **Glass cloth between hot-dipped hardware cloth.**
- **Glass ribbon seal so air cannot short circuit.**

Somers Hair Glass Filters consist of a 20 gauge hot galvanized frame holding galvanized wire cloth packed with hair-spun glass strands. The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair Glass being chemically inert, has no facility of absorption; it cannot rust and lasts indefinitely in service. Water either hot or cold may be used to clean it, without impairing its efficiency.

These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

SOMERS WASHABLE AIR FILTERS All Welded Vee Type Stock Sizes

Frame Size Height and Length	Frame Depth	Filter Surface Square Inches	For Average Dry Filter Installations	Wet Application
8″ x 12″	3⅛″	288	288 C.F.M.	144 C.F.M.
12″ x 12″	2⅞″	288	288 C.F.M.	144 C.F.M.
12″ x 20″	3⅛″	720	720 C.F.M.	360 C.F.M.
15½″ x 24½″	3 1/16″	1023	1023 C.F.M.	511 C.F.M.
15½″ x 24½″	3⅛″	1674	1674 C.F.M.	837 C.F.M.
15⅝″ x 24⅝″	2″	480	480 C.F.M.	240 C.F.M.
15⅝″ x 24⅝″	3⅛″	1110	1110 C.F.M.	555 C.F.M.
16″ x 20″	2″	384	384 C.F.M.	192 C.F.M.
16″ x 21½″	3″	816	816 C.F.M.	408 C.F.M.
16″ x 25″	2″	480	480 C.F.M.	240 C.F.M.
16″ x 25″	2½″	624	624 C.F.M.	312 C.F.M.
16″ x 25″	3″	864	864 C.F.M.	432 C.F.M.
16″ x 25″	3⅛″	1344	1344 C.F.M.	672 C.F.M.
16″ x 25″	3⅛″	1440	1440 C.F.M.	720 C.F.M.
16″ x 25″	3⅛″	1632	1632 C.F.M.	816 C.F.M.
16″ x 25″	3⅛″	1056	1056 C.F.M.	528 C.F.M.
18″ x 18″	3 1/16″	864	864 C.F.M.	432 C.F.M.
18″ x 18″	3⅛″	1134	1134 C.F.M.	567 C.F.M.
18″ x 24″	3″	1080	1080 C.F.M.	540 C.F.M.
19½″ x 19½″	2″	480	480 C.F.M.	240 C.F.M.
19½″ x 19½″	3″	819	819 C.F.M.	409 C.F.M.
19½″ x 19½″	3″	936	936 C.F.M.	468 C.F.M.
19½″ x 19½″	3″	995	995 C.F.M.	497 C.F.M.
19½″ x 19½″	3⅛″	1053	1053 C.F.M.	526 C.F.M.
19½″ x 19½″	3⅛″	1170	1170 C.F.M.	585 C.F.M.
19½″ x 19½″	3 3/16″	1696	1696 C.F.M.	848 C.F.M.
20″ x 20″	2″	480	480 C.F.M.	240 C.F.M.
20″ x 20″	2½″	600	600 C.F.M.	300 C.F.M.
20″ x 20″	2¾″	780	780 C.F.M.	390 C.F.M.
20″ x 20″	2¾″	840	840 C.F.M.	420 C.F.M.
20″ x 20″	3″	960	960 C.F.M.	480 C.F.M.
20″ x 20″	3 1/16″	1020	1020 C.F.M.	510 C.F.M.
20″ x 20″	3⅛″	1200	1200 C.F.M.	600 C.F.M.
20″ x 20″	3⅛″	1320	1320 C.F.M.	660 C.F.M.
20″ x 20″	3⅛″	1680	1680 C.F.M.	840 C.F.M.
20″ x 25″	2″	600	600 C.F.M.	300 C.F.M.
20″ x 25″	2⅞″	1020	1020 C.F.M.	510 C.F.M.
20″ x 25″	3⅛″	1560	1560 C.F.M.	780 C.F.M.
20″ x 25″	3 3/16″	1800	1800 C.F.M.	900 C.F.M.
20″ x 30″	3⅛″	1800	1800 C.F.M.	900 C.F.M.
20″ x 30½″	3 3/16″	2400x	2400 C.F.M.	1200 C.F.M.
23″ x 20″	3⅛″	1656	1656 C.F.M.	828 C.F.M.
23½″ x 23½″	3⅛″	1621	1621 C.F.M.	810 C.F.M.
23¾″ x 17¾″	3″	1068	1068 C.F.M.	534 C.F.M.
24″ x 25½″	3⅛″	1872	1872 C.F.M.	936 C.F.M.
25″ x 20″	3⅛″	1800	1800 C.F.M.	900 C.F.M.
26″ x 23½″	2½″	936	936 C.F.M.	468 C.F.M.
26″ x 23½″	2⅝″	936	936 C.F.M.	468 C.F.M.
26″ x 34″	3⅛″	2652	2652 C.F.M.	1326 C.F.M.
28″ x 33½″	2½″	1428	1428 C.F.M.	714 C.F.M.
29″ x 33½″	3⅛″	3045	3045 C.F.M.	1520 C.F.M.
30″ x 15″	3 3/16″	1800	1800 C.F.M.	900 C.F.M.
30″ x 20″	3⅛″	1800	1800 C.F.M.	900 C.F.M.
30″ x 24″	3″	1800	1800 C.F.M.	900 C.F.M.
31″ x 23¼″	3¼″	3162	3162 C.F.M.	1581 C.F.M.

Other sizes also available. Send for complete stock size list.

Frames zinc plated for 100 hour salt water spray test. Refill may be inserted if necessary.

Quotations and further engineering data, including master holding frame drawings will be sent on request.

Just a *few* users of Somers Filters

Chemical Plants

American Viscose Co.
American Zinc & Chemical Co
Davison Chemical Corp.

Automotive

Frederick Sterns Co.
Cadillac Motor Car Co.
Chevrolet Motor Car Co.
Chrysler Corp.
Fisher Body Corp.

Refrigeration and Air-Cond.

Frigidaire Corp.
Norge Div.—Borg Warner
Kelvinator Corp.
York Ice Machine Co.

Ships

Amer. Shipbuilding Co.
U. S. S. Saratoga
U. S. N. Lake City, Fla.
U. S. N. Daytona Beach, Fla.
U. S. N. Vero Beach, Fla.
U. S. N. Jacksonville, Fla.

Utilities and Municipalities

City of Kenosha
Michigan Consolidated Gas Co.
Detroit Edison Co.
New York Edison Co.
Westchester Lighting Co.

Dep't. Stores

S. S. Kresge Co.
S. H. Kress & Co.

Food Processing

Awrey Bakeries
Gilbert Chocolate Co.
Kellogg Co.

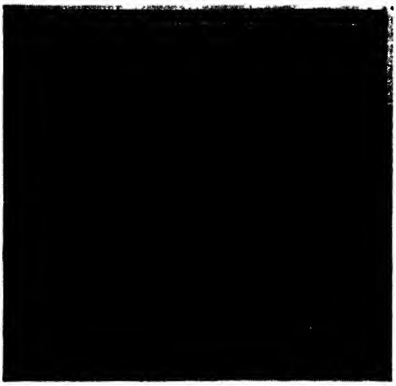
Manufacturers

Buffalo Forge Co.
Burroughs Adding Machine Co.
Clarage Fan Co.
Curtiss-Wright Airplane Co.
Glensder Textile Co.
Hoover Co.
International Heater Co.
Kearney & Trecker Corp.
Killian Mfg. Co.
National Carbon Co., Inc.
Pittsburgh Plate Glass Co.
Rockford Machine Tool Co.
Sunstrand Machine Tool Co.

Supreme Air Filter Company

126 West 21st Street, **New York 11, N. Y.**

SUPREME WASHABLE DRY TYPE AIR FILTERS
AUTOMATIC OR MANUAL CLEANING



Supreme Air Filter (Before and after cleaning)

Supreme Air Filters are made of a spun glass filter media, annealed under a patented process, and covered with a galvanized or copper wire mesh 25-06, encased in a galvanized or copper frame.

Supreme Filters offer a low resistance to the free flow of air, are light in weight and easy to handle. All cells are equipped with handles and interchangeable in the same bank.

Supreme Filters are made in flat and sawtooth types: The flat filter is encased in a ½ in. frame, for convenience in ordering and lower cost production. The sawtooth type is corrugated the depth of the cell frame thereby increasing the filtering surface in a given area.

The spun glass in Supreme Filters is FIRE, ROT and VERMIN proof and not attacked by any chemical other than hydro-fluoric acid; it is approved by the Board of Fire Underwriters.

The important factor in rating a filter is generally not the removal of a certain per cent of dust by weight but rather the effectiveness for taking out certain objectionable constituents, which may be either the finest particles present or the coarser ones within a certain range. The lower number per cent of dust particles in buildings and out-of-doors occur just after air washing by rain or snow storm, and the greatest number in busy streets on dry days.

Supreme Filters can be washed in hot or cold water, adding a little soap powder or solvent, and spraying with not over 25 lb water pressure.

New mats can be supplied for all Supreme cell frames where mats become worn out or defaced, thereby saving the cost of a new frame, which has patent pending.

TABLE OF SUMPREME AIR FILTER SIZES

Sizes	C.F.M.	Mat. SQ"	EFFY	Pd, W.G.	Sizes	C.F.M.	Mat. SQ"	EFFY	Pd, W.G.
1"					3"				
16" x 20"	480	480			16" x 20"	1500	1500		
16" x 25"	480	480	94.5	.14"	16" x 25"	1500	1500	97.1	.18"
20" x 20"	480	480			20" x 20"	1500	1500		
20" x 25"	480	480			20" x 25"	1500	1500		
2"					4"				
16" x 20"	800	800			16" x 20"	2000	2000		
16" x 25"	800	800	95.7	.16"	16" x 25"	2000	2000	98.0	.20"
20" x 20"	800	800			20" x 20"	2000	2000		
20" x 25"	800	800			20" x 25"	2000	2000		

Some Users of Supreme Air Filters

B. Altman Company
Bendix Radio Corporation
Bethlehem Steel Corporation
Christ Hospital
Colgate Palm Olive Peet Company
Commerical Bank & Trust Company
Conde Nast Company
Dime Savings Bank
E. I. DuPont deNeumours & Co., Inc.
Fifth Avenue Hotel
Garment Capital Center
Geigy Dye Works
Koss Restaurant
Masonic Temple
Modern Industrial Bank
New Amsterdam Theatre
New York Central R. R.
New York Telephone Bldg. Corp.
New York Trust
Ohio Power Company
Pabst Air Condg. Corp.
Panama Steamship Line
Posi Print Works
Riegel Paper Corp.

Supreme Air Filters have been approved by the Maritime Commission for use on the C-2 ships.

Gulf Shipbuilding Co.
Moore Dry Dock Co.
Mantowac Shipbuilding Co.
U. S. S. A. So. Africa
International Tel. & Tel.
Colonial Trust Co.
Western Electric Company

Write Supreme Air Filter Company for Complete information and prices

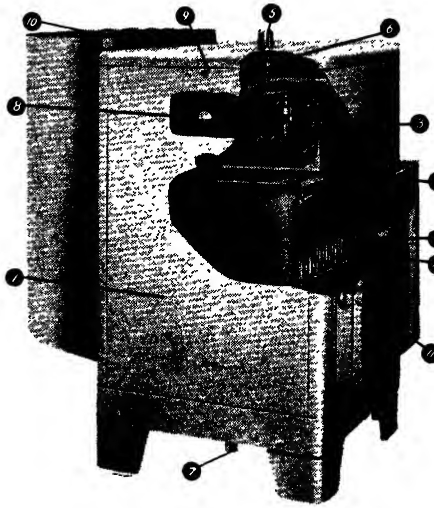
TRION, INC.*

1000 Island Avenue, **McKees Rocks, Pa.**
IN METROPOLITAN PITTSBURGH

TRION
ELECTRIC AIR FILTER
(Electrostatic Precipitator)

Wherever clean, pure air is desirable or mandatory—in residential, commercial or industrial applications—a Trion Electric Air Filter will probably solve the problem. When installed into the return air duct of warm air heating, air conditioning or ventilating systems (in accordance with specifications), Trion effectively removes more than 90 per cent of the dust, dirt, smoke, lint, pollen and other air-borne irritants from air streams passed through the filter—as determined by the Bureau of Standards "blackness" test.

The standard "packaged" unit is available in four sizes for handling air volumes up to 4000 cfm. Suspended cabinets and specially constructed filters for unusual or larger applications are designed to specification.



Illustrated above is a cutaway view of the 1200 cfm "packaged" Trion Electric Air Filter.

Shipped fully assembled . . . easily installed . . . economical to operate . . . simplified controls . . . requires no attention . . . absolutely safe . . . no moving parts to wear . . . built for a lifetime of service . . . component parts are factory serviced.

Only three distinct, functional units, housed in a heavy gage steel cabinet (1) comprise the entire Trion: The Ionizing-Collecting Cell (2), Power Pack (3) and a Water Wash System—under the housing (4). Each slides out like a drawer, to be quickly replaced, if necessary. The Electric (5) and Water (6) Lines enter the top of the cabinet and the dirt-laden water drains from the bottom (7). On the front of the cabinet is the Control Panel (8) with indicating instrument, power and water wash switches—interlocked for complete safety. No access may be had to the electrical parts except by the removal of the front panel, and the mere turn of the Safety Screw (9) de-energizes the unit. Vertical (10) and Horizontal (11) Adaptors allow the Trion to be readily fitted into existing air conditioning and warm air systems. The Horizontal Adaptor contains a Glass Wool Filter (12) on the clean air side to diffuse air and trap any blow off that might occur under unusual conditions. Universal design permits switch for left or right hand installation.

* Designers and manufacturers of equipment for electrostatic cleaning and purifying of air and other gases.

April Showers Company, Inc.

4126 Eighth Street, N.W. **Washington 11, D. C**

April Showers
RIGHT AS RAIN
PATENTED AUTOMATIC ROOF COOLING

(Trade Mark Reg. U. S. Pat. Off.)

AUTOMATIC EVAPORATIVE ROOF COOLING

Distributors and Dealers in Principal Cities

WATER COOLED ROOFS PREVENT SOLAR INFILTRATION

U. S. Government Office Building COOLED with APRIL SHOWERS

SURFACE COOLING has many converts. **THESE** installations show the trend toward space cooling by the sprayed surface method. Hundreds of installations, from Boston to Los Angeles have been made. The systems are automatic, and very little water is required:

Country Life Press Corp., Garden City, L. I	90,000 sq ft of roof
Hallicrafters Co., Chicago	140,400 sq ft of roof
Lily Tulip Cup plant (Westinghouse air conditioned), Augusta, Ga.	104,000 sq ft of roof
Ponemah Mills, Taftville, Conn.	52,015 sq ft of roof
Aerojet Engineering Corp., Azuza, Calif.	22,000 sq ft of roof
Westinghouse Electric Company, Hyde Park, Mass	12,544 sq ft of roof
General Electric Corp., Providence, R. I.	8,538 sq ft of roof
Bulova Watch Co., Providence, R. I.	31,000 sq ft of roof
Magnolia Paper Co., Houston, Tex.	37,576 sq ft of roof
Telechron, Inc., Ashland, Mass. (repeat orders)	54,500 sq ft of roof
Weston Electrical Instrument Co., Newark, N. J.	24,000 sq ft of roof

Write for Descriptive Literature *No Obligation for Estimate*

A FEW DEALERSHIPS STILL OPEN

APRIL SHOWERS is the trade name of an EFFICIENT fool proof method of preventing roof heat penetration or solar infiltration causing excessive heat in upper floors and in factories, stores, theatres, shops, etc.

City water under normal city pressure is usually adequate to serve your APRIL SHOWERS system. The sun operates it. Roofs LAST LONGER, greater comfort is assured, production during the summer's heat is kept apace, errors through fatigue are avoided, and MANAGEMENT acclaims APRIL SHOWERS a GOD-SEND.

Developed in 1933–34.

APRIL SHOWERS controlled roof cooling is protected by U. S. Patents; beware of imitators or infringers. USED BY AIR CONDITIONING ENGINEERS to cut solar load.

APRIL SHOWERS new residence heads now ready.

Foster Wheeler Corporation

165 Broadway, New York 6, N. Y.

District Offices

Atlanta • Boston • Chicago • Cincinnati • Cleveland • Dallas • Detroit
Houston • Kansas City, Mo. • Los Angeles • Philadelphia • Pittsburgh
San Francisco • Washington, D. C.

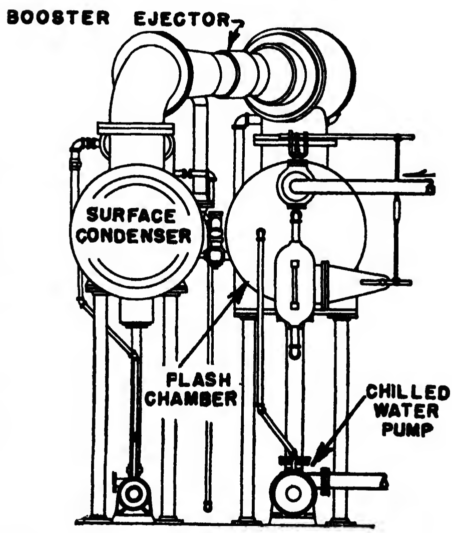
Cooling Towers

Foster Wheeler engineers have had wide experience in the design and construction of high-efficiency cooling towers for service under any climatic conditions. FW towers have been constructed all over the world to meet the cooling requirements of a variety of industrial needs, such as those encountered in chemical plants; public utilities; textile manufacture; office buildings and department stores; and oil refineries. Careful study of each proposed installation is undertaken before recommendations for the most suitable tower are given.

Left: Three of five induced draft towers, introduced by Foster Wheeler and installed in 1933 on the roof of the R. H. Macy department store in New York City. The first two towers cool circulating water for the condenser of a turbine; the other towers, only one of which is shown, cool water used in Macy's air conditioning system.

Vacuum Refrigeration

Schematic diagram showing arrangement of a typical vacuum refrigeration system. These systems, which cool water by subjecting it to high vacuum, supply chilled water for air conditioning or refrigeration. When a sufficient quantity of steam is available, vacuum refrigeration systems have several outstanding advantages—low initial cost; low maintenance cost; no toxic, explosive refrigerants; and, exclusive of pumps, no moving parts. Entire units designed and constructed by Foster Wheeler.



Binks Manufacturing Company

3130–36 Carroll Ave., **Chicago 12, Ill.**

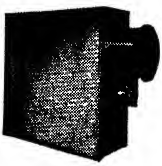
Representatives in all principal cities

Water cooling systems and nozzles . . . a size and type for every purpose

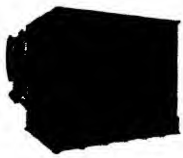
Binks atmospheric spray cooling towers

Small sizes, in a variety of standard units with capacities ranging from 10 to 125 gpm—larger units handle from 600 to 1200 gpm. Special designs furnished in sizes of exceptionally large capacity. Standard tower capacity and temperature performance are based on nozzle pressure of 7 lbs per sq in. Ask for *Bulletin 32.*

Binks horizontal induced draft cooling towers



The horizontal draft principle of operation results in a tower having relatively low height. Single fan units of the spray filled type have fans from 18 to 30 in. in diameter. Larger twin units have fans from 42 to 48 in. in diameter. Frequently installed in multiples for large capacities.
Ask for *Bulletin 34.*



Binks spray filled forced draft towers

Small, compact, quiet, specially suitable for use with packaged air conditioners. Nineteen sizes for systems ranging from 3 to 21 tons of refrigeration.
Ask for *Bulletin 35.*

Binks redwood atmospheric cooling towers

in single section units of solid redwood construction, are built in seven standard sizes to handle normal capacities of 20 to 125 gpm of cooling water.
Ask for *Bulletin 40.*

Binks steel cased induced draft cooling towers

Towers of this type are of the spray filled or deck filled type—made in 20 standard sizes of sq ft rated area. Larger models are engineered to specification. Air propulsion assemblies for either type can be arranged for V-belt or reduction gear drive, as required.
Ask for *Bulletins 36 and 37.*

Binks induced draft masonry cooling towers

Engineered for large scale air conditioning systems in towers that harmonize with the architectural features of the building. Spray or deck filled. Ask for *Bulletin 38.*

Binks spray pond equipment

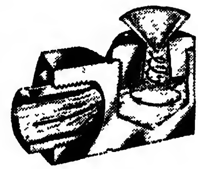
Thousands of Binks-equipped spray ponds provide great operating economy for cooling large quantities of water. Ask for *Bulletin 13.*

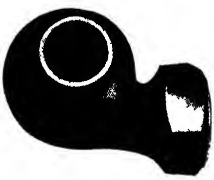
Binks non-clogging Rotojet spray nozzles

Non-clogging Rotojet nozzles are the heart of every Binks water cooling system. They account largely for the efficiency and satisfactory operation of Binks water cooling installations. In addition to cooling tower applications, Binks Rotojet nozzles have found a wide number of uses in brine-spray and quick-freeze refrigerating systems, air washing equipment, metal cleaning and treating machines, chemical plants, etc. Rotojets produce a uniformly fine fluid breakup in a hollow cone pattern. Standard small and medium Rotojet nozzles are machined from brass bar stock, but can be made on special order from monel, stainless steel, or other machinable metals. Large, heavy-duty Rotojet nozzles for use in large cooling towers and spray pond installations, are cast from high quality brass with precision machined threads and orifices. These nozzles may be cast in other metals for special purposes and processes.

Binks small and medium capacity Rotojet nozzles

to fit 1/8 to 3/4 in. pipe connections. Regularly supplied in brass, with male or female threads, as specified. Discharge orifices are available over a considerable range for each size. Rotojet nozzles of this type are designed on the side inlet whirl chamber principle, which produces a fine fluid breakup and a uniform spray pattern.
Full data is contained in *Bulletins Nos. 10 and 11.*





Binks heavy-duty Rotojet nozzles

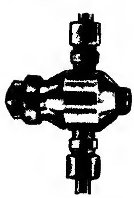
To fit 1 to 2½ in. pipe connections. Female threads only. Discharge orifices available in various sizes, from 9/16 in. to 1-13/16 in. The totally unobstructed involute type of whirl chamber produces a uniformly fine water breakup at low pressures (5 to 7 lb).
Ask for *Bulletin No. 12.*

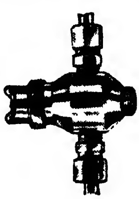
Binks Spra-Rite nozzles

Produce a solid mass cone spray pattern. Small sizes for 1/4 to 3/4 in. connections are widely used for air washing, cooling, brine refrigeration, rapid evaporation processes, filtering systems, chemicals, etc. *Bulletin 19.*

Binks large capacity Spra-Rite nozzles

to fit 1 to 3 in. connections meet a variety of heavy-duty uses in blast furnace gas washers, vibrating and revolving screen coal and gravel washers and water cooling. *Bulletin 19.*





Binks pneumatic atomizing nozzles

Series 50 nozzles are designed for use wherever conditions of controlled humidity must be maintained. as in the storage of perishable products, paper storage and printing plants, textile mills, greenhouses, etc. Nozzles of all brass construction deliver round or flat spray and are designed for use with automatic siphon or pressure feed installations. Described in *Bulletin No. 16.*

Engineering service and technical bulletins

Binks engineering service and facilities are available without obligation or cost to architects, heating and ventilating engineers and builders. We welcome the opportunity to be of service in planning and installing cooling systems that will fully meet every requirement of performance. Give us the details of your problem and we will submit our suggestions.

Technical bulletins describing Binks water cooling systems and industrial nozzles are available for all units described on these pages. Write for the ones that will be useful to you. They will be mailed promptly, without obligation.

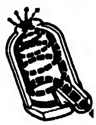
The Marley Company, Inc.

Fairfax and Marley Roads, **Kansas City 15, Kansas**

Representatives in All Principal Cities (Consult Classified Phone Directory)

Water Cooling Towers of All Types and Capacities. Spray Nozzles

Marley Non-Clog Spray Nozzles



Low pressure tangential inlet with high speed whirl and uniform fine spray.

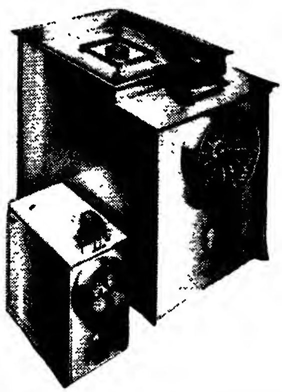
Two-Piece Nozzle for all cooling and spraying. Readily cleaned.

One-Piece Nozzle for cooling towers, spray ponds; large capacity.

Atomizing Nozzle for humidifying vaporizing; a "mist-like" spray.

ATMOSPHERIC SPRAY TOWERS

Marley atmospheric towers are designed for lowest cost, high efficiency installations on roof or ground where breeze is unobstructed. They are easily installed, simple and economical to operate. No fans, motors or other moving parts are employed. Towers are sturdily constructed of heart quality redwood for long, trouble-free life. These towers owe their efficiency to the use of Marley low pressure spray nozzles and distribution systems. They are made in a wide range of closely graduated sizes and are shipped complete with necessary hardware, ready to erect.

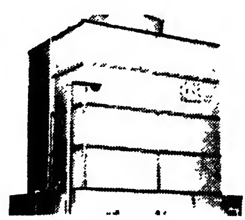


MARLEY AQUATOWERS

Marley Aquatowers are built and shipped as complete units and are remarkable for simplicity of installation and operation. Their small space requirement makes them ideal for many applications either indoors or out. They are steel cased and are filled with nail-less redwood filling that retains correct alignment. Aquatowers are produced in two styles and seven sizes for specified refrigeration loads.

SMALL SERIES VERTICAL STEEL

These attractive induced draft steel cased towers are designed for medium capacity refrigeration and air-conditioning service. Small Series Vertical towers are equipped with Marley Triple Effect eliminators, Marley low pressure spray nozzles and distribution system. They are prefabricated and piece-marked for simplified field assembly. V-belt fan drive is standard equipment for all models. Marley Gearducers are available for larger capacity towers.



MARLEY DOUBLE-FLOW TOWERS

For large capacity installations, Marley patented Double-Flow towers are designed to provide maximum efficiency and flexibliity.

EQUIPMENT DIVISION J. F. Pritchard & Company

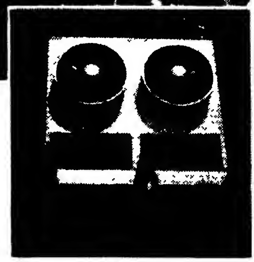
908 Grand Ave., Kansas City 6, Missouri

NEW YORK, CHICAGO, LOS ANGELES, HOUSTON, PITTSBURGH, ST. LOUIS, TULSA
Representatives in Other Principal Cities

THREE LEADING LINES—Cooling Towers, Heat Exchangers, Gas Equipment for FIVE MAJOR FIELDS—Chemical, Natural Gas, Petroleum, Power and Refrigeration

COOLING TOWERS—Capacities ranging from small steel models for air conditioning up to heavy-duty induced draft industrial towers. Patented fans and drives developed for specific duties show distinct advantages. Specifications, prices, rating and application recommendations furnished promptly, without obligation. Shipment from stock.

Medium Sized Industrial Induced Draft Type Tower



Horizontal Airflow Steel Towers for Air Conditioning Installations

Mechanical Draft Towers—Conventional or "coil type" induced draft counter-flow designs achieve greatest cooling efficiency. All sizes available with redwood or steel frames, fireproof sheathing, double-wall casing. Exclusive features include wrought Monel FEATHERWEIGHT FANS*, SEALDFLOW* ventilated fan drive units; low pumping head; trouble-free water distribution.

Natural Draft Towers—Deck towers with DRIFT-RETRIEVER* louver design, minimizing drift loss without restrictive secondary louvers. High operating efficiency, low operating costs, soundly engineered. Spray towers, any capacity for roof or ground installation. Slip-fit louvers. Simple to assemble. Redwood construction. Shipped complete.

Standard "Package Type Tower

Packaged Towers Available in steel (painted or galvanized), aluminum or Monel. Completely assembled or knocked down for field assembly. Ideally suited for refrigeration and air conditioning systems, air compressors, etc.

SEALDFLOW Fan Drive*

Replacement equipment for any make of existing towers includes SEALDFLOW* fan drives, conventional right angle fan drives, all-Monel direct drive self-contained units, and FEATHERWEIGHT* fans.

HEAT EXCHANGERS—All types of heat transfer apparatus, high or low pressure, shell-and-tube, atmospheric, etc., in standard or special design. QUINTAIR* air-cooled heat exchangers adaptable for jacket water cooling, steam condensing and AIRDFINS* for cooling vapors, gases, oils, acids, other fluids. Open, simple, sturdy construction allows for easy maintenance and mechanical reliability.

GAS EQUIPMENT—Includes HYDRYERS*, package type units for air and gas drying, using any one of several solid desiccant materials.

Total or controlled dehydration in any capacity. Standardized for specific applications. 3-ZONE* cleaners and separators, horizontal or vertical types for air, gas and steam; and special mist-extracting applications.

"Laboratory Size" Air Dryer

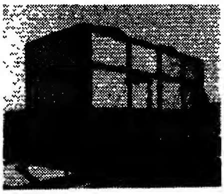
*Registered trade names for specialized equipment produced by J. F. Pritchard & Co.

Water Cooling Equipment Company

8613 New Hampshire Avenue

Affton Station, St. Louis 23, Missouri

MANUFACTURERS OF MECHANICAL DRAFT AND ATMOSPHERIC COOLING TOWERS



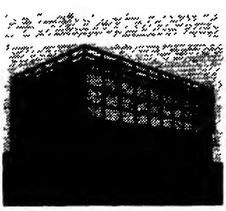
LOW HEAD, INDUCED DRAFT COOLING TOWER

WLH Low Head, Induced Draft Tower with multiple cell arrangement. Monel Metal fan ring, Monel Metal blades and hub are used in the fan Unit. Self-contained, built-in motor located inside fan hub, is used. Motor bearings factory lubricated and sealed requiring no further lubrication for the life of bearings. Outer casing can be cement asbestos board with contrasting trim strips for fire resistance and improved appearance.

REDWOOD "WATERFALL" ATMOSPHERIC SPRAY COOLING TOWER

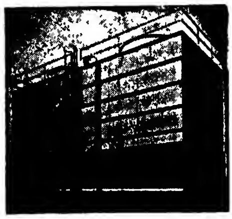
Factory fabricated and shipped knocked - down with all hardware, spray headers and spray nozzles. Complete erection instructions and drawings are furnished to assist in the assembly of the tower.

These towers are portable and can be knocked-down and moved simply by removing the louvres and bolts. Shipment can be made from stock. Write for Bulletin 125-A.



INDUCED DRAFT COOLING TOWER

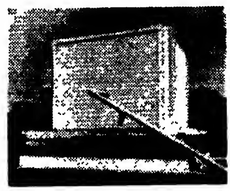
A 7700 gpm Induced Draft Cooling Tower constructed of all heart Redwood which was chemically treated with flame retardant and outer casing furnished of cement asbestos to conform to fire prevention regulations. Equipped with four 168-in. diam. stainless steel, 6-blade, adjustable pitch, propeller type fans.



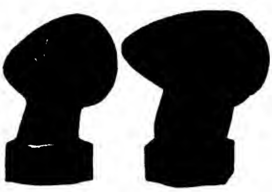
INDUCED DRAFT

This type of cooling tower is recommended for installations where noise is a prime factor of consideration. The noise of the mechanical items is carried upward and discharged into the air. The mechanical draft cooling tower assures a positive cooling of the water to a specified temperature independent of wind velocity.

LOW HEAD, INDUCED DRAFT COOLING TOWER



Redwood WLH Low Head Induced Draft Cooling Tower designed for roof or ground installation. Nail-less grid filling of advanced design and all - bolted construction, is employed. Water is distributed by the gravity-trough system resulting in a very low pumping head. Write for Bulletin WLH-801.



Patent No. 2,123,697

SPRAY NOZZLES

"Whirlcone" non - clogging, low pressure, centrifugal type spray nozzles for spray ponds, spray towers and other uses. Write for Bulletin 76-A.

ENGINEERING

Design and construction are based on sound engineering principles to meet specific requirements for cooling performance and structural strength. Redwood, steel or other suitable materials are used.

American Moistening Company

ESTABLISHED 1888

Providence 1, R. I.

ATLANTA 2, GA. BOSTON 9, MASS. CHARLOTTE 1, N. C.

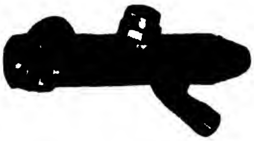
AMCO

UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

A few of many AMCO products with a Long Record of Dependable Performance

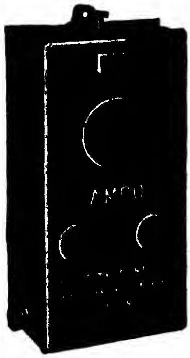
Self-cleaning Atomizers.
Sectional Humidifiers.
Ideal Humidifiers.
Amtex Humidifiers.
Hand Sprayers.
Fabric and Paper Dampeners.
Electro Psychrometers.
Sling Psychrometers.
Hygrometers.
Mine Sprays.

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification. Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance.



AMCO ATOMIZER—No. 5

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning. When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned.



AMCO HUMIDITY CONTROLS

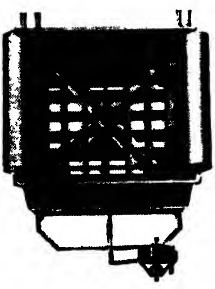
Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges.

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hygroscopic element operates electrical contacts which control the units.



AMCO EVAPORATIVE COOLING UNIT

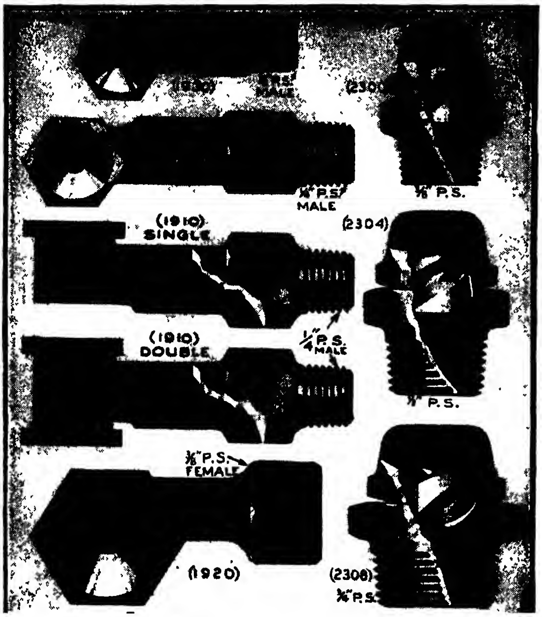
The Amco System of evaporative cooling contributes to smooth production at high speeds in two ways; it maintains the percentage of relative humidity best suited to the fibre and process involved, and at the same time promotes the comfort and efficiency of personnel by obtaining the maximum practical cooling effect from evaporation. It does this by introducing outside air into the room in varying amounts, regulated in accordance with climatic conditions and inside requirements.

A ductless system—very flexible and portable. Can be applied in conjunction with an existing humidifying system.

Jos. A. Martocello & Company

229–31 North 13th Street, **Philadelphia, Pa.**

SPRAY POND AND ATOMIZING SPRAY NOZZLES



MARTOCELLO

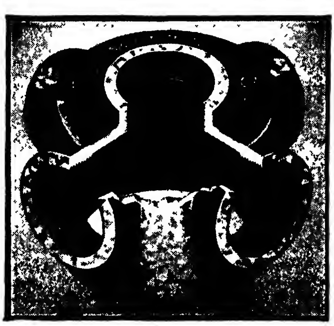
Atomizing Spray Nozzles produce a uniform, good, wide spray with less friction and at minimum pressure requirements.

Nozzles illustrated at the right are manufactured with precision in Brass Forgings and Bar Stock. Their design has been thoroughly tested for results and durability and will give you satisfaction.

Successful, Efficient Results depend largely upon selecting the proper number and type of Nozzles with Brass or Monel cap suitable for your job. Therefore we suggest you send us your specifications as we also have several Types other than illustrated and we will gladly assist you to obtain the most efficient application.

MARTOCELLO

Spray Pond Nozzles of a sturdy one-piece construction—cast of High Grade Red Brass with Inlet and Outlet accurately machined, are less clogging, offer less friction and give unsurpassed overall efficiency.



MARTOCELLO CLUSTER CASTINGS

Sturdy Grey Iron Construction with large area for reduced friction and even distribution. They are Hot Dipped Galvanized after fabrication.

Furnished with **Nozzles** and continuous Standard Steel **Long Sweep** Galvanized Pipe **Spray Arms** and **Center Nozzle** Nipple in accordance with layout required.

Sizes Carried in stock for **Prompt Shipment** 1½ in. P. S. Outlet, 3 in. P. S. Inlet and 2 in. P. S. Outlets, 4 in. P. S. Inlet Cluster Castings.

WRITE, PHONE OR WIRE

For Bulletin listing Capacities and Prices.
Prompt shipments from stock.

Monarch Manufacturing Works, Inc.

2509 E. Ontario St., **Philadelphia 34, Pa.**

SPRAY NOZZLES FOR WATER AND OIL

NON-CLOG AIR WASHER NOZZLES

produce an exceptionally efficient, evenly distributed hollow cone spray. Single large tangential inlet to swirling chamber minimizes any possibility of clogging. Also available in ⅜ in. to 1 in. pipe sizes inclusive, and of Brass, Stainless and Monel.

Fig. 631

Fig. 629

Capacities: Gallons per Hour

Sizes			Lbs. Operating Pressure					
Pipe	Orifice	Lead	10	20	30	40	60	100
1/4"	69	69			2.9	3.3	4.0	5.1
	61	61		4.0	5.0	5.5	7.0	8.7
	61	53	4.4	6.2	7.5	8.3	10.3	13.0
	53	53	5.8	8.2	9.8	10.9	14.1	17.3
	49	49	9.1	11.1	13.2	15.0	19.5	24.6
	3/32"	3/32"	11.8	16.6	20.4	23.4	29.0	36.5
	1/8"	3/32"	19.5	27.6	33.3	39.1	49.0	60.0
	1/8"	1/8"	24.3	34.6	42.8	50.0	64.0	78.5
	11/64"	1/8"	31.5	46.1	57.0	64.1	81.9	105
	3/16"	5/32"	52.2	75.0	92.5	109	138	189
	7/32"	3/16"	78.0	112	138	163	205	257
	7/32"	7/32"	82.0	121	152	180	225	300

AIR CONDITIONING AND OIL BURNER NOZZLES

Water Capacity in Gallons per Hour

Fig. F-80

Nozzle No.	Lb Operating Pressure				
	25	40	60	80	100
1.35		.57	.69	.83	.92
1.65		.75	.89	.99	1.12
2.00		.94	1.14	1.28	1.40
2.50		1.13	1.45	1.64	1.86
3.00	1.03	1.39	1.62	1.85	1.95
3.50	1.36	1.77	2.11	2.46	2.80
4.00	1.56	2.00	2.42	2.77	3.16
4.50	1.86	2.32	2.77	3.21	3.68
5.00	2.20	2.88	3.57	4.09	4.59
5.50	2.22	2.96	3.75	4.31	4.78
6.00	2.55	3.35	4.01	4.78	5.23
7.00	2.90	3.91	4.60	5.17	6.00

Produce finest breakup possible with direct pressure only. Capacities above are on water. "Nozzle No." is capacity on 34 second Saybolt viscosity oil at 100 lb pressure. Larger sizes up to Nozzle No. 60.00.

Furnished of all Brass for Water—Stainless Steel tip and disc for Oil. Standard with ⅛ in. or ¼ in. female pipe Brass adapter and Monel gauze strainer.

SPRAY POND NOZZLES

For recooling condenser water, etc. Operate on pressures from 5 lb upward. Made of Cast Red Brass and in pipe sizes 1 in., 1½ in., 2 in., and 2½ in. Capacities from 4.1 to 88 gpm at 7 lb pressure.

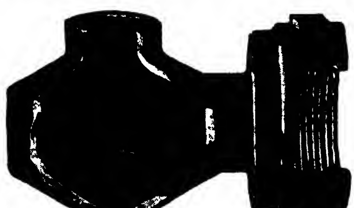


Fig. B-8

Write for Detailed Catalogs

Condenser Service & Engineering Co., Inc.

65 River Street
HOBOKEN, N. J.

ELIZABETH, N. J.
POTTSVILLE, PA. SCRANTON, PA.
EDGEMOOR, DEL.
BRUNSWICK, GA.

DESIGNERS AND MANUFACTURERS OF

Steam Generators—Steam Condensers—Steam Jet Air Ejectors—Lubricating Oil Coolers—Feedwater Heaters—Fuel Oil Heaters—Evaporators—Distillers—Oil Refinery Heat Exchangers—Process Industry Heat Exchangers—Strainers—Feedwater Filters—Grease Extractors—Sewage Ejectors—Pumps—Oil and Water Separators—Wizard Condenser Injectors—Flowrites—Salinometer Cocks—American Ball Steam Engines—Direct Warm Air Heaters.

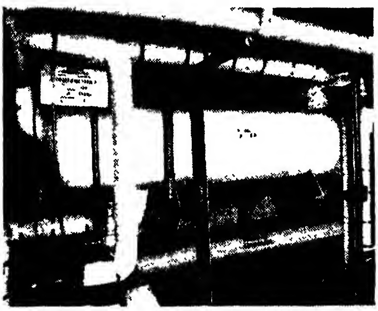
Twenty years of specialized experience *designing*, building, fabricating and a special feature—a designing organization which also services heat exchangers—assures sound design, quality construction and economical operation. Conseco builds heat exchangers for all types of air conditioning systems using, Freon, CO_2, ammonia, methyl chloride and SO_2.

Maintenance Tubing Service—Conseco maintains a highly-trained service organization equipped with special tools developed by us to save time, labor and material, ready for action any time of day or night, anywhere on the continent. Heat exchangers are retubed or repaired in the shortest possible time, and at low cost. Tube sheets, tubes, ferrules, all types of packing, and other necessary materials are always stocked, ready for instant shipment.

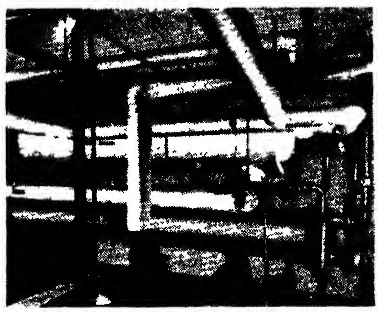
The keynote of our service organization is fast, accurate, dependable service performed by competent, properly equipped men working under qualified engineering supervision.

Every detail of a Conseco heat exchanger is designed to profit from good and bad features uncovered by years of service work on units of every type and make. This has resulted in the production of simple, efficient units, low in original cost and inexpensive to maintain.

Conseco engineers will be glad to serve you. If you want fast action, write, phone or wire.

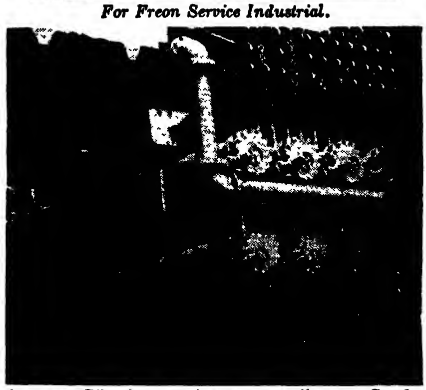


For Freon Service Aboard Ship.



For Freon Service Industrial.

Conseco heat exchanger installation for ammonia service aboard ship.



Conseco CO_2 heat exchanger installation, Stanley Theatre, Philadelphia, replaced double tube unit shown above it.

The Patterson Kelley Company, Inc.

124 Warren Street East Stroudsburg, Pa.

Offices or Representatives in Other Principal Cities

New York Office, 101 Park Avenue, Zone 17

PATTERSON PRODUCTS
for the
Heating Field

- Hot Water Storage Heaters
- Instantaneous Heaters
- Combination Heaters
- Heat Reclaimers
- Heat Exchangers
- Condensers
- Convertors
- Fuel Oil Heaters

PATTERSON PRODUCTS
for
Air Conditioning
and
Refrigeration

- Freon Coolers
- Water, Brine and other Coolers
- Condensers
- Slug Eliminators
- Balance Loaders
- Defrosting Systems

All of the above units are in the class of "heat transfer" equipment and it is in this class that The Patterson-Kelley Company has been specializing since 1880. Its products are carefully engineered and constructed of properly selected materials according to the most modern manufacturing methods. Literature describing any product will be sent on request.

If you have *any* requirement involving *any* type of heat transfer equipment, let us know. Our engineers will be glad to study your problem and recommend the proper type and size of unit.

Patterson Freon Cooler—Dry Expansion Type

Note the freon chamber machined from solid billet of rolled carbon steel with separating partitions welded in place. There are no exposed welds. New Bulletin on request.

AEROFIN CORPORATION

410 So. Geddes Street •

Syracuse 1, N. Y.

AEROFIN

Standardized Light-weight Heat Exchange Surface

Branch Offices

CLEVELAND, CHICAGO, NEW YORK, PHILADELPHIA, DETROIT, DALLAS, TORONTO

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the present and future requirements of this highly specialized field. All Standard AEROFIN Units are furnished as completely encased Units, ready for pipe and duct connections. The patented casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service. The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection.

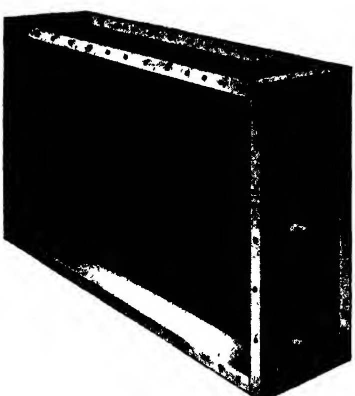


Fig. 1

Aerofin Non-freeze heater (Fig. 1) is non-freeze, non-stratifying spiral fin coil built into casing for air conditioning units or for installing in ducts. May be installed horizontally or vertically. Used on any two-pipe steam system for preheating or reheating. Modulating control on preheaters.

Available in 13 lengths and 3 widths, from net face area of 2.76 sq ft to 26.28 sq ft.

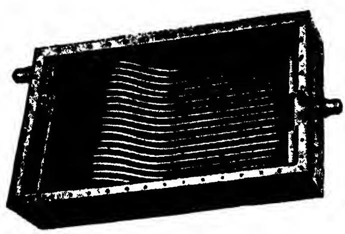


Fig. 2

Flexitube Aerofin (Fig. 2) is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains.

Headers—Steel.

Tubing—⅝ in. O.D. copper, admiralty or aluminum.

Joints—Where admiralty or copper tubes are used together with bronze or steel headers tubes are brazed to headers. Where both aluminum tubes and headers are used tubing is welded to headers.

Casings—Copper, aluminum or galvanized iron.

Design—Constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical.

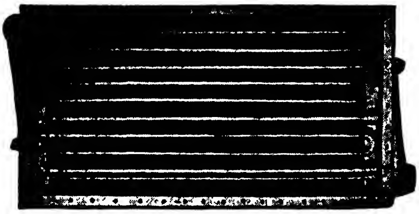


Fig. 3

Universal Aerofin (Fig. 3) is distinguished by its "S" bend construction of tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing.

Recommended where close control is desired.

Headers—Pressed steel.

Tubing—1 in. O.D. copper or admiralty.

Casings—Copper, aluminum or galvanized iron.

Fig.

Booster Aerofin—straight tube type, single pass construction for pressures from 1 to 200 lb gauge.

Headers—steel.

Tubings—⅝ in. O.D. copper.

Casings—copper, aluminum or galvanized iron. Recommended where small coils are needed or to raise the air temperatures in branch ducts.

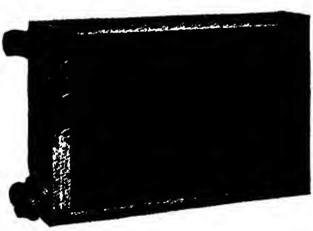


Fig. 4

Aerofin Heavy-Duty Industrial Heating Coil for use where extra-rugged cell is needed for close control. Steam pressures from 25 to 450 lb gauge; temperatures to 550 F.

Headers—Pressed steel.

Tubing—1 in. O.D. heavy copper.

Casings—12-gauge galvanized iron.

Fig. 6

Narrow Width Aerofin: (Fig. 6) recommended for water cooling or for flooded Freon systems. Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed. Construction similar to Flexitube Aerofin.

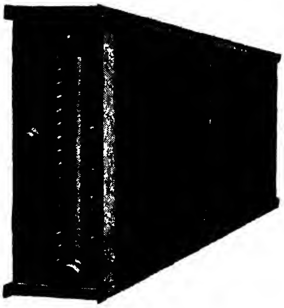


Fig. 7

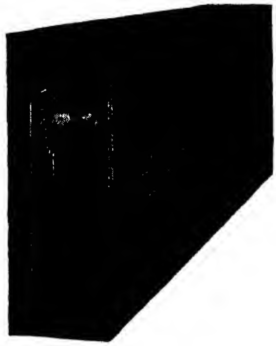


Fig. 9

Aerofin Continuous Tube Water Coils (Fig. 7) are designed for air cooling by circulating cold water through the AEROFIN and air over extended fin surface. Made for either horizontal or vertical air flow.

Tubes and fins are copper, completely tinned with permanent metallic bond between fin and tubes. Headers are made of steel and casings of heavy galvanized iron or copper.

Tested to 100 lb steam, followed by 450 lb air with coil submerged in water.

Fig. 8

Aerofin Cleanable Tube Units (Fig. 8) for cooling only made with headers removable to permit cleaning tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water.

Headers—Cast iron.

Tubing—Copper or admiralty.

Casings—Copper or galvanized iron.

Aerofin Direct Expansion Units: (Fig. 9) Centrifugal Header Type—For cooling air, using Freon expanded directly into the coil.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

Continuous Tube: 13 standard lengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Centrifugal Header —11 standard lengths, three widths, 2-3-4-5-6 rows deep.

Steel Supporting Legs: 18 in. and 24 in. high. Punched same bolt hole centers as standard casings. Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

Write Syracuse for Heating Bulletin G-32; Direct Expansion Bulletin DE-34 on refrigeration type units; Continuous Tube Bulletin C. T. 34 for Water Cooling Coils; or phamplet on Cleanable Type AEROFIN for cooling.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

SQUARE FIN TUBING

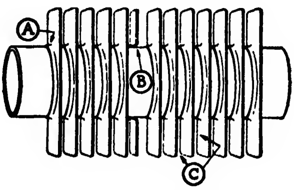
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes.



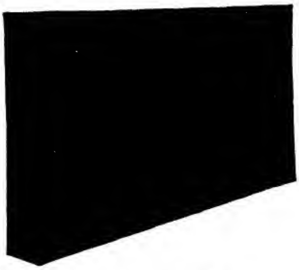
A—Generous Fin Collar provides large contact area between Tube and Fin.
B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—complete thermal contact.
C—Free air-flow passages; non-clogging.

STANDARD SIZES

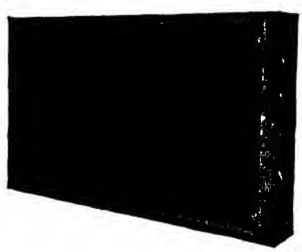
O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8″	7/8″ sq.	6	0.80 sq. ft.
3/8″	7/8″ r'd.	6	0.60 sq. ft.
5/8″	1 1/8″ r'd.	6	0.87 sq. ft.
3/4″	1 1/2″ r'd.	6	1.55 sq. ft.
3/4″	1 5/8″ sq.	6	2.40 sq. ft.
1″	2 1/8″ sq.	6	4.00 sq. ft.
1 3/8″	2 3/8″ r'd.	4	2.33 sq. ft.

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES

G&O Finned Radiation Coils for industrial applications are available in a wide range of sizes.



Universal U-102



Standard No. 10

Send for Catalog and Price List

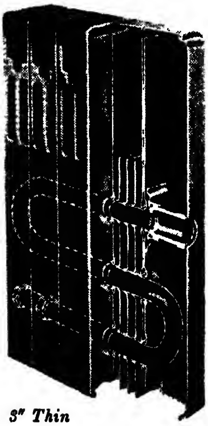
Shaw-Perkins Manufacturing Company

Use SP RADIATORS

Pittsburgh 19, Pa.

CONVECTOR-RADIATORS

Developed far beyond ordinary types of radiators and convectors, the Shaw-Perkins units have brought to radiation pleasing appearance and greater operating economy. Users have been particularly impressed by their highly efficient heat distribution. Installed in homes, ships, hospitals, office and industrial buildings. Because the heating medium in Shaw and Perkins convector-radiators is confined in heavy copper tubing and rugged non-ferrous compression fittings, these units are entirely free from the corrosion that attacks less durable metals.



3" Thin

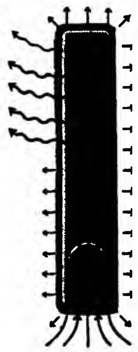
The Shaw Construction

The heat from the steam or hot water contained in the copper tube is conducted through all the steel radiating fins over which large quantities of low temperature air pass constantly and joins with the radiant heat from the exposed metal surfaces to produce a quick and economical form of comfortable warmth.

The Shaw Convector-Radiator

The Shaw Convector-Radiator is an attractive self-contained unit requiring no cabinet or expensive recessing. Occupies little floor space and harmonizes with modern decoration.

Designed to operate on high or low pressure steam or forced hot water. Frequently the high pressure steam feature alone justifies its selection for with such an installation smaller heating units and piping are required resulting in considerable installation and operating savings.



Wide air spaces between smooth steel fins induce the movement of large volumes of low temperature air thereby assisting in the fast and even distribution of heat through the room.

Because of its high heat output, it combines sturdy construction with a weight less than half that of ordinary exposed radiators of the same capacity.

The Perkins Convector-Radiator

A Modern Industrial Heating Unit for High or Low Pressure Steam

This Advanced Radiator embodies a heavy copper tube arranged in the form of a continuous coil to which are mechanically bonded a series of steel plates, the whole assembly being locked into a rigid unit by a combination of tie rods and spacers.

The Perkins convector-radiator has been designed to move large volumes of low temperature air in order to produce quick, uniform, economical heating. The amount of air is greater than that of other types of radiators.

It is designed for operation on either high or low pressure steam. At high pressure the heat emission is greatly increased, so there is a reduction in the number of heating units and auxiliary equipment. Furthermore by modulating the pressure of the steam, the heat output of these radiators may be made to balance the heat loss of the building under varying conditions, resulting in a remarkable steam economy.

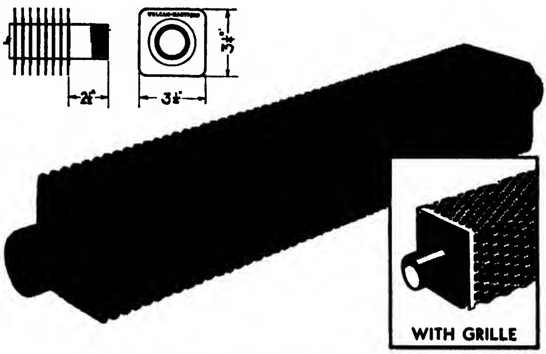
Representatives in principal cities. Send for catalog.

The Vulcan Radiator Company

26 Francis Avenue • Hartford 6, Conn.

MANUFACTURERS OF FINNED TUBES FOR OVER TWO DECADES

Vulcan Radiation is used in railroad cars, ships, hospitals, schools, churches, homes and industrial plants. Available in steel or copper . . . easy to install . . . light in weight . . . requires few fittings and supports . . . tube ends threaded or chamfered for welding. Heat distribution is uniform. Steel radiation comes in two sizes . . . 2 in. IPS, rated—$5\frac{1}{4}$ sq ft per lineal ft at 1 lb steam and 70 deg air . . . for $1\frac{1}{4}$ in. IPS see illustration—this size also available in copper.

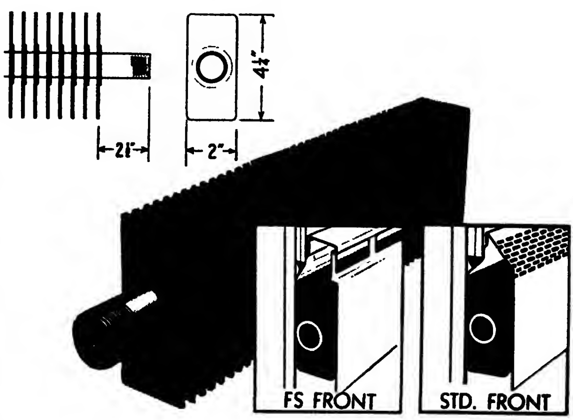


1¼ in. IPS, rated—4¼ sq ft per lineal ft.

Vulcan Radiation is fabricated by mechanically imbedding offset fins or plates on seamless steel pressure tube or copper water tube. The patented offset fin construction gives complete rigidity to the entire assembly and extends the heating surface of the tube.

Because of its comparatively light weight and compactness, Vulcan Radiation responds quickly to thermostatic control. Full heat output is obtained almost immediately after steam is supplied. Since most of the heat is given off by convection, the result is EVEN, UNIFORM HEAT from floor to ceiling.

Vulcan Baseboard Radiation . . . fin-on-tube construction with grille covers combines radiant and convection heating. High safe working pressure . . . either hot water or two-pipe steam systems. Light in weight . . . easy to install . . . requires few fittings or supports. Comes in two sizes . . . $1\frac{1}{4}$ in. IPS . . . steel fins $2\frac{3}{4}$ in. wide by $3\frac{3}{4}$ in. high. Rated 4.0 sq ft per lineal ft with St'd. Front. 3.7 sq ft per lineal foot with FS front. For 1 in. IPS see illustration. Also available in copper.



1 in. IPS . . . steel fins 2 in. wide, 4¼ in. high, Rated—2.6 sq ft per lineal ft at 1 lb steam and 70 deg air with Std. Front. 2.4 sq ft per lineal ft with FS front.

Contact Representatives in Principal Cities or Send for Catalog

Baker Refrigeration Corporation

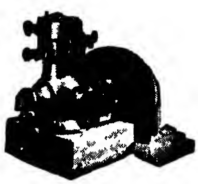
SALES AND SERVICE IN PRINCIPAL CITIES

South Windham, Maine
Omaha, Nebraska
Cable Address: BAKERICE

BUILDERS OF DEPENDABLE REFRIGERATION EQUIPMENT SINCE 1905

●PRODUCTS: Ammonia Compressors; Self-Contained Ammonia Units; Self-Contained "Freon-12" Units; Shell and Tube Condensers, Vertical or Horizontal; Refrigeration Valves and Fittings; Water or Brine Coolers, Shell and Tube or Multi-Unit Type; Coils and Cooler Units of all sizes; Automatic Refrigeration Controls and Accessories; Evaporative Type Condensers.

BAKER AMMONIA COMPRESSORS

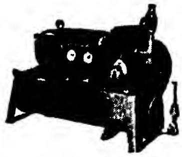


From 1 to 100 tons capacity. V-belt drive or direct connection to motors or engines. Vertical enclosed single-acting type. Duplex or multiple installations to obtain any desired capacity.

BAKER AMMONIA COMPRESSOR UNIT

Model F6B from 7½ to 20 hp. Bore and stroke 3½ in. x 3½ in. Four cylinders, reciprocating single acting type. Force feed lubrication. Timken Roller Bearings. This model available in single or multiple compressor or condensing units as desired.

BAKER "FREON-12" UNITS

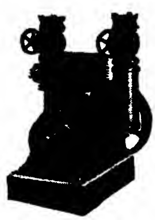


"Freon-12" Units of the self-contained automatic type from ¼ to 20 hp capacity in single units. Furnished in air-cooled type ½ to 20 hp and water-cooled type 3 to 20 hp.

"FREON-12" COMPRESSOR UNITS

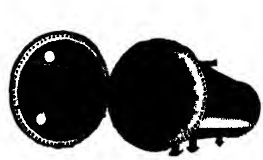
Arranged for use with evaporative type condenser or separately mounted shell and tube condensers. Sizes from 3 hp to 20 hp. Two and four cylinder types. Automatic controls.

BAKER AMMONIA BOOSTER COMPRESSORS



For sub-zero temperature work. Lubrication under pressure. Single or multiple compressor installations. V-belt or direct drive. Large gas manifold at compressor suction ports allows complete filling of cylinder at variable speeds.

BAKER SHELL AND TUBE CONDENSERS AND COOLERS



ed single units to 150 tons, multiple units any capacity. Shells 8 in. or 50 in. diameter. Removable heads. Vertical or horizontal, singlepass or multipass. Brine or water.

Baker Also Manufactures a Complete line of Industrial-Type Cooling Units, Ammonia Valves, Screw-End Fittings, Capped Valves, Flanged-Type Fittings

Curtis Refrigerating Machine Division

of Curtis Manufacturing Company

1959 Kienlen Ave., **St. Louis 20, Mo., U. S. A.**

ESTABLISHED 1854

Full Line of Units from ¼ to 30-hp

Unit Coolers and Evaporator Coils

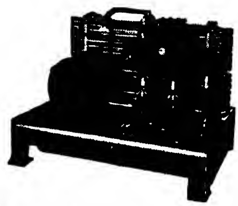
PRODUCTS: Complete Refrigerating Equipment for Dairies, Creameries, Ice Cream Cabinets, Ice Cream Making Plants, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling, Processing and Air Conditioning Installation, Packed and Remote Types.

¼ to ½ hp Self-Contained Condensing Unit.

Commercial Refrigeration

Air cooled condensing units from ¼ to 3 hp, inclusive, and water cooled units from ⅓ to 30 hp, inclusive. All models available for either Freon (F-12) or Methyl Chloride. Mechanical advantages include Timken Bearings, Centro-Ring Positive Pressure lubrication.

Special models are available for ice cream, frozen food cabinets and for the dairy industry.



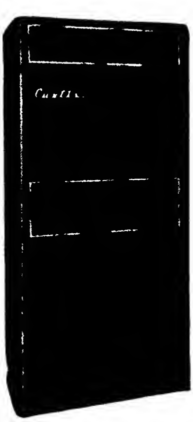
5 hp Water Cooled (Counterflow) Condensing Unit. Other sizes from ⅓ to 5 hp.

1½ hp Air Cooled Condensing Unit. Other sizes from ¼ to 3 hp.

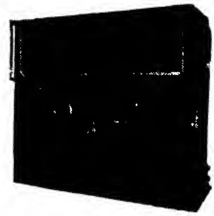
15 hp Cleanable Shell and Tube Condensing Unit. Other sizes from 3 to 30 hp.

Air Conditioning

For today's Air Conditioning requirements Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.



3 and 5 ton Packaged Type Air Conditioner.



7½-10-15 ton Remote or Central Type Air Conditioner

Frick Company

(Incorporated)

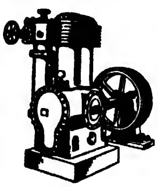
Air Conditioning, Refrigerating and Ice-Making Equipment

Waynesboro, Penna.

Atlanta
Boston
Buffalo
Charlotte
Chicago
Cincinnati
Dallas
Detroit
Kansas City
Los Angeles

Memphis
New Orleans
New York
Oklahoma City
Palatka
Philadelphia
Pittsburgh
St. Louis
Seattle
Washington

Distributors in FRICK **Principal Cities**



Enclosed Freon-12 Machine. Bulletin 508.

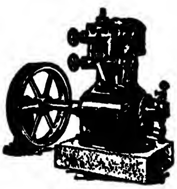
Ask for Frick Bulletins 502, 503, 504, and 505 on Air Conditioning

AIR CONDITIONING

Complete Frick Systems; also refrigeration for use with equipment supplied by others. Thousands of installations attest the value of Frick air conditioning. Successful experience with exacting commercial and industrial jobs enable us to solve your problems.

FREON-12 REFRIGERATION

Frick "New Eclipse" and the larger F-12 compressors provide a complete and efficient line. Coils, coolers, condensers and controls to suit. Patented Flexo-Seal at shaft, pressure lubrication from submerged pump, capacity controls, and other superior features make Frick machines your logical choice.



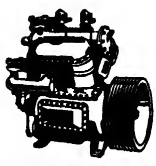
Enclosed Ammonia Compressor. Bulletin 112.

AMMONIA REFRIGERATION

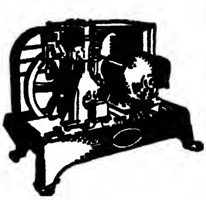
Combined units and vertical enclosed compressors, with two or four cylinders, in sizes from ½-ton up. Widely used for air conditioning, with material savings. Ask for Bul. 503 on this subject.

LOW-PRESSURE REFRIGERATION

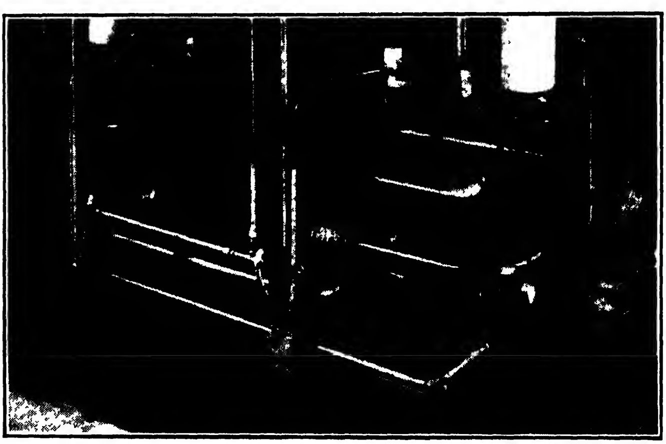
Commercial and industrial units in sizes from 4 hp up. Charged with Freon-12. Air and water-cooled condensers. Coils, coolers, and air conditioners. Get in touch w th your Frick Distributor; ask for Bul. 97. Our service includes estimates, layouts, manufacture, installation, and maintenance.



"New Eclipse" Freon-12 Compressor. Bulletin 100.



Low Pressure Refrigerating Unit. Bulletin 97.



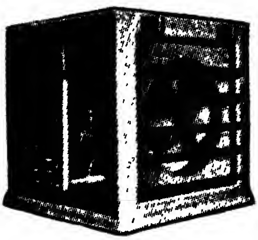
Two of Three "New Eclipse" Compressors of the Brass Rail Restaurant, New York City

Marlo Coil Co.

6135 Manchester Ave.. St. Louis 10, Mo.

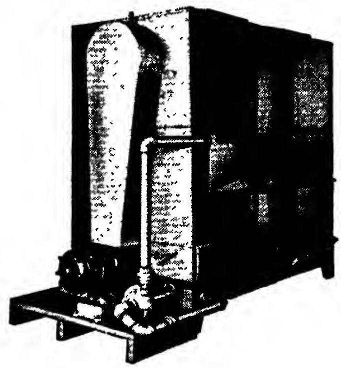
Manufacturers MARLO - HEAT TRANSFER Equipment
Since 1925

Industrial Coolers—Unit Coolers—Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils—Cooling Towers—Diesel Engine and Oil Evaporative Coolers.



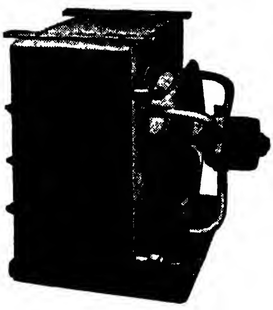
COOLING TOWERS

Triple-Type: Induced Air—Wetted Surface—Water Spray. Compact in space, weight and price. Outdoor—Indoor—3 to 100 tons—Built sectionally. Write for *Bulletin* 406.



EVAPORATIVE CONDENSERS

3 to 100 tons—All refrigerants—All prime surface coils—No fins—Quiet—Motor Unidrive. Indoor or outdoor units—Durable construction. Write for *Bulletin* 404.

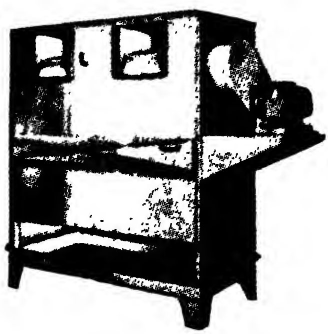


UNIT COOLERS

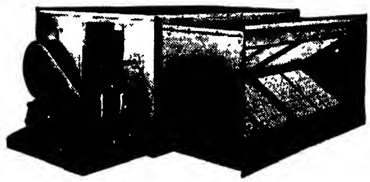
Pull-through (DUC) and Blow-through (UC) types—for all refrigerants. 11 unit sizes—4 and 6 row coils—full range of capacities. 675 to 4160 cfm—Venturi fan ring—Deflector louvers. Sturdy — Economical — Quiet. Also Wall Panel Type. Write for *Bulletins* 412 and 392.

INDUSTRIAL COOLERS

15 unit sizes—1000 to 24,000 cfm—Floor type. Galvanized frame and pans—Sectionally built. Variables: (1) Rows of coil and fin spacing. (2) Circulating brine spray or dry coil. (3) Defrost sprays optional. (4) All Refrigerants. Write for *Bulletin* 403.



BLAST COILS Air conditioning—Industrial Refrigeration—Heating. Any material—All refrigerants—Every application. "BALL-BONDED"—mechanically expanded tubes to fins. Write for *Bulletin* 396.

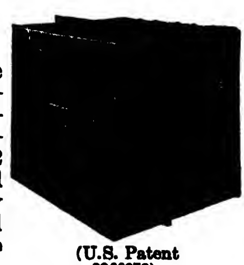


AIR CONDITIONING UNITS

Cooling — Heating — Dehumidifying—Humidifying. 10 sizes—3 to 35 tons—900 to 13,000 cfm. Ceiling suspended or floor types. Write for *Bulletin* 409.

ELECTRIC DEFROST LT UNITS

Compact ceiling type—High capacity—Low cost. 7 sizes—Ammonia or Freon—½ to 2½ tons at 12 deg TD. Defrosted electrically. Quickly installed — Sectional doorway-sized. Write for *Bulletin* 408.



(U.S. Patent 2266373)

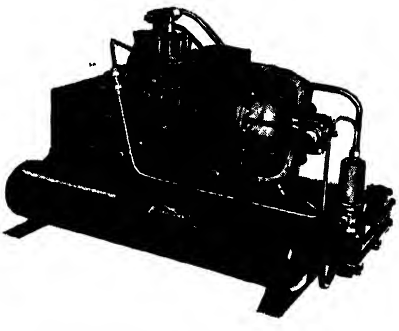
Servel, Inc.

Electric Refrigeration Division

Evansville 20, Indiana, U.S.A.

"Powered by Servel" Identifies A Better Product

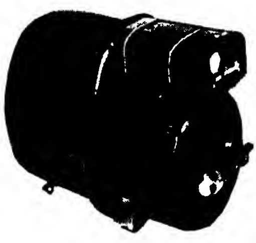
SERVEL SUPERMETIC



Servel Supermetic 3 HP condensing unit.

Servel offers a complete line of hermetic condensing units designed for all popular requirements in the commercial refrigeration and air conditioning field. Sizes range from $\frac{1}{4}$ to 5 hp, with a wide selection of both air and water cooled models. Servel Supermetic units are compactly constructed of highest quality materials. They have been use-tested in more than a hundred thousand installations. Each unit is backed by over a quarter century of experience in the manufacture of dependable, trouble-free refrigeration equipment.

Through careful workmanship, simplified design and many new improvements, Servel Supermetic condensing units are today outstanding for dependable operating efficiency, ease of installation and economy of maintenance. All moving parts are machined and finished to microscopic accuracy, thus assuring long and dependable service. Force-feed lubrication on every bearing, wrist pin and piston provides a constant film of oil to guard against wear and prolong the life of every vital part.



Fractional HP Power Units $\frac{1}{4}$ HP to $\frac{3}{4}$ HP.

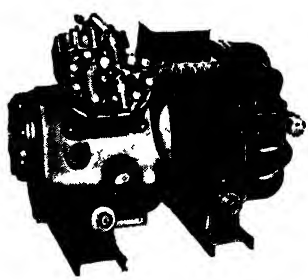
Electrical accessories and inter-connections are fully connected ready to run. All units are completely dehydrated, and furnished with holding charge of Freon-12 refrigerant and normal charge of Servel approved oil.

FOR MANUFACTURERS AND CONTRACTORS—Servel offers a choice of complete units, power units and "systems" for practically all types of fixture requirements. Servel Supermetic is ideally suited for window units, room coolers, store coolers. Three fractional hp power units for air conditioning and other high temperature applications include models DX (approximately $\frac{1}{3}$ ton); E2A (approximately $\frac{1}{2}$ ton); F2Q (approximately $\frac{3}{4}$ ton). Integral HP sizes include models G4A (1 ton); J4F (1-$\frac{1}{2}$ ton); K6F (2 tons); N6D (3 tons); P6K (5 tons). Truck body builders also find Servel Supermetic fully meets their requirements for dependable, low-operating cost refrigeration.

Servel's Engineering Department will be glad to cooperate in a selection of components and extend product testing assistance in its factory laboratories. Sample units are available to responsible manufacturers without obligation. Field sales representatives will be glad to make appointments for consultations on these matters upon request.

FOR DISTRIBUTORS—Servel offers its complete line of hermetically sealed and belt-driven condensing units to all parts of the world through franchised distributors and authorized retailers. Sales and service assistance is extended these customers by field representatives and factory personnel. Servel's national advertising assures acceptance everywhere.

Address inquiries to Servel, Inc., Electric Refrigeration Division, Evansville 20, Indiana.



Four and six-cylinder power units 1 HP to 5 HP.

Air Controls Inc
Div. of
The Cleveland Heater Co.

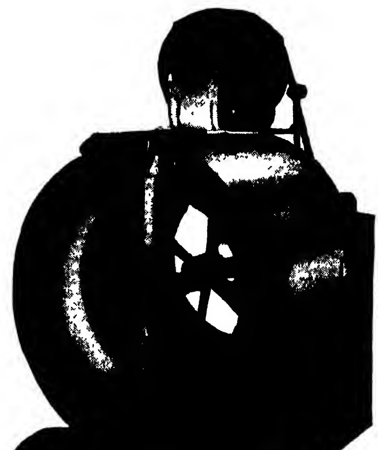
MEMBER
NATIONAL WARM AIR HEATING
INDOOR COMFORT
AND AIR CONDITIONING ASSOCIATION

2310 Superior Ave. **Cleveland 14, Ohio**

REX BLOWERS

DESIGNED and Engineered for MAXIMUM PERFORMANCE

Engineered to meet all modern needs and proof-tested for performance, **REX** Blowers set the pace for leadership by better design, smooth operation and trouble-free service. You can give your equipment an added sales feature by using **REX** Blowers —it's a feature you cannot afford to overlook.



Outstanding Features Of The REX Blower Line

Accessible, long-lasting oiling element.

The new streamlined bearing bracket with the sight-feed oil gage, and oil reservoir cap near the top of the blower, permits ease of lubrication. This revolutionary bearing holds 2 to 4 times as much oil as conventional bearings.

Adjustable cutoff . . .

Equally efficient against high or low resistances by simple adjustment.

A wide opening of the blower outlet produces better performance against low resistance and a narrower opening in the outlet is more efficient against high resistance. The adjustable cutoff provides a simple method of meeting this problem.

Rigidity . . .

The extra heavy construction of the new **REX** Blower line, insures sturdiness. Cross-bracing maintains alignment and quiet operation.

Reliable . . .

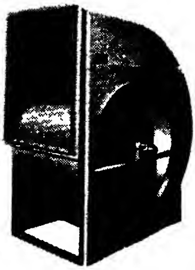
The operating record of thousands of **REX** Blowers which are in actual operation on air conditioning equipment, is the best testimonial of their reliability.

For Complete Information, Write for Catalog No. 247 Which Contains Complete Performance & Specification Data.

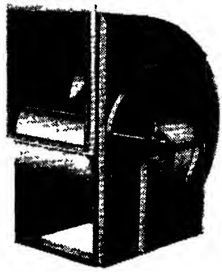
Aladdin Heating Corporation

2222 San Pablo Ave., Oakland 12, Calif.

Manufacturers of Centrifugal Blowers,
Heating and Ventilating Equipment.



FC Fan



BB Fan

EX Fan

The Aladdin FC Fan having a forward curved rotor is built in 14 standard sizes, single or double width, of 8 arrangements of drive and 8 directions of discharge. The low tip speed which is characteristic of this fan makes it ideal for general application where quiet operation is essential. Write for Bulletin No. 490.

The BB Fan is a backward curved fan with the non-overloading horsepower characteristic. This fan is built in 12 standard sizes, single or double width of 8 arrangements of drive and 8 directions of discharge. These fans are available in class I, II, III or IV and can be built for special application where required. Write for Bulletin No. 485.

The EX Fan is used chiefly for the conveying of materials, fume exhaust, etc. These fans are reversible and can be furnished in 13 standard sizes of 8 arrangements of drive and 8 directions of discharge. They can be desirable for special applications such as for handling abrasive materials or for acid fumes. Write for Bulletin No. 460.

The RB Fan having a radial curved rotor is used chiefly for kitchen exhaust duty. They are well suited for handling grease and other sticky materials, also for exhausting fumes and vapors from tanks, hoods, etc. This fan is built in 12 sizes, single width only, of 8 arrangements of drive and 8 directions of discharge. Write for Bulletin No. 450.

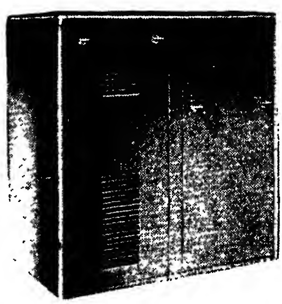
Fuseair ceiling outlets are manufactured in a complete range of sizes both in the supply type and the combination supply and return type. These units are fabricated from spun aluminum and all standard units are given an aluminum finish. Write for Bulletin No. 520.

Aladdin manufactures several types of forced air furnaces in a complete range of sizes. Many years of experience combined with the "know how" of our technical staff is reflected in the performance of these units. All Aladdin Forced Air Furnaces feature a large blower to insure the utmost in quiet operation. Write for Bulletin No. 500.

RB Fan

Fuseair

Forced Air Furnace



American Coolair Corporation

3606 Mayflower Street, Jacksonville 3, Florida

Exhaust Fans and Related Equipment for Industrial and Home Cooling.

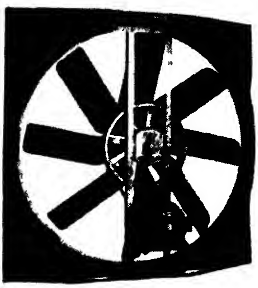
DISTRICT SALES MANAGERS:

EAST CENTRAL—Kurt Sprengling, 2002 Dallas Ave., Cincinnati 24, Ohio
SOUTH ATLANTIC—Robert A. Magee, 7402 Columbia Ave., College Park, Md.
SOUTHWEST—J. D. Clower, Whitewright, Tex.
WEST CENTRAL—Karl P. Schulze, 231 Phosphor Ave., New Orleans 20, La.

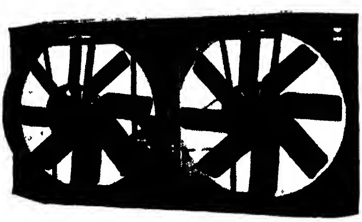
COOLAIR BELT DRIVE FANS are specifically engineered to move large volumes of air *quietly* and at low cost.

CERTIFIED RATINGS. Air delivery ratings are in accordance with ASHVE Standard Test Code for Centrifugal and Axial Fans (1938).

SOUND ABSORBING SPRINGS, a patented Coolair feature, assure extremely quiet operation. SKF ball bearings in all models.



TYPE O—for industrial, commercial and home use. Reversible. U. S. Patent 2191418. Fans with smaller motors listed are spring mounted.



TYPE OT (Twin)—This unit often fits perfectly where limited space prevents use of a single fan large enough for the job. U. S. Patents 2109838, 2191418.

TYPE S—For large industrial jobs. Heavy-duty double frame construction, pillow-block ball bearings.

Write for special bulletins and catalog sheets on above units, and the following equipment not shown here: Window Fans, Attic Packages, Shutters, Direct Drive Fans.

PERFORMANCE DATA and DIMENSIONS
Coolair V-Belt Drive Exhaust Fans

Fan Size	Q.R.*	hp	rpm	cfm	Nom. Size	Overall Dim. Ht.	Overall Dim. Width
2-O	A	1/4	570	6200	26	30⅝	30⅝
	D	1/3	630	6800			
2½-O	B	1/4	411	8000	32	36⅝	36⅝
	C	1/3	454	8800			
	D	1/2	522	10100			
	D	3/4	600	11700			
3-O	B	1/4	312	10000	38	42⅝	42⅝
	C	1/3	345	11000			
	C	1/2	398	12700			
	D	3/4	450	14400			
	D	1	500	16000			
3½-O	B	1/3	261	13000	44	49	49
	C	1/2	300	15000			
	C	3/4	345	17000			
	D	1	380	19000			
	D	1½	440	22000			
4-O	B	1/2	258	19000	50	55⅛	55⅛
	C	3/4	317	22000			
	D	1	353	25000			
	D	1½	405	28000			
4½-O	B	1/2	224	22000	56	61⅛	61⅛
	C	3/4	255	25000			
	C	1	276	27000			
	D	1½	319	32000			
	D	2	355	35000			
5-O	B	1/2	200	27000	62	67⅝	67⅝
	C	3/4	225	30000			
	C	1	245	33000			
	C	1½	282	38000			
	D	2	310	42000			
	D	3	355	48000			
2-OT Twin	B	1/4	455	9800	26	30⅝	61½
	C	1/3	500	10800			
	C	1/2	570	12400			
2½-OT Twin	B	1/3	359	14000	32	36⅝	73½
	C	1/2	411	16000			
	C	3/4	470	18200			
3-OT Twin	B	1/2	312	20000	38	42⅝	85½
	C	3/4	360	23000			
3½-OT Twin	B	3/4	272	27800	44	49	98
	C	1	300	30700			
4-OT Twin	B	1	258	38000	50	55⅛	110½
	C	1½	317	44000			
	D	2	345	48000			
6-S	B	1	155	35000	72	75½	75½
	C	1½	177	40000			
	C	2	195	45000			
	D	3	224	50000			
	D	5	270	60000			
7-S	B	2	170	58000	84	87	87
	C	3	195	67000			
	D	5	230	76000			
	D	7½	270	85000			
8-S	B	3	150	80000	96	99⅝	99⅝
	C	5	178	95000			
	D	7½	205	110000			
	D	10	225	120000			
9-S	B	5	150	110000	108	112½	112½
	C	7½	170	125000			
	D	10	190	138000			
	D	15	215	154000			

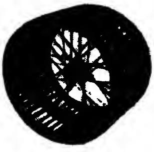
* Q.R., QUIETNESS RATING. A—Adjustable for quietness. B—*Very Quiet*, for homes, etc. C—*Quiet*, for stores, offices, etc. D—Industrial.

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee 14, Wis.**

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

TYPE "F" PLEXIFORM FANS AND TYPE "AP" AEROPLEX FANS



Plexiform and Aeroplex Fans are both designed for heating, ventilating and air conditioning service. The **Type "F"** fan is using a multi-blade wheel (see cut at left) while the type **"AP"** fan is built around a wheel of high speed design (see cut at right.)

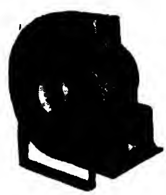
The wheels determine the character of the performance or the fan "Characteristics," thus the Type "F" is a slow speed fan having a rising power curve, while the **"AP"** is a high speed design and has self-limiting power characteristics. Both fans are highly efficient and can be built single inlet, single width, or double inlet, double width in ten different arrangements of drive and eight directions of discharge. The standard fans range in capacity from 1200 to 300,000 cfm and these fans are readily available in Class I or II designs for various types of duty. Class III and IV can be furnished on special application.

Type "EX" Fans

For all exhaust problems, pneumatic conveying, fume exhaust, etc. **"EX"** is highly adaptable, feasible and serviceable fan. It is sturdy in construction, easily installed and maintained. The design is reversible and standard sizes Nos. 15 to 80 inclusive are available.

Type "H" Fans

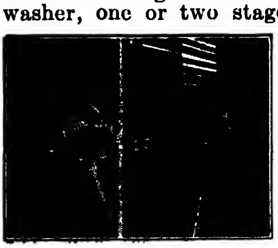
For pressure applications from 6 in. to 42 in. for belt drive or direct connection to motor, **Type "H"** is a most practical design. Standard sizes Nos. 25 to 80 are listed but the design is suitable for modification of wheel diameters to meet various motor speeds.



Both **"EX"** and **"H"** fans can be designed to fit specific applications. The design permits easy modification of details and the fans can be used for induced, forced draft, cupola service, primary or secondary air supply to oil burners, etc. Special details, outlet and inlet, or mounting can be designed to suit your special assembly.

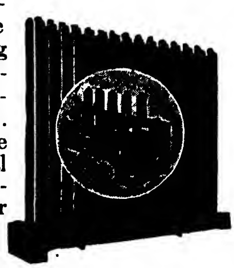
Turbo Spray

The Turbo Spray for industrial and comfort conditioning, cooling and humidifying, is a practical design as the atomization is by mechanical means, thus there can be no clogging or interruption of service even in atmosphere heavily laden with fibers or other foreign material. Single or multiple bank washer, one or two stage designs can be furnished from 2500 to 100,000 cfm capacity. To fit space requirements width and height can be modified to suit.



Chinook & Chinookfin

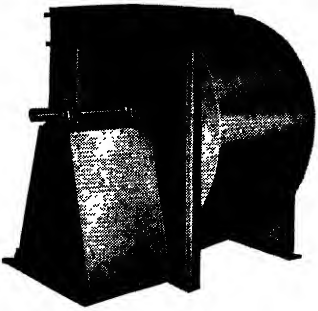
Both **Chinook and Chinookfin Coils** are based on the tube-within-a-tube design, originated by Bayley. The principle involved permits single header with all tube ends free expanding, thus eliminating the likelihood of freeze ups and cracking usually due to expansion and contraction strains. Both type coils are designed for usual hot blast heating and cover a wide range of capacities and applications.



The Bishop & Babcock Mfg. Co.

Massachusetts Blower Division

4901 Hamilton Ave. • Cleveland 14, Ohio

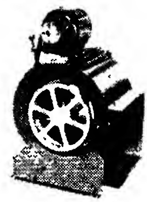


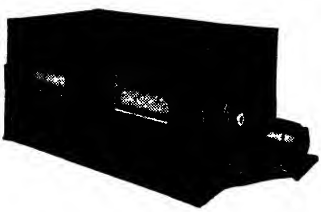
SQUIRREL CAGE AND POWER FIXED FANS

Squirrel Cage Fans, outstanding in performance, slow speed characteristics.

Power Fixed Fans are backward curve blade type, with non-overloading characteristics. Double width, double inlet, Class I or Class II construction, built in a wide range of sizes. Rating and dimension tables available. Write for ca alogs.

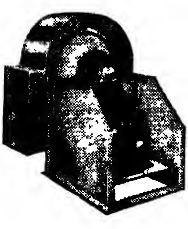
The new Design 2 Air Conditioning Furnace Blowers are now available, with a wide variety of stock combinations and discharge arrangements. They can be furnished in special widths or in multiples. Also available are wheel assemblies, housings and housing sides. Write for catalog.





Unit Heaters. Blower type. Floor and Ceiling type made in 13 standard sizes, with regular or non-freeze coils, filter and damper sections. Ratings from 50,000 Btu up. Propeller Fan type "H," Horizontal made in 16 sizes. Type "V" Vertical projection for ceiling mounting. Write for catalogs for full information.

Propeller Fans available with Belt drive with wheel sizes 24 in. to 48 in. Direct Drive with wheel sizes 12 in. to 36 in. Both types available with single or 2 speed motor. A complete line of Automatic Shutters and Fan Houses are obtainable. Write for catalog.



Ventilating Sets. Series MSV Design 2. Belt Driven, with or without motor and drive cover. Type "S" Design 2 are direct connected, designed for either supply or exhaust on small ventilation problems. Write for catalog.

Buffalo Forge Company

450 Broadway, **Buffalo, N. Y.**

Manufacturers of Unit Heaters, Multiblade Fans, Air Washers, Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Blowers, Exhausters, Disk Fans, Spray Nozzles. For Complete Information, Write for Bulletins, or See Your "Buffalo" Representative.

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TYPE "LL" FANS. Designed to ventilate large areas economically, these fans have the "Limit-Load" characteristic which makes motor overload impossible. Many new sizes, to allow exact selection for the job. BULLETIN 3675.

AXIAL FLOW FANS. For straight-line duct-mounted installation in ventilating or air conditioning service, these light, efficient, space-saving fans also have the "Limit-Load" feature. BULLETIN 3533-C.

AIR WASHERS. Available in combinations for air spraying, surface cooling, heating and filter cleaning to produce any desired air condition. Simple installation, maintenance requirements. BULLETIN 3142-C.

BREEZO-FIN HEATERS. Economical, attractive units finished in dull-lustre metallic brown. Suspended, out of the way, they get quick heat where you need it. Heating element is one-piece copper tube with square copper fins, spaced for maximum radiation. BULLETIN 3137-B.

INDUSTRIAL EXHAUSTERS. Made with sturdy, all-welded wheels and housings for more efficient air or material handling. Models for hot or corrosive gases. BULLETIN 3576.

PC CABINETS. Compact air conditioning units for (1) simple cooling, (2) cooling and de-humidifying, (3) heating and humidifying, (4) continuous air cleaning. Easily serviced. BULLETIN 502-A.

Burden Company

1000 North Orange Drive, **Los Angeles 38, Calif.**

Manufacturers of Fan Blades & Blower Wheels

Burden Blades and Blower Wheels are designed to operate smoothly and efficiently, on a minimum of horsepower, and to give maximum volume of air flow. All Burden Blades and Blower Wheels are tested for dynamic and static balance by the patented Burden Electronic Balancer. Burden products are outstanding for their low noise level, resistance to corrosive action, slight weight and great strength. For specifications of Blades and Blower Wheels and for types not listed, write factory.

ATTIC TYPE BLADES

Quiet and highly efficient, with substantial aluminum paddles on heavy gage, one-piece steel spiders.

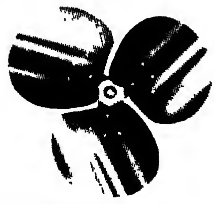
Sizes, 30 in., 36 in., 42 in., 48 in.

Diameter, 1 in. bore.

4 BLADE PRESSURE PROPELLERS

Have attained national recognition for their high efficiency. Maximum air volume with almost silent operation. Light weight and balanced to add years to motor life.

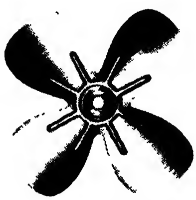
Clean and attractively finished aluminum blades, cadmium plated steel hub and spider. From 8 in. to 20 in. diameter, ANY pitch up to 30 deg.



3 BLADE SEMI-PRESSURE PROPELLERS

Smooth and quiet in operation, this fan is noted for its economy. The initial cost is low and only minimum horsepower is required for operation. Its engineering simplicity and rugged construction make it a favorite with industrial users, especially in the larger sizes.

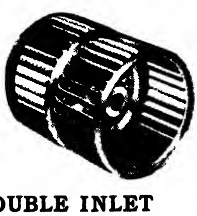
Available in 8 in. to 30 in. diameters with pitches up to 33 deg.



ONE-PIECE FREE AIR BLADES

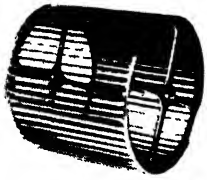
New, broader blades designed to give maximum cfm from smallest fractional motors. Attractive appearance in bright finish aluminum or can be delivered anodized in colors—Red, Copper, Green, Blue—at slight extra cost.

Standard sizes are as follows: 6 in., 7 in., 8 in., diameters. Available in 17 deg, 23 deg, 29 deg, pitches.



DOUBLE INLET WHEEL

Available in a variety of sizes, from 3 in. to 8¼ in. diameter, 3¾ in. to 7½ in. width.



DOUBLE INLET WHEEL

(Spoke End Ring Type) From 10¼ in. x 8⅛ in. to 16¼ in. x 16 in.

Fabricated from one continuous strip, the vanes of Burden blower wheels are lightweight but rugged, of uniform pitch and curvature, with excellent high pressure characteristics. Exclusive Burden machines and methods result in low cost. Each wheel balanced by electronic equipment, a Burden feature.

SINGLE INLET BLOWER WHEEL

Sizes: 3 in. through 8¼ in. diameter; 1⅞ in. through 3¾ in. width.

Champion Blower & Forge Co.

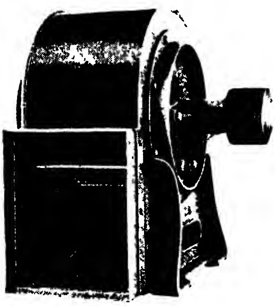
Manufacturers and Engineers

Plant & Offices: Lancaster, Pa.

Address Correspondence to Div. 9

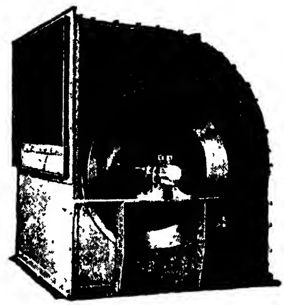
Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates

Representatives in Principal Cities



Type S

Type S Forward curve ventilating fans, single and double width.
Sizes 30 in. to 60 in. wheel dia. →

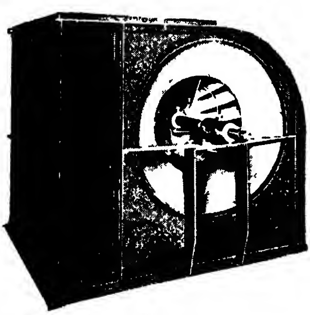


Type S

Type S Forward curve ventilating fans, single and double width, and electric drive up to 36 in. wheel diameter. ←

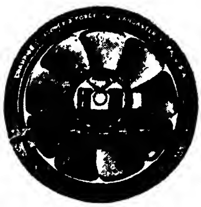
Type CE Electric cast iron exhaust and forced draft fans. Also volume control dampers. →

Type CE



Type BC

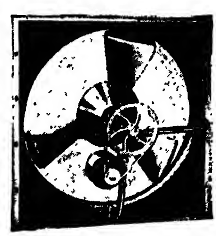
Type BC Backward curve ventilating and exhaust fans, single and double width; belt driven and direct connected electric.
Sizes 12 in. to 60 in. wheel dia. ←



Type SV

Type SV Super Ventilating fans, direct motor drive up to 36 in. diameter. Motor belt drive up to 48 in. size. ←

Type A Industrial Ventilating and domestic attic fans. Built in sizes 30 in., 36 in., 42 in., 48 in. →



Type A

Chelsea Fan & Blower Co., Inc.

1206 Grove St., Irvington 11, N. J.

REPRESENTATIVES IN PRINCIPAL CITIES

Charter Member Propeller Fan Manufacturers Association

Trade Mark Registered

EXPORTED TO ALL COUNTRIES OF THE WORLD BY WESTREX CORPORATION

(*Western Electric Export Corporation*)

CHELSEA INDUSTRIAL FAN - TYPE IND

Cat. No.	C.F.M.	Motor HP
IND24	5500	1/3
IND30	8500	1/2
IND36	11200	1/2
IND42	16200	3/4
IND48	21500	1
IND54	27000	1 1/2
IND60	32000	1 1/2

Fans for corrosive fumes or excessive heat quoted on request.

This fan is one of the most efficient fans built today. Under static pressure, that is, operating in moderate length ducts or against automatic louvers, the Industrial fan can move more air with less power than most fans on the market. The carefully designed venturi orifice and blades make this efficiency possible.

The fan is belt-driven for three reasons:—first, to reduce power consumption through increased efficiency of higher speed motors; second, to give more quiet operation through reduced fan speeds; third, to reduce original cost of fan to consumer.

The frame is all steel, welded to a steel orifice. It is rigid, yet light in weight, simplifying installation.

On special order, these fans can be built to operate against static pressures up to 2 in.

TYPE OPJ—OCTOPUS JR.
Portable Exhauster and Blower

3-4" HOSES
CFM Per Hose
10' = 412 50' = 297

TYPE DXB—BOOSTER
Spray Booths, Fumes, Excessive Heat

16" to 36"
2000 to 9500 CFM

BB—ALL PURPOSE
Removes Smoke, Steam, Heat and Fumes

12" to 30"
1200 to 7000 CFM

Bulletins and Engineering Data furnished on request

Air Delivery Ratings of all Chelsea Products are in accordance with the *Standard Test Code for Centrifugal and Axial Fans* of the *Propeller Fan Manufacturers Assn.* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. INSIST ON CERTIFIED RATINGS. LOOK FOR THIS SEAL.

Certified Ratings
Air Deliveries are in accordance with Standard Test Code for Centrifugal and Axial Fans
MEMBER OF
PFMA
PROPELLER FAN MANUFACTURERS ASSN

DeBothezat Fans Division

American Machine and Metals, Inc.

Main Office and Factory—East Moline, Illinois

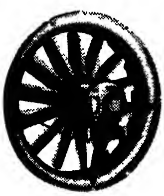
In Canada: American Machine and Metals (Canada) Ltd., 1144 Weston Road, Toronto, Ontario.

Foreign Sales Office: Woolworth Building, New York 7, N. Y.

SALES ENGINEERING OFFICES IN ALL PRINCIPAL CITIES

Certified Wind Tunnel Ratings • Non-Overload Power Characteristics

EFFICIENT AGAINST PRESSURE



Axial-Flow Vent Set

DeBothezat Axial-Flow Fans are built in a wide range of sizes, from 16 inches through 12 feet in diameter. Volume fans (4 blades) are for low to moderate static pressures. Pressure fans (14 blades) are for high static pressures, up to 2 inches SP. Non-overloading power characteristic prevents motor burn-out and eliminates need for oversize motor. All DeBothezat Axial-Flow Fans have certified performance ratings. Catalog furnished on request.

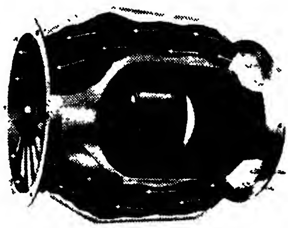
DeBothezat Giant Fans, 5 feet through 12 feet in diameter, are suitable for gear, "V" belt or direct drive, for use with electric motor, steam turbine or gasoline engine. Catalog furnished on request.

CONTROLLED VENTILATION

Power-Flow Roof Ventilator

Motor driven fan with weatherproof housing provides positive controlled ventilation at all times, independent of wind or weather. Streamlined appearance, low height. Operates efficiently with or without duct system. Hinged hood permits ready access to fan and motor. Available in four sizes, with fan wheels 16 inches through 36 inches in diameter. Catalog furnished on request.

FUME REMOVAL



Bifurcator (Cut-away view)

For exhausting air that is abnormally hot, corrosive, flammable or explosive. Motor is mounted in separate, through-ventilated chamber completely isolated from air stream. Destructive fumes are by-passed (bifurcated) around motor. DeBothezat Bifurcators install directly in the duct, in any position from horizontal through vertical. Available in eight sizes, with fan wheels 18 inches through 48 inches in diameter. Catalog furnished on request.

SPOT COOLING

"Hy-V" Air Jet

DeBothezat "Hy-V" Air Jets are extraordinarily efficient man coolers, particularly for workmen who are exposed to radiant heat from ovens, furnaces, forges and molten metals. Arranged for either column or floor mounting, they are adjustable to blow a concentrated blast of cooling air in any direction to remote spaces without the aid of ducts. Available in 18 inches through 30 inch sizes. Catalog furnished on request.

General Blower Company

8622 Ferris Ave., **Morton Grove 5, Ill.**

Engineering Offices in all Principal Cities

FOR BETTER AIR MOVING RESULTS USE GB LUNGS FOR INDUSTRY

Engineering • Manufacturing • Application of Blowers, Fans and Exhausters • for Blowing • Heating • Cooling • Conveying • Aerating • Separating • Agitating

A—Turbo Blowers

More than 20 years' experience in the construction of Blowers, Fans, and Exhausters for leading Industrial Users in widely diversified fields.

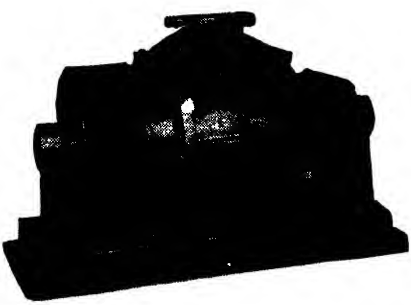
Engineering skill and integrity in the Building and supplying of Centrifugal and Rotary Positive types of Blowers, Exhausters and Gas Boosters enables us to supply you with the type of equipment best suited to your particular need, whether "standard" or "custom built."

B—Steel Plate Exhausters

We maintain Sales Engineering offices in all large cities. It is desirable and advantageous to consult with our local engineering representatives on all types of air-handling applications. They are at your service to help with your problems and to bring the engineering skills and resources of the General Blower Company to your organization.

Write for our new, illustrated Products Bulletin
"LUNGS FOR INDUSTRY"

C—Welding Fume Exhausters



F—Rotary Positive Blowers

E—Exhaust Fans

D—Multiblade Exhausters

Hunter Fan & Ventilating Co., Inc.

Exclusive Fan Makers Since 1886

400 S. Front Street, **Memphis, Tenn.**

Sales Offices

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SAN ANTONIO—218 Chevy Chase Dr., Terrell Heights
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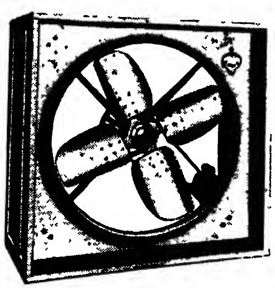
BELT DRIVEN PROPELLER FANS, WINDOW FANS, AIRSPREADS, CEILING FANS, OSCILLATORS, LIGHT DUTY EXHAUST FANS.

For Home or Industrial Ventilating and Cooling Service

Consult the Hunter Representative nearest you or write the Hunter Fan and Ventilating Company, Inc., P. O. Drawer 2858, Memphis 2, Tennessee, for our manual which contains ventilating and cooling methods and installation details. Our Engineering Department will assist you with your air cooling or ventilating problem. See our section in Sweet's.



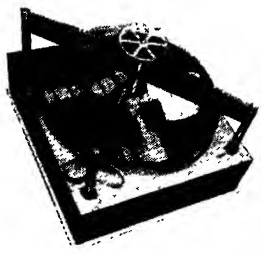
FEATURES

Ball bearing throughout
Designed for installation in any position
Streamlined orifice—integral part of 14 gage steel frame
Heavy die formed blades—balanced to assure quiet, vibrationless operation
Long life, ball-bearing motors with built-in thermal overload protectors
Certified air delivery ratings are in accordance with tests by A & M College of Texas using Standard Test Code for Centrifugal and Axial Fans of the A.S.H.V.E. and *PFMA*.
Underwriters' Laboratory Label with re-inspection service
Fan guaranteed for five years—Motor guaranteed for one year

MECHANICAL SPECIFICATIONS, CAPACITIES, WEIGHTS, ETC.

Cat. No. Complete Unit	C.F.M. Del. Static Pressure Zero	C.F.M. Del. Static Pressure 1/10"	Blade Diameter	Fan R.P.M.	Motor Specifications H.P.	Motor Specifications Volts	Overall Dimensions Fan Crate	Shipping Weights
50-AXB	5100	3800	24"	580	$\frac{1}{6}$	115	34 x34 x18"	122
75-BXB	7500	5500	30"	420	$\frac{1}{4}$	115	$42\frac{1}{4}$x$42\frac{1}{4}$x21"	177
10-DXB	11400	8300	36"	360	$\frac{1}{3}$	115/230	$48\frac{1}{4}$x$48\frac{1}{4}$x21"	197
14-FXB	16000	12000	42"	340	$\frac{1}{2}$	115/230	$54\frac{1}{4}$x$54\frac{1}{4}$x$25\frac{3}{4}$"	279
18-HXB	20000	14200	48"	295	$\frac{1}{2}$	115/230	$60\frac{1}{4}$x$60\frac{1}{4}$x$25\frac{3}{4}$"	309
18-HXB-3	22500	18400	48"	330	$\frac{3}{4}$	115/230	$60\frac{1}{4}$x$60\frac{1}{4}$x$25\frac{3}{4}$"	319

NOTE: 220–440 volt, 3-phase motors in all sizes except $\frac{1}{4}$ H.P. available at no extra charge. Two-speed motors available at extra charge.



Package Attic Fan

Hunter package attic fans are offered in two sizes—7500 cfm and 9500 cfm. Specifically designed for vertical discharge, with manually-operated shutter. Automatic lock and release mechanism starts and stops the fan as shutter opens and closes. The unit is complete with fan, motor, shutter, and trim. No accessories are required—ready for instant use summer or winter. Ideal for small home or recreation room or for year round ventilation of many different applications.

Window Fan

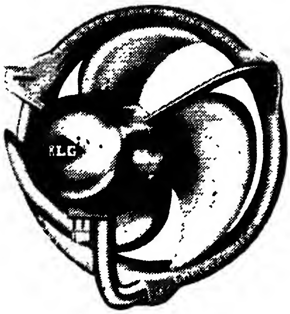
A two-speed direct-drive window fan which delivers 4250 cfm on high speed and 3600 cfm on low speed. The fan cabinet is 32 in. x 32 in. x 9 in. overall. Adjustable mounting brackets permit the fan to be installed in any average size window. Louvres in the compact light ivory colored case offer beauty and protection and are streamlined for smooth quiet air flow.

ILG Electric Ventilating Co.

2876 North Crawford Ave., **Chicago 41, Ill.**

Offices in more than 40 Principal Cities

Propeller Fans, Centrifugal Fans, Unit Heaters, Kitchen Ventilators, Night Cooling Fans

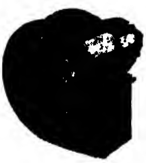


ILG Direct-Connected Self-Cooled Motor Propeller Fans

Used for exhaust of stale air, fumes, heat, dust, odors, etc. Self-cooled motor combines protection of enclosed motor with low operating cost of open motor—constantly cooled by fresh, clean air, circulated internally—never "gums-up" from contact with foul air—saves 5 to 10 per cent on power costs. Rugged, heavy-duty framework. Dynamically-balanced fan wheel, direct-connected to motor. Smooth, quiet, effortless operation—economical, long lived. "ONE-NAME-PLATE" Guarantee. Certified ratings. Sizes, 8 in. to 72 in.

ILG Direct-Connected Centrifugal Fans

Type "BC"—Load-limiting type with backward curved blades. Motor load remains constant over wide range of air volume and change in static pressure. Wheel mounted directly on motor shaft with motor partially recessed in side of casing. No motor base required. Unobstructed inlet. 10 sizes. Also available for belt-drive in 12 sizes.



Type "B" Volume Blowers

Small volume, low pressure, quiet running. Multiblade wheel direct-connected to motor shaft. Cast iron base. Universal discharge. 12 capacities.

Type "P" Volume Blowers

For exhausting dust, fumes, removal of steam, vapors. Four discharge positions to avoid friction in short bends. 7 capacities.

Type "6S" Utility Blowers

Designed for building into apparatus which requires ventilation or air movement. Extremely flexible in arrangement, furnished with or without inlet flange, outlet flange stand, etc.

Kitchen Ventilators

Built-in becomes permanent part of kitchen wall. Quiet high capacity, efficient, equipped with ILG Self-Cooled Motor. 3 sizes. Also models for window installation.

Night Cooling Fans

Portable model for use at attic or downstairs window. For permanent installation in attic, use ILG Self-Cooled Motor Propeller Fans (top of page).

JOY MANUFACTURING CO.

General Offices: Henry W. Oliver Building, **Pittsburgh 22, Pa.**

MANUFACTURERS OF VANEAXIAL FANS

DOMESTIC OFFICES

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AND MORE THAN 500 DISTRIBUTORS THROUGHOUT THE WORLD

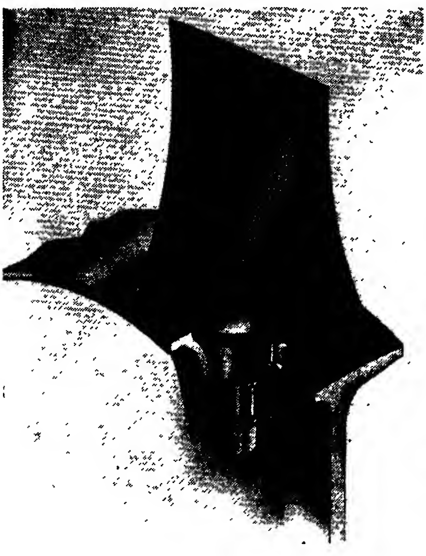
SERIES 1000 AXIVANE* INDUSTRIAL AND COMMERCIAL FANS

Joy Series 1000 adjustable blade **AXIVANE*** Industrial fans are available in 124 sizes ranging in volume capacity up to 100,000 cfm with pressures up to 9.6 in. W. G. Housing diameters range from 18 in. to 60 in. For complete specifications, construction details, and selector charts giving pressure-volume range for each fan, write for bulletin number J-605.

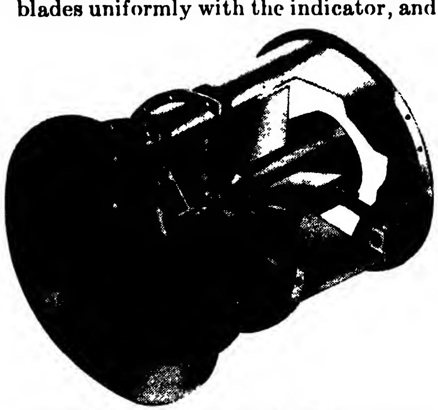
Joy **AXIVANE*** Series 1000 fans are efficient, quiet, compact, flexible, and easy to install.

ADJUSTABLE BLADES

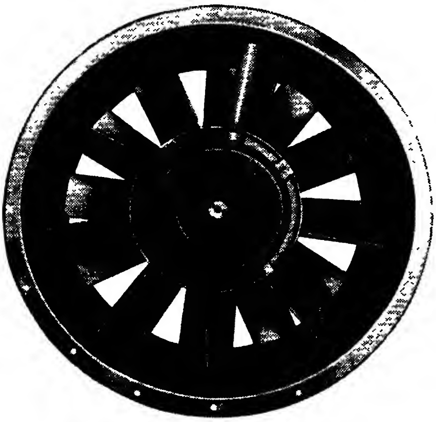
Joy **AXIVANE*** Industrial Fans have the extra performance flexibility of adjustable blades. Adjustable blades are standard equipment on all Series 1000 fans. The factory blade setting can be quickly changed to provide either a wide pressure range for any particular volume or a change in volume simply by loosening a lock nut with a wrench, setting the blades uniformly with the indicator, and



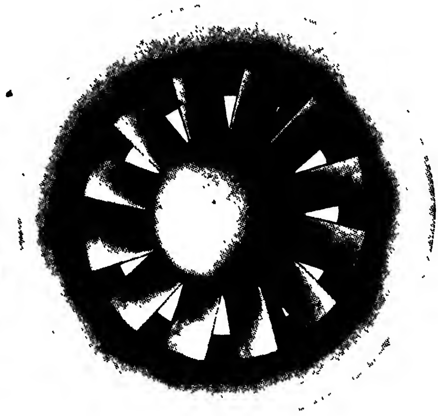
Blades are adjustable on the job by loosening one lock nut



Cutout drawing showing location of motor and compact construction



Rear view, showing vanes and motor



Front view, showing simplicity of construction

retightening the lock nut. A permanent stop prevents setting blades in a position likely to overload the motor. Minimum blade settings are limited by the fan housing.

Adjustable blades permit on-the-job correction for unpredictable duct resistance or for poorly installed duct work.

MORE EFFICIENT

Stationary straightener vanes, located immediately behind the rotor, partially recover the rotative energy imparted to the air by the rotor, and re-establish axial flow to the air leaving the vanes. This eliminates excess turbulence at the point where the air enters the duct system and increases efficiency by decreasing pressure loss.

The Joy **AXIVANE*** fan utilizes an aerodynamically efficient blade and stationary vane design.

QUIETER OPERATION

For equal weight and space the Joy **AXIVANE*** fan is quieter than a centrifugal type fan of equal volume and pressure. The streamlined airflow from an **AXIVANE*** fan makes sound insulation a simple and inexpensive operation when required for the ventilation and air conditioning of quiet spaces such as hospitals, auditoriums, radio stations, etc., where insulation against system noise must be used.

MORE COMPACT

Joy **AXIVANE*** fans are built around the motor, the fan housing becoming an actual part of the duct system. This produces a more compact design than is possible with a centrifugal fan. An **AXIVANE*** fan, installed on an in-line connection with ventilation ducts, parallel to and close by an overhead structure, may require 70 per cent less space than a conventional belt-driven centrifugal fan. The compactness of a Joy **AXIVANE*** fan assures a maximum of net operating or rentable area. Fan rooms are virtually eliminated.

EASIER TO INSTALL

The Joy **AXIVANE*** Series 1000 fan develops a greater volume and pressure per pound of fan and motor because of its compact, in-line construction. This light weight permits a simplicity of installation that minimizes installation costs and total weight by eliminating heavy foundations, complex duct offsets and elbows, drives, and guards. **AXIVANE*** fans can be installed quickly and easily, even by relatively inexperienced or un-skilled labor.

MATCHED ACCESSORIES

Inlet bells, screens, and fan supports are accessories designed to fit all **AXIVANE*** Fan housings. In ordering, it is only necessary to state the model number with or without the accessories, as desired. If required with accessories, these will be furnished to fit the fan model ordered without special number.

No matter how carefully a duct system is planned, an incorrectly selected inlet bell will reduce fan efficiency by increasing intake turbulence. This excess turbulence will also increase the noise level of the fan. When a fan takes its air directly from the weather, a plenum, a fan room, or from a duct system larger in circumference than the fan housing, a bell should be used.

*Reg. U. S. Pat. Off.

The Lau Blower Company

2007 Home Avenue, Dept. H, **Dayton 7, Ohio**

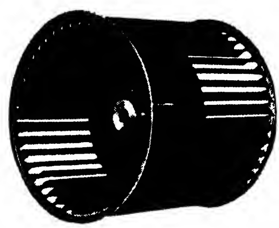
Engineers and fabricators of general Air Handling Equipment
Blower Assemblies • Blower Wheels • Propeller Fans • Accessories

NEW Series "A" Blower Assemblies

Special Features Patents Applied for

The Lau Series "A" Blower Assembly—result of years of exhaustive tests of all types of blowers—years of research and design evolution—is the all-time, outstanding achievement in the blower field. Greater mechanical strength. Greater efficiency. A more compact unit (overall size considerably smaller than formerly). Will fit more jobs. Embodies many new and revolutionary features exclusive with Lau. Includes new 3-point suspension type bearing bracket—an integral part of the shroud—identical for various angles of discharge. New frictionless, self-aligning bearing—completely encased in Neoprene. New center suspension wheel (see below). New discharge outlet design and construction. Cut-off cannot set crooked on outlet. No wavy edges. Faster installations. New 1-piece motor mounting easily convertible, rear to top or vice versa, by simple use of two sheet metal screws. Many other features. Complete range of sizes. Every size tested and rated for performance in accordance with A.S.H.V.E. and *N.A.F.M.* Codes.

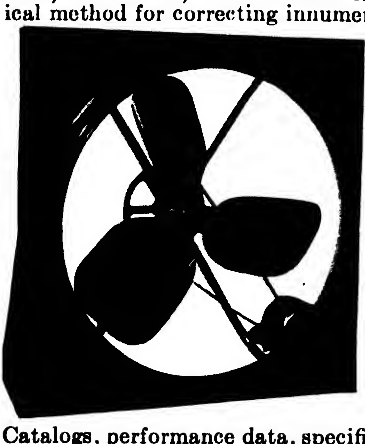
NEW Series "A" Blower Wheels

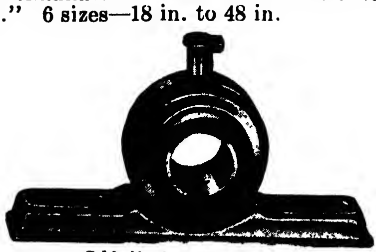


New center-suspension wheel *tested and proved* by us to have greater mechanical strength, truer concentricity, and far more efficient performance than ordinary types of wheel. Complete details supplied to interested parties upon request.

Propeller-type "Niteair" Fans

For a wide variety of applications where it is necessary or advantageous to exhaust undesirable air and provide fresh air from the outside. Equally applicable for industrial, commercial, residential and farm building installations. Efficient and economical method for correcting innumerable air-control problems—removing dust-laden, foul, contaminated, or excessively hot air, fumes, gases, smoke. Circulating cool night air through living and sleeping rooms for greater summer comfort. Venturi-type entrance housing reduces air "drag" and turbulence—eliminates most common cause of "air noise." 6 sizes—18 in. to 48 in.





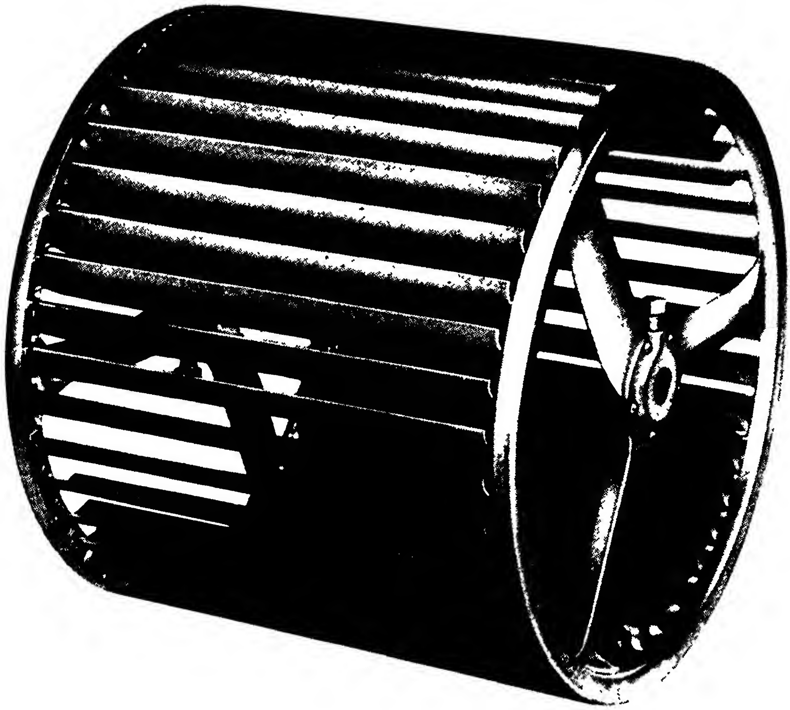
Self-aligning Pillow Blocks

Catalogs, performance data, specifications, and prices available on request on above and other air handling equipment. Inquiries solicited for any application. Our engineers will gladly assist you. ***Write Dept. H regarding your requirements.***

Morrison Products, Inc.

A 25 Year Firm

East 168 Street and Waterloo Road, **Cleveland 10, Ohio**



MORRISON AIRSTREAM BLOWER WHEELS—*Made Exclusively for original equipment manufacturers* in heating, ventilating and air conditioning.

Morrison Airstream Blower Wheels are double width—double inlet in standard diameters from 10 in. to 16 in., and in width from 6 in. to 16 in.

One-Piece Blade Construction. Three-Piece Balanced Assembly—one-piece blade and two pressed rings with integral hubs welded together.

Equalized Weight Distribution with end mounting. No shaft whip—reduced deflection.

Available with Morrison Airstream Blower Wheels is Complete Engineering Service. Included are templates, shop drawings, tables, data, cost analysis, graphs, charts, sources of component parts. Housing Squares and Scroll Sides available for Low Cost Assemblies.

Catalogs: Morrison Blower Wheels, Morrison Engineering Service, Morrison Cost Analysis Estimate Book—Copies Mailed Upon Request.

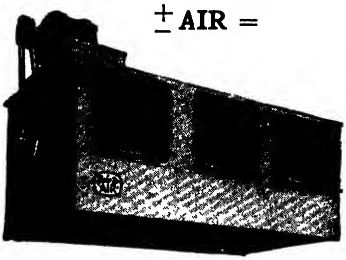
The NEW YORK BLOWER COMPANY

GENERAL OFFICES
3145 SOUTH SHIELDS AVENUE • CHICAGO 16
FACTORIES at LA PORTE, IND and CHICAGO, ILL

Representatives in Principal Cities

FANS ● BLOWERS ● UNIT HEATERS ● MAKE-UP AIR UNITS
AIR WASHERS ● HEAVY DUTY HEAT SURFACE

Make-up ± AIR =



M15—15,000 cfm

A unit that delivers, warmed, filtered, outside air to industrial spaces to replace exhausted air and balance minus pressure. Corrects draft conditions and uncontrolled infiltration. Made in 4 sizes from 5,000 cfm to 20,000 cfm. Described in *Bulletin 453*.

Type ME Centrifugal Fans

Capacities up to 101,000 CFM

Slow speed wheels offered from 7½ to 15 in. and 36 to 66 in. Capacities up to 70,000 cfm.

Quiet operating medium speed wheels with non-overloading horsepower characteristics for heating, ventilating and air conditioning or industrial applications. Wheel diameters from 18 in. to 66 in., with any speed or discharge required. Class I, II, III or IV construction. *Write for Bulletin 471*

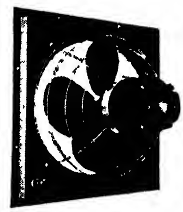
Steelfin Hot Blast Heating Surface

Extra heavy duty, fin-and-oval tube, all-steel, welded construction. A hot dip metallic coating over all, including headers, affords perfect bonding and conductivity. Suitable for continuous heating service on steam pressures up to 150 lb.

Comet Unit Heaters

Heavy duty, welded steel, fin-and-tube heating element. Suitable for continuous heating service on steam pressures up to 150 lb or more. 9 sizes with capacities from 31 Mbh to 300 Mbh. *Bulletin 485*

Comet Exhaustair



Delivers large volumes of air at low resistance and low current consumption. All wheels are machine balanced for smooth, vibrationless operation. Made in two types and eight basic sizes. Wheel diameters from 12 in. to 60 in. Direct or belted drive. Capacities from 400 cfm to 22,000 cfm. *Ask for Bulletin 462.*

Type GI Industrial and Heat Fans

For dust and gas removal, conveying of materials and handling hot gases. Housings, drives, and discharge arrangements to meet any requirement. Wheel diameters from 16 in. to 66 in. Capacities from 450 cfm to 60,500 cfm. *Details and engineering data in Bulletin 482.*

General Purpose Fans

Portable, self-contained units for Class I industrial and ventilating applications. Recommended for ease of installation, low maintenance and space saving features. Made in three types and eight basic sizes. Capacities 400 cfm to 18,000 cfm. *Bulletin 481.*

PROPELLAIR Div.

ROBBINS & MEYERS, INC.
1947 Clark Boulevard
SPRINGFIELD, OHIO

VENTILATING SPECIALISTS IN ALL PRINCIPAL CITIES

For Ducts, Walls, Windows, Hoods, Roof Ventilators

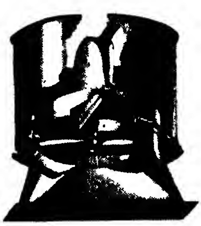
PROPELLAIR DIRECT-CONNECTED FANS

For use wherever motors may operate within the air stream, from free air to medium and relatively high resistance. A compact, durable design with fan having two to six blades. Sizes: 12 in. to 60 in. Capacities: 800–85,000 cfm. Type "CD."

For Heat, Acids, Alkalies—Fumes, Gases, Dust

BELT-DRIVEN PROPELLAIR DRUM-TYPE

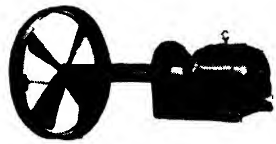
A complete fan unit in short duct section ready for installation in lines from 20 in. to 48 in. diameter. Type "CS" may be used for severe acid or alkaline conditions, explosive fumes and gases. Type "CSV," for excessive temperatures, circulates outside air through belt and fan shaft tubes to keep drive and bearings cool. Also furnished without duct section. Capacities: 4100 to 43,000 cfm.



For Roof Ventilation

PROPELLAIR VERTICAL EXHAUSTER

Dependable and economical power roof ventilators. Butterfly dampers open wide the instant fan is started, close automatically as fan coasts to a stop, offer virtually zero resistance as heat, fumes, moisture, dust shoot high into air. Rain is prevented from entering by fan when operating. Drainage gutter prevents leakage when dampers are closed. Sizes: 20 in. to 60 in. Capacities: 3700 to 77,000 cfm.

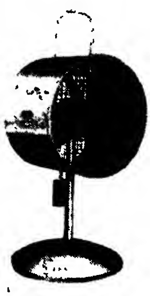


For Heat, Moisture, Fumes, Dust, and Gases

PROPELLAIR EXTENDED-SHAFT FANS

This design locates motor outside air stream when fan is installed in duct at right angle turn, elbow, "Y," or offset. Simple installation usually can be supported by duct without auxiliary bracing. Drive shaft is enclosed and sealed within steel tube. Sizes: 12 in. to 60 in. Capacities: 2000 to 68,000 cfm. Type "CE."

For Cooling Men and Materials

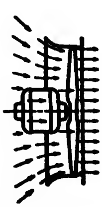


PROPELLAIR PEDESTAL & CRADLE-MOUNTED FANS

Pedestal-type, high velocity mill type portable fans have sturdy steel drum protected front and rear, formed steel base, rigid support, and crane hook. Also made with adjustable cradle mounting (Type "CU"). Capacities to 20,650 cfm.

AIRFOIL-SECTION Blades

Airfoil Principle Entrance Ring



Propellair fans have airfoil-section blades with variations of pitch, curvature, and thickness to compensate for different lineal speeds of points at various radii. Air movement is uniform over whole fan area. The Propellair curved entrance ring eliminates eddy currents; helps efficient Propellair blades deliver highest pressure and volume.

Schwitzer-Cummins Company

1145 E. 22nd Street

INDIANAPOLIS, U. S. A.

Engineers and Manufacturers of Centrifugal Blowers and Blower Wheels—Attic Fans—Exhaust Fans—Ventilating Fans of all Types

PRODUCTS OF THE VENTILATING DIVISION

HY-DUTY BLOWERS FRESH-AIR MAKER FANS

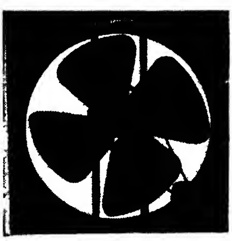
Over a hundred models to fit the big majority of ventilating and cooling jobs—applicable as well to specifications where the fan or blower is to be built into a manufacturer's product—designed for fine performance, quiet operation—all ruggedly built and simplified for reasonable cost.

• **HY-DUTY BLOWERS**—wheel diameters—8½ in.-25 in., single inlet, double inlet. Blower Wheels—4½ in.-50 in., single or double inlet—our sizes or your specifications—galvanized wheels a specialty.

• **FRESH-AIR MAKER—ATTIC FAN**—24 in.-48 in., ball bearings for vertical or horizontal positions.

• **FRESH-AIR MAKER—EXHAUST FANS, PORTABLE FANS, ADJUSTABLE WINDOW TYPES**—12 in., 16 in., and 20 in.—3 speed-large output-good looking-beautifully balanced.

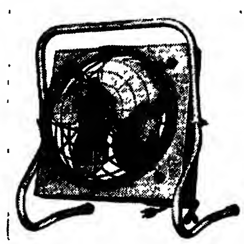
Send for catalog on standard models or submit your engineering specifications.

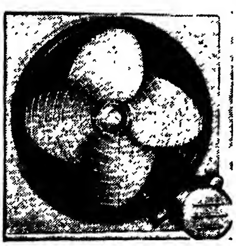


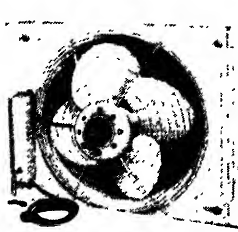
FINE FANS

for

OVER 30 YEARS







TRADE-WIND CLIPPER BLOWER

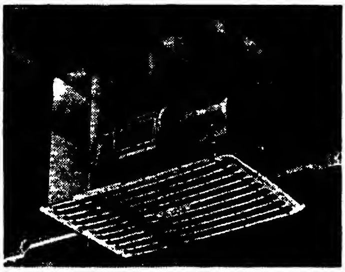
Trade-Wind Motorfans, Inc.

5725 So. Main St.

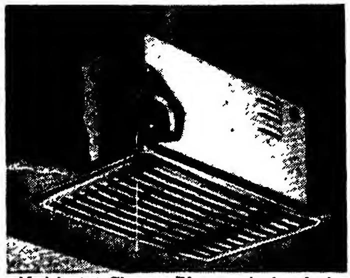
Los Angeles 37, Calif.

Representatives In All Principal Cities. Carried In Stock By Many Electrical Jobbers

CLIPPER BLOWERS



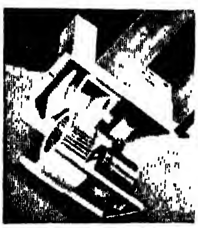
Phantom View of Model 1301 shows partition which isolates motor from air stream, increasing motor life.



Model 1401 Clipper Blower, single wheel, vertical discharge, with "dripless" grille

The Trade-Wind Clipper Blower is a small capacity centrifugal blower, primarily used for exhaust application. While used extensively for home kitchen ventilation, it is widely used for other applications. Many thousands of these units have been in service ten years or more in school toilets, therapy and treatment rooms in hospitals and clinics, dental laboratories, ticket booths, offices, x-ray and photographic dark rooms, and hundreds of kindred desirable uses.

This is a "Unit Package" assembly, complete with ceiling grille and proper electrical connections. It is Underwriters approved and listed and carries a (prorata) five year guarantee, made possible by the use of the highest quality electrical motor component. The installation requires only the application of a discharge duct of the same dimension as the outlet collar and the optional addition of an automatic shutter for the end of the duct.



Motor and blower assembly easily removed without tools.

The assembly is inherently quiet and is installed rigidly in the structure without need for resilient mountings or flexible duct connections. Variable speed controls are available on single wheel models for the purpose of controlling variable air capacity. The blower is normally quiet at its maximum speed. The motor and wheel unit is removed easily without tools through the ceiling inlet opening. The patented construction of the Clipper entirely isolates the motor from the air stream. This exclusive feature adds to the service life of the motor and keeps the motor free from contaminated air. **Write for bulletin on Type K Super Clipper for installation in cabinets over the stove.**

Two-speed control unit fits standard single switch box.

Automatic shutters available for single and double blowers.

Selection Chart and Specifications
CLIPPER BLOWERS—110 Volt, 60/50 Cycles, A.C.

Cat. No.	Description	Type Blower	Net Air CFM*	Recommended Max. Room Size Cu Ft	Duct Size	Motor Watts	H.P.
1301 B Clipper	Horizontal Discharge Complete with Grille	Single Wheel	275	1500	7" x 4½"	100	1/30
1401 B Clipper	Vertical Discharge Complete with Grille	Single Wheel	275	1500	7" x 4½"	100	1/30
2501 Duplex Clipper	Horizontal or Vertical Interchangeable Discharge Complete With Grille	Two Wheels	425	2500	10½" x 5½"	90	1/20

* This rating is based on ⅛ in. S.P. (W. G.) or the approximate resistance of 30 ft of duct the same size as outlet on Blower. The Blower will operate against a maximum of 3/10 in. S.P. with a reduction of approximately 40% rated air capacity.

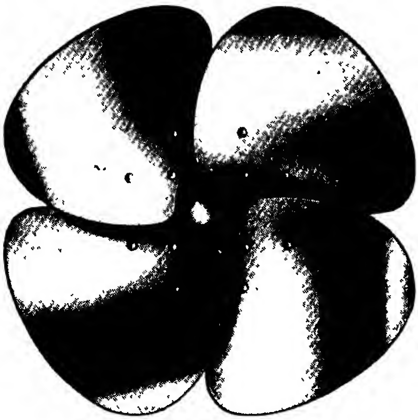
The Torrington Manufacturing Co.

50 Franklin Street, Torrington, Conn.

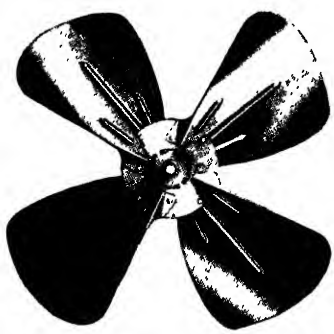
Manufacturers of Blower Wheels and Propellor Type Fan Blades.

AIRISTOCRAT Quiet Propeller Fan Blades

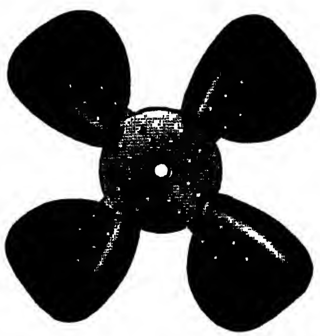
AIROTOR Blower Wheels



4-Blade Airistocrat Fan "E" Series



4-Blade Airistocrat Attic Fan "M" Series



4-Blade Airistocrat Attic Fan "B" Series

"E" Series AIRISTOCRAT Fan Blades—Outstandingly high efficiency is the chief characteristic of this truly *new* fan blade. It delivers more air for any given horsepower. Size for size it looks bigger, more powerful.

In impartial tests, eight competitive fan blades were recently compared with "E" Series blades of the proper diameter and pitch. In every case, air delivery was sharply increased. Within the same space limitations and with the same power, the "E" blade delivered as much as 28 per cent more air.

This high efficiency is the result of four years of research and development which from the beginning was devoted to bringing out a superior fan blade.

Convincing proof of the superior performance of this new fan may be found in the ***NEMA*** and ***NAFM*** tables prepared as a guide to selection. The catalog containing these tables and specifications will be mailed upon request.

Specifications: Three-blade models in 10 in., 12, 14, 16, 18 and 20 in. diameters; four-blade models 8 in., 10, 12, 14, 16, 18, and 20 in. diameters. Five pitches in most sizes. Aluminum blades, steel spider and hub. Standard finishes.

AIRISTOCRAT "M" Series Attic Fan Blades—Three outstanding features of this new design are: (1) Extremely high efficiency, which gives maximum cfm per horsepower; (2) knockdown construction which drastically lowers shipping costs; (3) quiet operation—a point of major interest to the consumer.

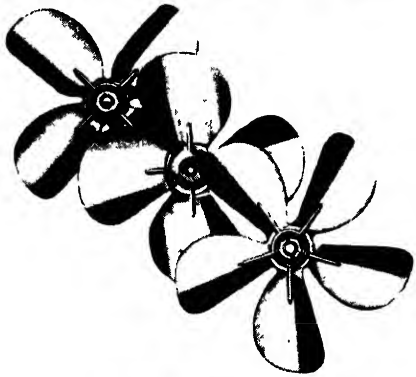
This all steel four-blade fan is manufactured for attic use exclusively, in 36 in. and 42 in. diameters, in 40 deg pitch only.

AIRISTOCRAT "B" Series Attic Fan Blades have the same proportions, proved aerodynamically correct, in all diameters. New larger center disc and heavier spider arms increase strength and a new blade shape adds to the appearance of this carefully designed product. Available in 3, 4 or 5 blades in standard diameters 24, 30, 36, 42 and 48 in. All steel construction. Available in the following finishes: 1. Plain. 2. All one color lacquer.

4-Blade Airistocrat Pressure Fan "U" Series

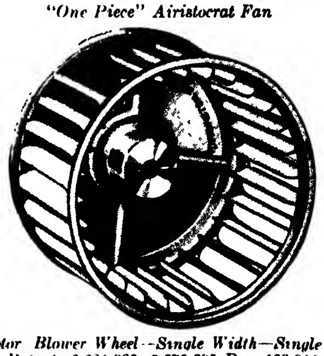
Pressure "U" Series—Four blade models of steel designed for pressure operation. Sizes 20 in., 22, 24, 26, 28 and 30 in. diameters.

"One-Piece" Airistocrat Fan Blades—Exceptionally rigid models blanked from one piece of metal. Made in both steel and aluminum. Sizes 3 in., 4, 4½, 5, 5½, 6, 6½, 8, 9, 10, 12 and 16 in. diameters, all four blades; also 5½, 7, 8, 9, 10 in. 5-blade. Available in the following finishes: 1. Plain. 2. Lacquered. 3. Zinc or cadmium plated (steel only).



"One Piece" Airistocrat Fan

Torrington Airotor Blower Wheels are light, sturdy and inexpensive—incorporate new principles of design and construction, which insure rigidity and concentricity. **Single Width—Single Inlet** wheel is of simple four-piece construction. **No rivets or welds are used;** concentric rib serving as backing for blade strip is formed at same time as hub socket, insuring trueness of wheel. Rigid radial ribs prevent deflection by thrust. Three thicknesses of metal in rims make for maximum strength. Excellent for many heating and ventilating uses. Manufactured in both aluminum and steel in 1½ in., 2, 3, 3⅝, 4½, 5, 6, 7½, 9 and 10½ in. diameters. Clockwise or counterclockwise rotation. Same sizes available in DA type double width, double inlet wheels.



Airotor Blower Wheel—Single Width—Single Inlet
Patents 2,231,062; 2,272,695 Des 126,043

Torrington Airotor Blower Wheel—Double Inlet—Spider End Plates. Has blades punched and formed in a single strip, rigidly held by flanged single piece end rings. Hubs are rigidly mounted by peening. Wheels of 2½ in., 3⅝ in., and 10½ in. diameter are available at present. Additional sizes now being developed.

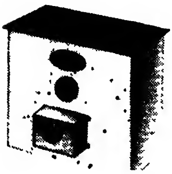
Airotor Blower Wheel Double Inlet—Spider End Plates

Utility Appliance Corporation

FORMERLY UTILITY FAN CORPORATION

4851 S. Alameda St., **Los Angeles 11, Calif.**

Utility Gas-Fired Heating Equipment, Evaporative Coolers and Blowers

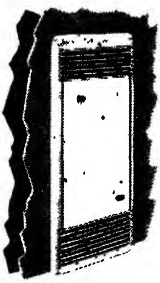


FLOOR FURNACES

Dual and flat register models . . . all-welded construction . . . vented . . . manual or thermostat control . . . draft diverter . . . die-stamped steel grille.

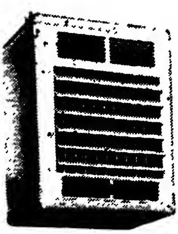
FORCED AIR FURNACES

Compact design permits use in basement or closet . . . dynamically-balanced blower rubber mounted . . . Fiberglas Dustop filters . . . heavy gauge steel, die-formed and welded . . . baked enamel finish.



UTILITY WALL HEATERS

Delivered ready for installation, locked to 2 in. x 4 in. mounting studs . . . fits standard 4 in. stud wall without added furring. Single and dual discharge models. Thermostat or 3-position manual control. Vented.



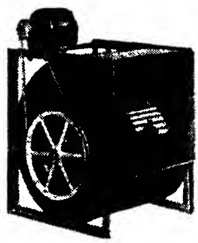
UNIT HEATERS

Suspended type. Burner, heat exchanger, draft diverter, motor, fan and all other parts self-contained in enameled steel cabinet. Adjustable grilles. 90,000 and 150,000 Btu models.

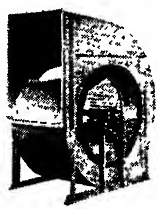
EVAPORATIVE AIR COOLERS

Comfort cooling . . . residential, commercial, industrial . . . dynamically balanced centrifugal blower . . . uniform water distribution to "No-Sag" cooling pads.

STANDARD BLOWERS

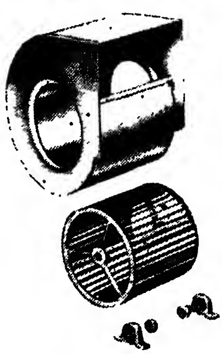


Dynamically balanced, multiple-vane centrifugal blowers. Four-side angle iron frame increases rigidity, eliminates vibration . . . permits installation with any of four discharge positions.



HEAVY DUTY BLOWERS

For increased air deliveries at higher static pressures. Dynamically balanced. Rigid construction. Single and double width.



BLOWER KITS

Consist of scroll, wheel and bearings. 9 in. through 26 in. wheel sizes. DIDW, SISW, DISW arrangements in all sizes. DI-¾W in 10 in. and 12 in. sizes. Pulleys and motor mounts also available.

Utility Blowers and Coolers are tested in accordance with the A.S.H.V.E. Code.

Utility heating appliances are *A.G.A.* approved.

Write for complete information, catalogs and price.

Western Blower Company

Founded in 1908

Manufacturers of Heating, Cooling and Ventilating Equipment for Public Building, Industrial, Commercial and Residential applications.

Main Office and Plant
1800 Airport Way
Seattle 4, Washington

Sales Offices in the principal cities West of Rocky Mountains.

TURBINE MULTIBLADE FANS—forward curved blade, Type TR, for heating, ventilating, drying, mechanical draft, etc. Bulletin No. 31.

TURBINE STREAMLINED FANS—backward curved blade, Type S, with non-overloading horsepower characteristics, for same general purposes as Multiblade Fans. Bulletin No. 30.

TURBINE ELECTRIC VENTILATING UNITS—direct connected, and **PULLEY DRIVEN UTILITY SETS**—V-belt driven. For general utility duct ventilating systems. Bulletin No. 31.

WESTERN FURNACE FANS and BOOSTER FANS—for forced circulation of air in modern home heating and light ventilating duty. Bulletin No. 60.

RB VOLUME AND PRESSURE FANS—of the radial blade type, either direct connected or belt driven for varied ventilating and conveying applications. Bulletin No. 39.

SLOW SPEED PLANING MILL EXHAUSTERS—single or double for shaving exhaust systems. Bulletin 32-3.

MILL EXHAUSTERS—motor driven, furnished single or double to suit motor speeds. Bulletin No. 32-3.

SPIROVANE PROPELLER FANS—furnished either direct connected or V-belt driven, for commercial or industrial ventilation. Bulletin No. 50.

WESTERN UNIT HEATERS—vertical and horizontal, for general heating and drying applications. Bulletin No. 53.

VOLUME HEATERS—with one or more centrifugal fans for heating, ventilating and air conditioning, available in vertical and horizontal cabinet units with or without filters. Bulletin No. 54.

OLYMPIC Heat Exchangers, Converters, Side Arm Heaters, Immersion Heaters, Oil Heaters, and Condensate Coolers.

AIR WASHERS—for cleaning, cooling, humidifying and dehumidifying.

Bulletins as listed above furnished upon request.

Westinghouse Electric Corporation
Sturtevant Division

Air Conditioning, Heating, Ventilating, Dust Control and Fume Removal Equipment, Electronic Air Cleaners, Compressors, Mechanical Draft Equipment

Hyde Park **Boston 36, Mass.**

Westinghouse-Sturtevant Division Manufactures a complete line of air conditioning, heating, ventilating and air handling equipment. Shown here are selections from the complete line. Individual product catalogs are available where more detailed information is required on any item.

Sturtevant Puts Air to Work

Equipment designed to "Put Air to Work" is available for all commercial and industrial buildings, to improve comfort and efficiency and to provide industrial process air conditioning. Applications range from the small ventilation and air conditioning systems to the large specially engineered installations.

Engineering Assistance

Sturtevant offers unbiased engineering assistance in the selection of the correct product for the specific application.

Many years of experience in the design and manufacture of these products combined with the benefits of close co-operation with consulting architects and engineers, assure you of reliable recommendations.

Offices and Distributors are located in all principal cities. Refer to your local classified telephone directory.

REFRIGERANT COMPRESSORS

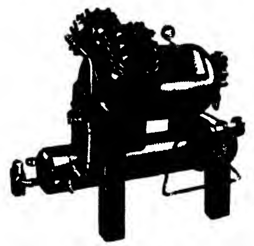
Hermetically-Sealed, Freon 12, Type CLS refrigerant compressors are designed for air conditioning and industrial refrigeration applications requiring suction evaporating temperatures ranging from 10°F to 50°F.

Westinghouse Special Features include:

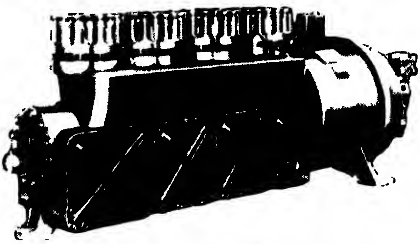
1. Shaft seals eliminated.
2. Direct drive—no belts, pulleys and couplings.
3. Sealed-in mechanism.
4. Refrigerant-cooled motor.
5. Compact size and light weight.

The hermetically-sealed compressors are reciprocating multi-cylinder, in-line type in sizes 7½ to and including 60 hp; 90 deg V-type on the 2, 3, 5, 75 and 100 hp sizes. They are protected by manually reset high and low pressure switch. Driving motors are polyphase, refrigerant-cooled induction type. Motors are protected by thermal overload protection in the De-ion magnetic linestarters.

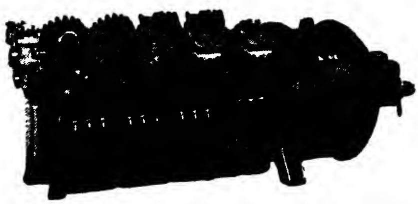
Mounting supports are available to permit mounting the compressor and suitable Westinghouse water-cooled condenser as a unit. Type CLS compressors are made in 12 sizes ranging in nominal capacities of from 2 to 100 tons refrigeration effect.



Two Cylinder V-Type Compressor Type CLS 110/188



Eight-Cylinder In-Line Type Compressor Type CLS-1320

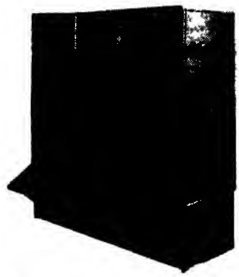


Sixteen-Cylinder V-Type Compressor Type CLS-2250

Westinghouse—Sturtevant Division

EVAPORATIVE CONDENSERS

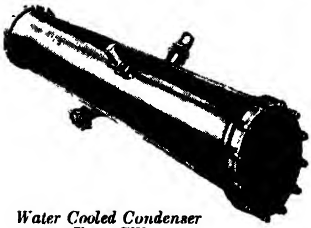
Where water is scarce or expensive, or its use or disposal restricted, the "Aquamiser" provides savings in water consumption. Westinghouse Aquamisers are available in sizes to give a range of capacities from approximately 5 tons to 100 tons each, net refrigeration effect.



"Aquamiser" Evaporative Condenser Type EVA

WATER COOLED CONDENSERS

Where sufficient water is available from city water supply or cooling tower the water-cooled condenser is economical and efficient. To economically match refrigeration load requirements, Westinghouse water-cooled condensers are available in 14 sizes, ranging in nominal capacities from 2 to 100 tons net refrigeration effect.



Water Cooled Condenser Type CW

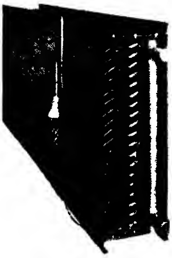
WATER COOLERS

For the chilling of water for distribution to a number of cooling coils, for surface dehumidifiers and for use in industrial processes, these efficient water coolers range in size from 5 to 110 tons net refrigeration effect. The cooler, located close by the compressor, eliminates long refrigerant piping.

Water Cooler Type LC

HEAT TRANSFER SURFACES

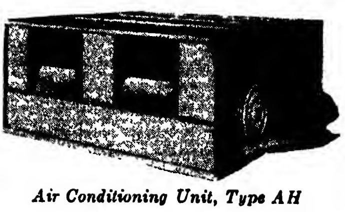
Direct expansion coils with patented refrigerant distributor, chilled water coils, steam and hot water coils are available in a great many sizes for all types of systems.



Evaporator Coil Direct Expansion Type EA

AIR HANDLING UNITS

Two types are available—a horizontal type for ceiling suspension and a vertical type for floor installation. In the horizontal and vertical types, year-round air conditioning is accomplished by the installation of both cooling and heating coils for humidification and cooling or heating as required. Refrigerant or chilled water is supplied from an external source.



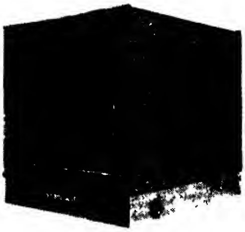
Air Conditioning Unit, Type AH

Westinghouse—Sturtevant Division

AIR WASHERS

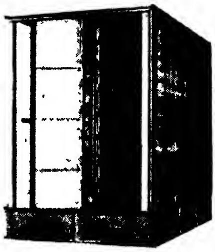
Three types are available—Types S and H, with one bank of nozzles, used primarily for humidifying; Type C, with two banks of nozzles, used for evaporative cooling, humidifying and dehumidifying.

Air Washer Type C



FILTER-WASHERS

Self-cleaning filter-washers employ a bank of mineral wool, or glass fiber filter pads, in conjunction with water sprays to clean and humidify the air. Made in 72 standard sizes and to specifications.

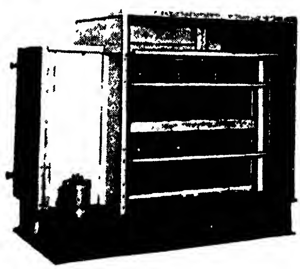


Filter Washer

SURFACE DEHUMIDIFIERS

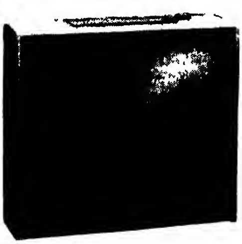
Available for either chilled water or directly expanded refrigerant, these wet surface units control accurately wet and dry bulb conditions in industrial and comfort air conditioning systems. Capacities range from 2000 to 50,000 cfm, with cooling capacities of 5 to 250 tons of refrigeration.

Unit Dehumidifier



AIR BLENDERS

Widely used for central air conditioning and ventilating systems in commercial buildings, the Air Blender mixes cool dehumidified air or heated humidified air from a central system with predetermined quantities of recirculated air to maintain room temperatures for both winter and summer, air conditioning It has no moving parts, and is equipped with a heating coil for winter operation. All units are built with 7 in. front-to-back dimensions, and in nominal lengths of from 18 in. to 60 in.



Air Blender

UNIT CONDITIONERS

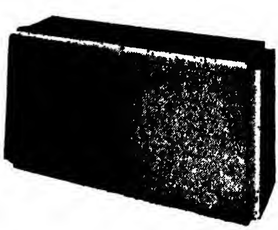
Unit Conditioners, Type UC, provide year 'round air conditioning-cooling-dehumidifying to individual room areas in such structures as hotels, apartments, office buildings and hospitals.

For cooling or heating, water is supplied at the required temperature from a central system. Air circulation and air cleaning is provided by the unit throughout the entire year.

Unit Conditioners may be used in combinations with a conditioned, central ventilating air system to recirculate and cool the air conveyed to a room, or may be individually applied to cool and dehumidify recirculated room air.

Air capacities range from 300 to 700 cfm, Btu per hour from 5400 to 16,200.

Unit Conditioner



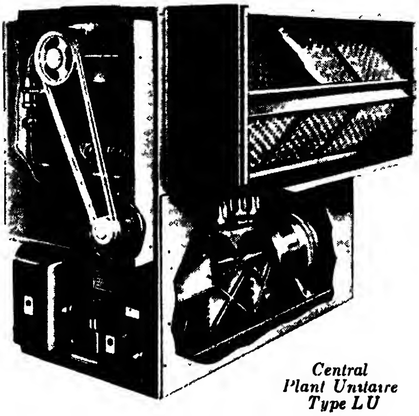
Westinghouse—Sturtevant Division

UNITAIRE®
AIR CONDITIONING UNITS

Central Plant Type Unitaires are completely self-contained, with all the component parts of an air conditioning system, designed especially for installation with supply and return air ducts. Unitaires are completely assembled, piped, refrigerant-charged, wired and adjusted at the factory. Each is given a complete operating test to assure satisfactory performance when installed. All that is required is connection of water and electrical service and attachment to the duct system.

The condensing unit is a standard Westinghouse hermetically-sealed, direct-connected compressor, with water-cooled condenser, isolated from the supporting structure by rubber anti-vibration mountings.

Central Plant Type Unitaires are available with 7½, 10, 15, 20 and 25 hp compressors.

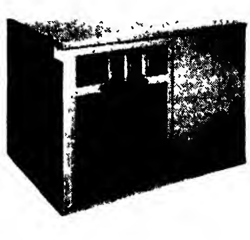


Central Plant Unitaire Type LU

Self-Contained Type Unitaires are widely used in small stores, restaurants, office suites and similar establishments. Usually they are installed in the room to be air conditioned, but in many cases they are remotely located with ducts supplying one or more spaces with air conditioned air. Available in sizes of 1, 2, 3 and 5 hp. Westinghouse Unitaires offer many important features:

1. Compact Size - Largest unit occupies less than 7 sq ft of floor space; smaller units often installed on shelves.
2. Attractive Appearance.
3. Completely Factory-Built.
4. Flexible Application.
5. Westinghouse Hermetically-Sealed Compressor.
6. Quiet Operation.

Self-Contained Unitaire Type SU

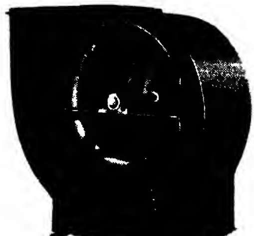


Self-Contained Unitaire Type CU

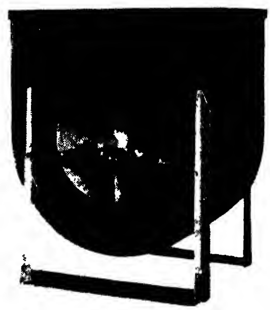
Westinghouse—Sturtevant Division

CENTRIFUGAL FANS

Silentvane ®, Design 10 Centrifugal Fans are highly efficient and aerodynamically improved over previous designs. They are made in twenty one sizes with rotors from 12¼ in. to 108¾ in. in diameter. Improved wheel with backwardly inclined blades permits medium speeds and decreased turbulence of air flow through the wheel. This results in an extremely quiet fan ideally suited for heating, ventilating and air conditioning applications. Class I Silentvane, Design 10, Fans have a 9000 fpm maximum tip speed and 3¾ in. wg total pressure. Class II Fans have 12,000 fpm maximum tip speed and 6¾ in. wg total pressure. Capacities range from 500 to 480,000 volume cfm.



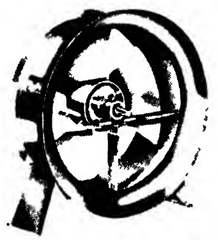
Silentvane Centrifugal Fan Design 10



Elbow Fan Design 2 AE

AXIFLO ® FANS

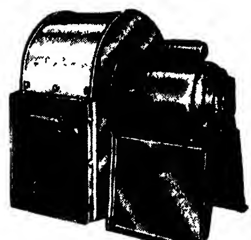
Axiflo pressure fans are of the axial flow type especially designed to meet the requirements of industrial service. They are available in both straight through or elbow types, with either 3-bladed aluminum or 8-bladed steel wheels. All designs or arrangements can be used in either vertical or horizontal position. The elbow design provides a 90 deg change in direction of air flow and offers the unquestioned advantages inherent in the fact that no motor or belts or bearings are located in the air stream. Both types are widely used in air conditioning, heating and ventilating, dust and fume removal, equipment and machinery cooling, mechanical draft for combustion, industrial drying, industrial processing, and similar applications.



Standard Axiflo Fan Design 2A Arrangement 3

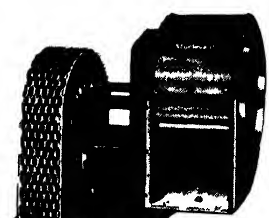
VENTILATING SETS

Rexvane ® Ventilating Sets, with radial blade wheels designed for medium speed operation, are suitable for ventilating, fume exhausting and air conditioning. Rexvane Ventilating Sets are made in 7 sizes, wheel diameters from 6 in. to 17⅛ in. capacities to ¾ in. water gage static pressure, and air delivery to 4000 cfm.



Rexvane Ventilating Set

V-Belt Ventilating Sets are completely self-contained units consisting of Multivane fan and motor with V-belt drive. These sets are particularly suitable for a wide variety of heating, ventilating and air conditioning applications. Wheel diameters from 12 in. to 24 in., and air deliveries from 930 to 11,000 cfm.

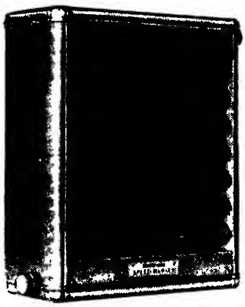


V-Belt Ventilating Set

Westinghouse—Sturtevant Division

SPEEDHEATERS ®

Sturtevant Speedheaters, Design 14, are suspended type unit heaters adaptable to commercial and industrial installations to provide an even flow of heat over a large area. They are available in capacities of 25,800 to 300,500 Btu, and 398 to 5200 cfm. Attractive modern design makes Speedheaters suitable wherever appearance is important. Their quiet, efficient operation fits them ideally for locations where economical, thorough speed heating is desired.

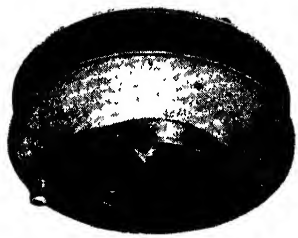


Horizontal Speedheater Design 14

DOWNBLAST SPEEDHEATERS

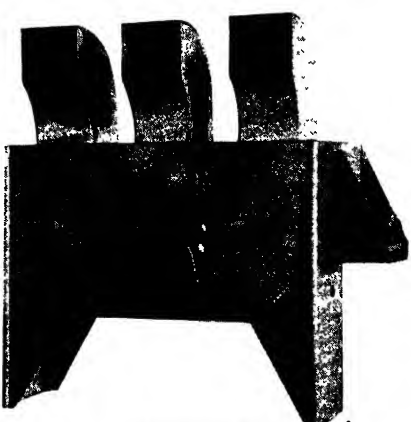
Downblast Speedheaters are suspended type unit heaters for use with steam or hot water systems. They are designed for installation near the ceiling and are available in capacities of 40,000 to 400,000 Btu, and 695 to 5800 cfm. Downblast Heaters project the heated air downward to the working level, resulting in outstanding heating performance in buildings with high ceilings, or wherever it is necessary to project heated air downward between storage bins and other obstructions.

Downblast Speedheater Design 12



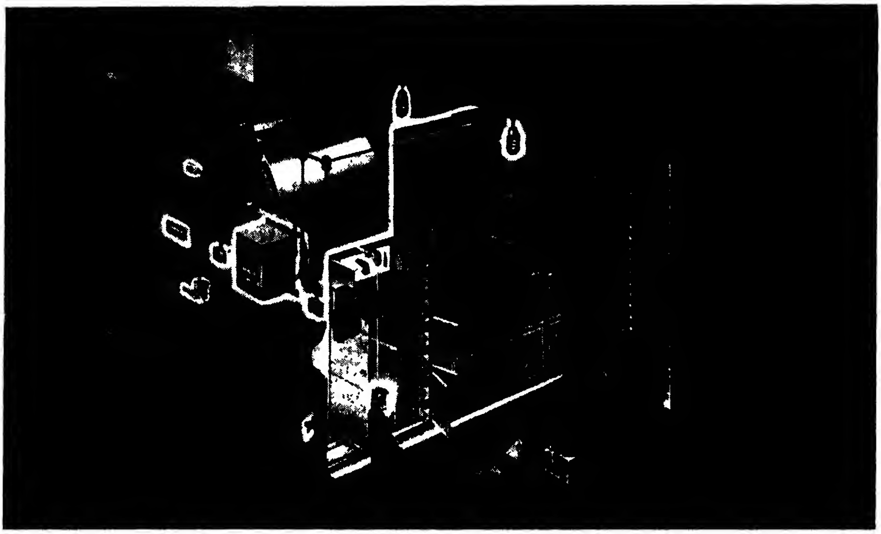
MULTIVANE ® HEATERS

Multivane Heaters are large capacity type for large area, severe heating jobs. They can be installed on floors, fastened to walls, hung from ceilings. Air motivation mechanism consists of multiple slow speed Multivane Fans, mounted on a common shaft—resulting in large volumes of air handled at low outlet velocity, low horsepower consumption, and quiet operation. Capacities range from 157,200 to 1,191,000 Btu, and 2100 to 18,000 cfm.



Multivane Heater

Westinghouse—Sturtevant Division



PRECIPITRON ®

Westinghouse Precipitron—the electronic air cleaner—removes harmful, microscopic dirt, dust, smoke, soot, and other solid particles from normal air. Precipitron is installed in heating, ventilating or air conditioning systems as illustrated above. Typical installations include department stores, banks, office buildings, restaurants, theaters, hospitals, libraries, power plants, textile mills, etc.

SPECIAL FEATURES

Smallest Particles Removed—Precipitron removes airborne particles of submicroscopic sizes. Air cleaning efficiency is 90 per cent—proven by a Blackness Test adopted by the National Bureau of Standards.

No Moving Parts—Only stationary parts are incorporated into Precipitron design and construction. Operation is quiet, maintenance negligible as there are no moving parts to wear out and replace.

Lightweight Aluminum Construction—Components are easily assembled and handled. Aluminum construction eliminates corrosion and its resultant hazards.

Complete Accessibility—All operating parts are completely accessible, thereby simplifying maintenance and reducing washing time.

OPERATION

Basically, Precipitron consists of three major elements: (1) ionizers, (2) collector cells, and (3) power packs.

Air and entrained contaminants are first passed through the ionizer. The ionizer may be denoted as the dust and dirt "charger," since it is here that the airborne particles receive a positive electrical charge which prepares them for eventual collection. Particles are charged as they pass through a strong electrostatic field set up between the ionizer components by high voltage furnished by the power pack.

Charged particles then pass between plates of the "collector" cells. Within the cell an electrostatic field exists between each two adjacent plates. This field is set up by charging one plate with high voltage from the power pack and electrically grounding the adjacent plate. The electrostatic field forces the positively charged airborne particles to collect on the grounded plate. An adhesive coating on the plate surfaces firmly holds the particles.

The cleaned air then passes on through the heating, ventilating or air conditioning system.

Westinghouse—Sturtevant Division

Washing—Over a period of time, collector cell plates reach their dirt holding capacity. When this point is reached, the plates are cleaned by spraying with water from an ordinary garden hose. After each washing, plates must be resprayed with a thin coating of adhesive. Convenient applicators are provided.

Blackness Test Proves Efficiency

The high cleaning efficiency of Precipitron is shown by the *U. S. Bureau of Standards* Blackness Test. This test measures the cleaning efficiency by comparing the relative dirt contents of the air before and after the air cleaner. The amount of dirt is indicated by passing samples of the air through chemical filter paper and comparing the blackness spots.

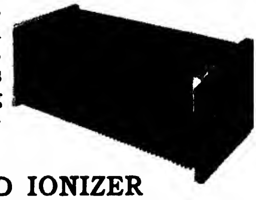
By Blackness Test, Precipitron shows cleaning efficiencies as high as 90%.

The Blackness Test samples illustrated below decisively show the superiority of Precipitron Air cleaning.

Uncleaned Air — *Air Cleaned with Conventional Filter* — *Air Cleaned with Precipitron*

CELLS

Constructed entirely of aluminum, except for the insulators. Precipitron cells are corrosion resistant and light in weight. Unit erection and maintenance are relatively easy as one man can lift a cell into position. Clean-cut structural design provides unobstructed plate surfaces which reduces cell washing time to a mini-



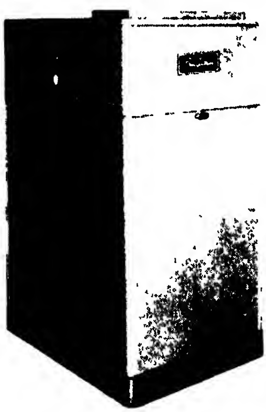
FRAME AND IONIZER

Within the Precipitron frame are mounted the basic air cleaning components, the ionizer and collector cells. The ionizer is built in as an integral part of the frame, and the collector cells may be easily inserted or removed from the rear. Each frame holds two collector cells. Hinged air distribution baffles are mounted on the front of the frame. Frames may be combined into a number of tier and column arrangements. This flexibility permits installation of Precipitron where space is limited.

POWER PACKS

Precipitron power packs supply the ionizers and cells with the required d-c voltages from 115 volt a-c source. Power pack indicating panel indicates normal or abnormal operation and provides thermal overload protection.

HOME UNIT



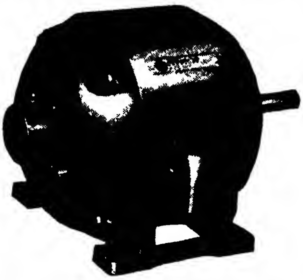
The Westinghouse Precipitron Home Unit is designed primarily for use in homes. It performs the same function, on a smaller scale, as the larger units do for commercial and industrial applications. No longer a luxury, the Westinghouse Precipitron electronic air cleaner minimizes the endless chasing of dust and dirt in the home.

It is comparable in size to a gas-fired furnace, and occupies about the same floor area. Provision is made on the unit for connection to the duct work of a forced air heating or air conditioning system. Placed in the air intake to the heating unit, the Precipitron removes more than 99.5 per cent of the dirt (by weight), and more than 90 per cent of all airborne particles.

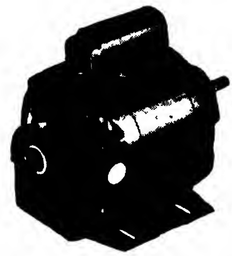
MOTORS FOR HEATING, VENTILATING, AND AIR CONDITIONING

General Electric offers a complete line of motors for compressors, fans, and pumps, from which you can select easily the motors with electrical and mechanical characteristics best adapted to your equipment. Many of the most common applications are listed below. Information on other motors—vertical, enclosed, etc., with various electrical and mechanical modifications—can be obtained at a G-E office near you.

For additional information, ask for Motor Catalog GEA-3580.



Tri-Clad induction motor, Type K, polyphase*



Fractional-horse-power capacitator-motor, Type KC

SOME G-E MOTORS AND THEIR USES

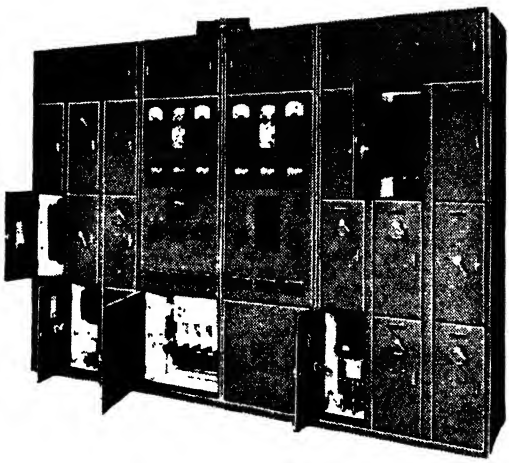
Application	Speed	Type Winding	Type	Horsepower Range	Power supply Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD	1/8—200	Direct Current
Reciprocating Pumps and Compressors		Compound	B & CD	1/8—200	
	Constant	Capacitor, Normal torque	KC	1/4—3	
		Capacitor, High-torque	KCJ	1—3	
Small Direct-connected Fans	Constant	Resistance, Split-phase	KH	1/40—1/3	Single-phase Alternating-current
		Shaded-pole	KSP		
	Constant or 3-speed	Capacitor, Low-torque	KCP	1/50—5	
Belted Fans, Centrifugal Pumps	Constant	Capacitor, Normal-torque	KC	1/4—3	
		Repulsion-induction	SCR	5—10	
Reciprocating Pumps and Compressors	Constant or Multispeed	Squirrel-cage, Normal-torque	K	1/4—1000	Polyphase, Alternating-current
		Squirrel-cage, High-torque	KG	5—200	
Pumps, Compressors, Fans	Constant or Adjustable-varying-speed	Wound-rotor	M	1/2—1000	
	Constant	Synchronous	TS	25—2000	

Types of Enclosures: Open (dripproof)—protected from falling objects or dripping liquids. Splashproof—where wetness is a factor. Totally enclosed—for complete protection. Explosion-proof—for inflammable gases. Dust-explosion-proof—for combustible dusts.

* Trade-mark reg. U. S. Pat. Off.

CONTROL FOR HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS

General Electric offers a complete line of standard manual and automatic controls for all types of motors driving compressors, fans, pumps, etc. Publications describing these items, as well as such control accessories as pressure governors, pressure switches, float switches, electrically operated valves, and indicating selsyns, are available on request. For special applications, G-E control to meet your exact requirements can be designed. Most frequently, however, the needs of the air-conditioning industry are best served by one of the many possible G-E "packaged" control combinations designat d as Cabinetrol* equipments.



Typical Cabinetrol unit, showing open motor-starter panels

The Cabinetrol system of motor control is based upon the use of standardized enclosures equipped with standard control devices. General Electric can build quickly most air-conditioning control systems—simple or complex—by properly combining standard control units and accessories into the required number of basic Cabinetrol sections. If future expansion should require further control equipment, additional Cabinetrol units can easily be added to the basic system.

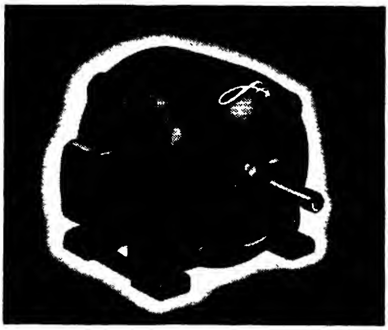
The standard enclosures used in the Cabinetrol system will mount standard motor-starting devices up to and including NEMA Size 4. Ample space is provided for incoming-line, feeder, and metering equipment. Bulletin GEA-3856 details more fully the advantages of this new system of centralized low-voltage control.

The General Electric Company will gladly assist in the solution of any electrical problem related to air conditioning.

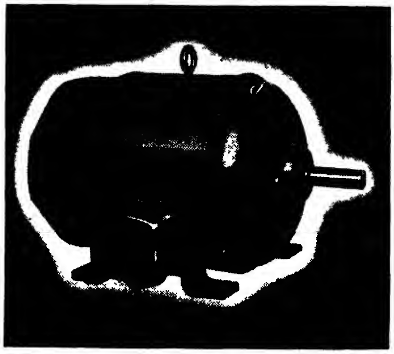
* Trade-mark reg. U. S. Pat. Off.

The Louis Allis Co.

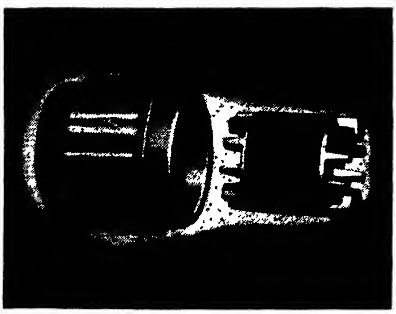
Milwaukee 7, Wisconsin



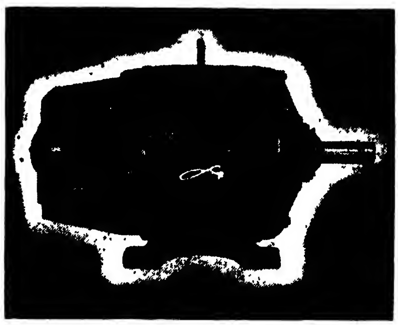
Louis Allis open, drip proof type Squirrel Cage motors (Type OG) are specially engineered and designed for applications requiring greater protection.



These Direct Current motors are correctly engineered and precision built—available in a very wide range of both electrical and mechanical modifications. The extra rugged construction of these motors assures long life of dependable performance.



The wide-spread experience of our engineering staff will be placed at your disposal to assist in powering your product with these compact efficient Shaftless motors. Available in a wide range of sizes and electrical modifications.



Wound Rotor (slip ring) motors are ruggedly constructed throughout. A wide range of speed variations are obtainable with comparatively simple control. Available with mechanical modifications for every industrial use.

We manufacture a size and type electric motor for every industrial requirement.

Wagner Electric Corporation

EST. 1891

6464 Plymouth Avenue
St. Louis 14, Mo., U.S.A.

Sales Offices in 29 Principal Cities

SINGLE-PHASE
POLYPHASE
DIRECT-CURRENT
MOTORS

For Heating, Ventilating and Air Conditioning Equipment

SINGLE-PHASE MOTORS

Repulsion-Start Induction

1/6 to 15 hp, all standard frequencies and voltages; sleeve and ball bearings; open or totally enclosed; horizontal and vertical; rigid, flange, or resilient mounted (in smaller ratings).

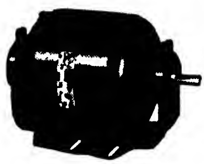
Permanent Split-Capacitor

1/20 to ¾ hp, constant speed, two-speed, or adjustable-speed, all standard frequencies and voltages; sleeve bearings; totally enclosed; round frame with rubber rings.

Capacitor-Start Induction

1/6 to ¾ hp, all standard frequencies and voltages; sleeve and ball bearings; dripproof and totally enclosed; horizontal and vertical; rigid, resilient and flange mounted.

Shaded-Pole Fan Duty



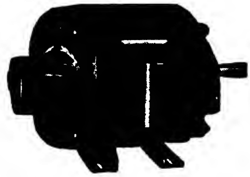
1/125, 1/80, 1/40 and 1/30 hp, 50 or 60 cycles, 115 or 230 volts; sleeve bearings; totally enclosed; rigid, round frame and resilient mountings; with or without 3-speed regulator.

POLYPHASE MOTORS

Squirrel-Cage Induction

1/6 to 400 hp, 2 and 3 phase, all standard frequencies and voltages; sleeve and ball bearings; open or totally enclosed; horizontal and vertical. Built in several electrical types.

DIRECT-CURRENT MOTORS



1/20 to 3 hp, all standard voltages, sleeve or ball bearings, dripproof frames, rigid mounted. Wagner direct-current motors in the 44 frame are shunt-wound, all others are compound wound.

Write For These Bulletins

MU-40—Lists part numbers and prices of Wagner motor repair parts, arranged for quick reference.

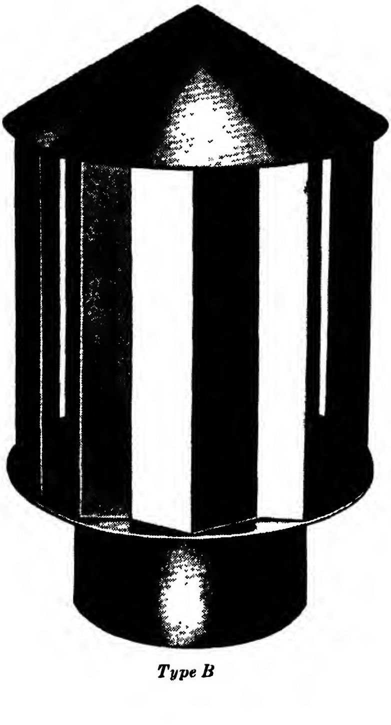
MU-185—Describes and illustrates all types of Wagner motors.

G. C. Breidert Co.

3129 San Fernando Road, **Los Angeles 41, Calif.**

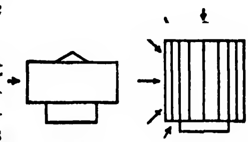
Representatives Located in Principal Cities of the U. S.

BREIDERT AIR-X-HAUSTERS
FOR ROOF VENTILATING, VENT FLUES & CHIMNEY TOPS



Type B

The Breidert Air-X-Hauster introduces a new principle in ventilator design. Because of the revolutionary, aerodynamically-correct design of the Breidert Air-X-Hauster, wind currents striking it from *any* angle are converted into a powerful suction force that rapidly exhausts stale air from the interior of the house, kitchen or building. The Breidert remains stationary, has no moving parts. *Back-drafts are eliminated where there is no interior negative* .



Old Method *Breidert Method*

The Breidert ventilator offers *certified* capacity ratings based on tests made with wind blowing at *all* angles (as shown). These high capacities were proved and certified by Smith, Emery Co., Pacific Coast branch of Pittsburgh Testing Laboratories. *Insist* on certified ratings based on directional wind tests at various vertical angles as shown in considering *any* ventilator. Breidert Air-X-Hausters were used extensively during the war on many types of combat and cargo ships, war housing, military barracks, and other government buildings. They also are widely used on all types of factories, commercial buildings, and residences.

For Kitchen Ventilation . . . The Breidert system provides a continuous, silent, effective circulation of air that exhausts heat and odors *at their source*, with no operating or maintenance expense. There are no "hang-over" cooking odors because the exhaust action of the Breidert is continuous. The neat, compact appearance of the "Type A" Breidert especially recommends it for residences.

For Vent Flue Caps . . . The Breidert does not have the defects of conventional types of caps and accessories. It eliminates the necessity for down-draft diverters, with accompanying dangers of explosion in case unburned gas accumulates or is blown into the room.

For Chimney Tops . . . By stopping all down-drafts (interior negative pressure excepted), a Breidert Air-X-Hauster on the chimney absolutely prevents the fireplace from smoking and damaging furnishings. *It provides positive "draw" regardless of wind direction!*

Write for Free Engineering Data Book . . . contains specifications and installation data, certified capacity ratings, etc. Address Dept. HV.

The Swartwout Company

18511 Euclid Avenue, **Cleveland 12, Ohio**

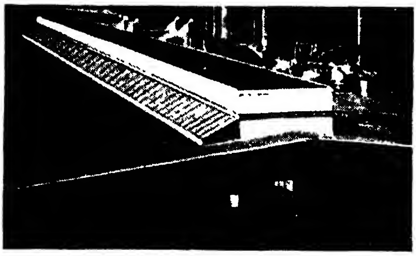
Representatives in Principal Cities

Gravity and Powered Industrial Roof Ventilators

GRAVITY ROOF VENTILATORS

Swartwout-Dexter Heat Valve

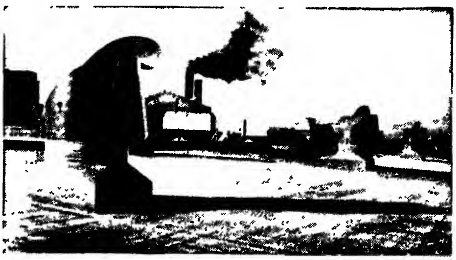
Continuous opening natural draft ventilator particularly effective for ridge of peaked roof, saw-tooth construction or skylights; adaptable to flat or slant roofs. Air takes but one turn, flows directly upward. Made in throat opening sizes 4 in., 6 in., 9 in., 12 in., 15 in., 18 in., 24 in., 30 in., 36 in. and 42 in. Ten foot lengths may be assembled continuously for any roof length. Adjustable damper.



Swartwout AIRMOVER

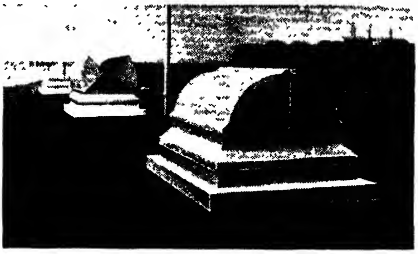
A horizontal type multiple heat valve featuring exceptionally short air travel, built in units 10 ft x 7 ft 6 in. x 32 in. high; 30 sq ft of opening per unit. Large scale ventilation achieved by use of continuous runs, or can be used as single units over heat concentration centers. Installed on curbs of wood, steel or concrete. Can be adapted to any type of roof.

POWERED ROOF VENTILATORS



Swartwout AIRJECTOR

Streamlined rotary head type roof ventilator using propeller type fan. Curved head turns with wind—outlet points away from wind to take advantage of suction effect. Made in 13 throat sizes, from 12 in. to 72 in., with wide range of capacity ratings. Adaptable to special conditions where high temperatures or fumes are a problem. Available without fan as a rotary gravity unit.



Swartwout JECT-O-VALVE

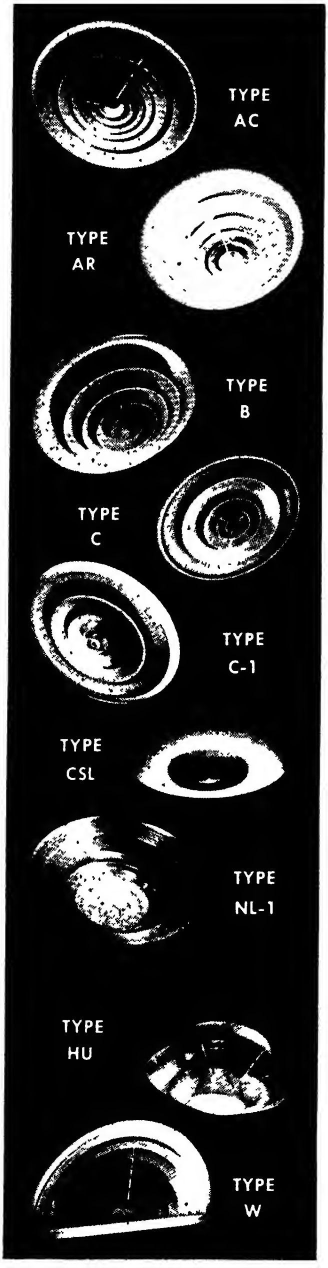
Powerful exhauster of the straight-through type, particularly effective over vats, furnaces, foundry pouring floors, etc. Air stream opens top dampers, which close when fan is off. Completely weathertight at all times. Hinged top section permits easy access for servicing. Made in 28 in., 36 in., 40 in., 44 in., and 48 in., throat size, in a variety of cfm capacities.

Swartwout Industrial Intake Louvers are available in a wide variety of sizes.

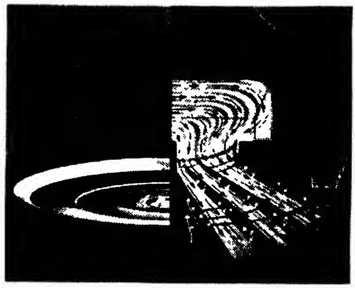
All of the above made in galvanized steel as standard; can be supplied in copper, aluminum, A.P.M. or stainless steel. See *Sweets Architectural or Engineering File* for further data, or send for complete Swartwout general catalog.

Anemostat Corporation of America

10 East 39th Street **ANEMOSTAT** Representatives in
New York 16, N. Y. **DRAFTLESS** *Aspirating* **AIR DIFFUSERS** Principal Cities



THE ANEMOSTAT PRINCIPLE



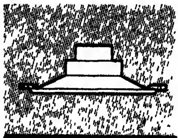
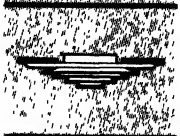
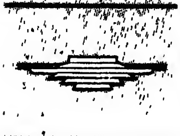
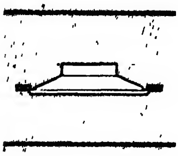
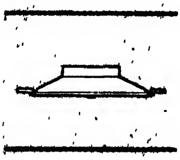
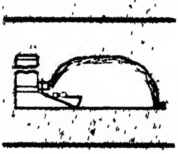
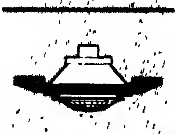
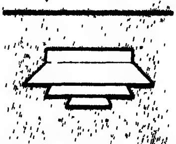
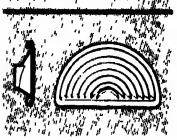
The Anemostat Air Diffuser consists of a series of conical shaped members of specific design and assembled in definite relationship to each other. Due to these exclusive and patented features, the Anemostat Air Diffuser breaks the primary air stream into a multiplicity of planes traveling in all directions, and at the same time creates an equal number of counter currents of room air traveling toward the device. This causes a quantity of room air equal to as much as 35 per cent of the supply air to be drawn into the device and mixed with the supply air before it is discharged. This important effect, which distinguishes the Anemostat Air Diffuser from all other air outlets, is known as Aspiration, and should not be confused with secondary air motion. The Anemostat Air Diffuser, moreover, permits air expansion within the device, instantly reducing velocity.

The mixture of primary and room air discharged from the Anemostat Air Diffuser entrains additional room air, which, due to the turbulence of the discharged air, is readily drawn into and mixed with the discharged air. This will continue until the movement of the discharged air has ceased.

The Anemostat Air Diffuser, therefore, thoroughly diffuses and mixes the incoming air with the room air, closely equalizing temperature and humidity throughout the room, and promptly converts the velocity energy of the incoming air into pressure energy which prevents drafts in the occupancy zone.

ANEMOSTAT Draftless Aspirating AIR DIFFUSERS

This table shows basic data only . . .
Consult complete catalog for full information.

TYPE		Basic Function	Sizes	Aspiration*	Applications: Heating	Ventilating	Cooling	Refrigeration	Neck Velocity Limits	Location
AC		Combination supply and exhaust	10 to 60	30%	Yes	Yes	Yes	No	700 to 2500	On exposed duct or flush to ceiling
AR		Supply Frequent air change with low air velocity in room	15 to 130	30%	Yes	Yes	Yes	Yes	700 to 3000	On exposed duct or flush to ceiling
B		Supply	10 to 95	35%	Yes	Yes	Yes	Yes	700 to 3000	On exposed duct or flush to ceiling
C		Supply	10 to 95	35%	Yes	Yes	Yes	No	700 to 2500	Flush to ceiling only
C-1		Supply Adjustable pattern	12.5 to 45	Up to 35%	Yes	Yes	Yes	Yes	700 to 2500	On exposed duct or flush to ceiling
CSL		Supply Combination diffuser and light	20 to 95	35%	Yes	Yes	Yes	No	700 to 1500	Combination with light cove
NL-1		Combination diffuser and light	15 to 40	30%	Yes	Yes	Yes	No	700 to 1500	On exposed duct or flush to ceiling
HU		For use with unit heaters	25 to 82.5	25%	Yes	No	No	No	700 to 2500	Projection heaters
		Use on duct work for heating and ventilating	25 to 82.5	25%	Yes	Yes	Yes Up to 10° Differential	No	700 to 2500	On exposed duct
W		Wall type supply	5 to 40	35%	Yes	Yes	Yes	No	700 to 2500	Wall or ceiling

"NO AIR CONDITIONING SYSTEM IS BETTER THAN ITS AIR DISTRIBUTION"

Air Devices, Inc.

Air Diffusers • Exhausters • Air Filters
Filter Holding Frames • Hot Water Generators

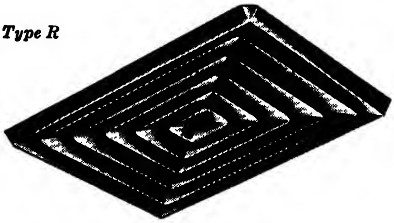
17 East 42nd St.
New York 17, N. Y.

AGITAIR PRODUCTS

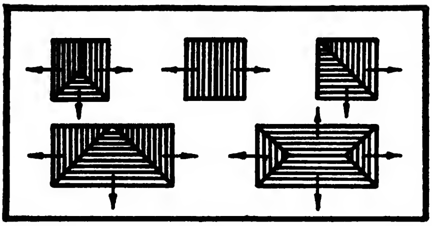
Agents in All Principal Cities

AGITAIR DIFFUSERS

Type R

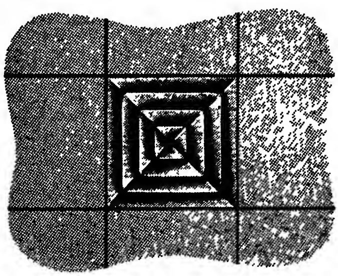


Square or Rectangular in Shape



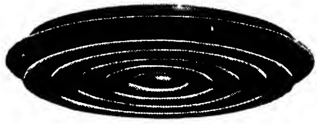
Type R AGITAIR is the only diffuser that offers the higher efficiency of the square or rectangular shape in diffusing air quietly, draftlessly and with rapid temperature equalization throughout any shaped room. Its patented construction can be assembled into patterns which discharge the required amount of air in one to four directions.

Each side delivers a quantity of air proportional to the areas served. Thus the engineer or architect can select an attractive diffuser that fits his design—rather than make his design fit the diffuser.



ACOUSTICAL CEILINGS

An improved, re-styled air diffuser known as the AGITAIR **Type RTC** square or rectangular in shape, especially designed for installation in acoustical ceilings. Made in sizes to conform to standard tile dimensions.

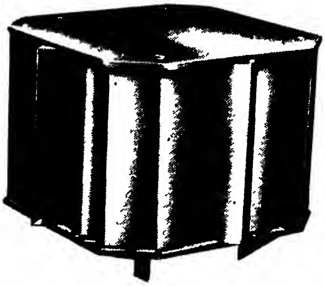


Type A

Type CSF with Type B Damper

Circular AGITAIRS combine beautiful design with finest operating features to give rapid temperature equalization and draftless diffusion of air. Model CSF is quiet, easy to install. Pressure losses at minimum. Type A diffusers are smaller, weigh less, are easy to install. Type CM is for marine use. In all sizes for all types of mounting and with lighting combinations. Dampers are provided where needed.

The AGITAIR Diffuser Data Book, available to architects and engineers, will help you design and install air distribution systems. Consult our engineers.



AGITAIR WIND-ACTUATED EXHAUSTERS

Provide proper ventilation regardless of wind direction, and with positive elimination of down-draft. Functions at peak efficiency at average low wind velocities. Will not restrict the flow of air or gases when there is no movement of outdoor air across the head.

Fan-equipped units also for higher ratings.

The Auer Register Co.

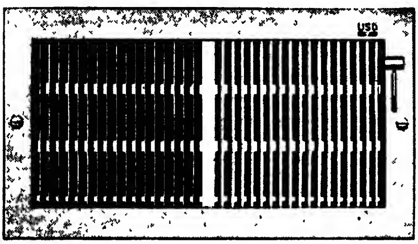
3608 Payne Avenue, **Cleveland 14, Ohio**

Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Metal Grilles for Radiator Enclosure, Ventilation, Concealment

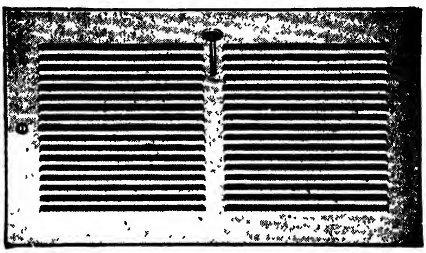
AIR CONDITIONING REGISTER AND GRILLES

The Auer line of registers and grilles for all warm air heating and air conditioning systems is complete and modern, with a wide choice of styles for every purpose. Only a few are shown here. For gravity systems, the Auer Heat-Rite is a popular, streamline model adjustable for up-or-down flow. For extra strength, also for floor furnaces, use Dura-Bilt Registers and intake, with cross-bars set edgewise and all joints inter-locked and mortised. All Auer models are designed with due regard for air capacity, and supplied in all standard sizes and finishes. Latest Auer Register Book showing entire line sent on request.

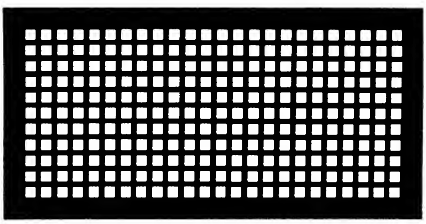
Auer flat stamped metal grilles are made in many designs, all sizes, for all purposes. Furnished in steel (or stainless), aluminum, brass, or bronze, and finished in prime coat, special finishes or platings. Send us your specifications. Ask for special Auer Grille Catalog "G", with full scale details and tables of openings and free areas.



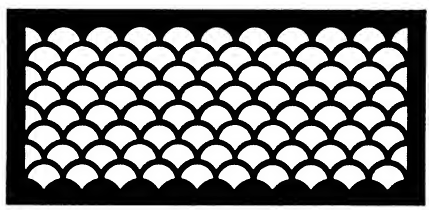
Airo-Flex No. 4432 Register—Multi-louvres adjustable up, straight or down. Grille bars adjustable for right or left flow. Grille to match.



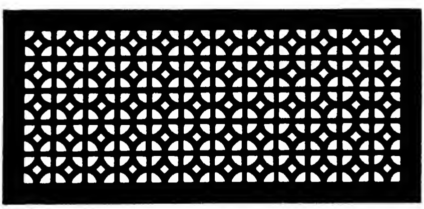
Airo-Flex No. 7032 Register—Grille bars set to direct air downward at 22½ deg, but adjustable for other angles. Single louvre. Grille to match.



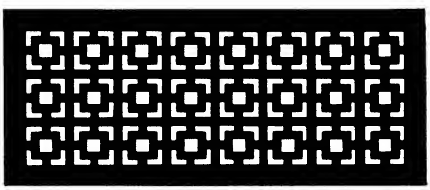
5A—Square Lattice Grille



32A—Shell Grille



4A—Moorish Grille



35A—Square Link Grille

Barber-Colman Company

Rockford, Illinois

ENGINEERED AIR DISTRIBUTION OUTLETS

VENTURI-FLO
Overhead Air Diffusers

Venturi-Flo overhead air diffusers have flow characteristics similar to those of the well known fluid-flow measuring device—the Venturi Meter. The relationship between the neck area of the unit proper and the Venturi-Flo throat area is so proportioned as to create a slight back pressure in the neck at all times, thereby automatically insuring uniform distribution around the entire periphery of the unit.

Three types of overhead diffusers are available, the recessed, the surface, and the thermostatically controlled types. A wide range of sizes permits handling air volumes up to 15,000 cfm per unit. Fittings for attaching any standard light fixtures to the outlets may be obtained for all designs. The surface and recessed types can also be furnished as combination supply and exhaust units and with adjustable dampers.

Uni-Flo Grilles and Registers are especially designed for air conditioning applications. They are engineered and prefabricated with directional flow fins for each individual application. Proper air distribution is assured and the necessity for adjustment after installation obviated.

Uni-Flo Grilles and Registers can be furnished in a variety of shapes and sizes for plain and curved surfaces.

Registers are similar in construction to grilles but with the addition of spring loaded positive closing fan or key-operated dampers.

Electroplated Finishes: Gunmetal, brushed bronze, plain zinc, and satin copper; also available in gray prime coat and satin aluminum.

Uni-Flo Air Distribution Accessories: Include the Volocitrol and the Airturn. The Airturn is a scientifically designed and highly efficient air turning member. The aero-dynamically correct vanes reduce losses caused by eddies, reverse air flow and low pressure areas at the turn to a minimum. Airturns are prefabricated and are available in units of one piece construction in sizes up to and including 48 in. x 48 in.

The Volocitrol has been designed to noiselessly provide positive and adjustable control of air volume, pressure and distribution across a supply outlet. They are available in two models, one for use with side wall and the other for overhead systems of air distribution.

Venturi-Flo-Recessed Type

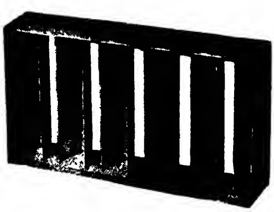
Venturi-Flo-Surface Type

Venturi-Flo-Thermostatically Controlled Type

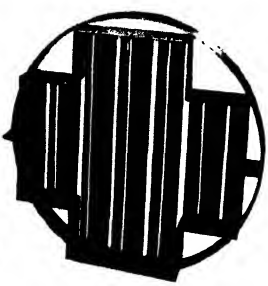
Uni-Flo Grille Model "M"

Uni-Flo Register Construction

Airturn



Volocitrol

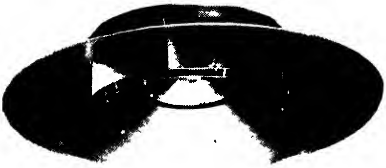


Circular Volocitrol

Charles Demuth & Sons, Inc.

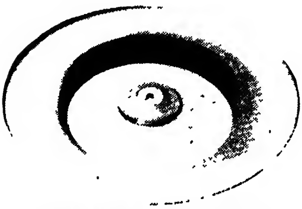
Mineola, N. Y.

Air Distributors—Oil-Fired Conditioners



For exposed duct work & general applications. Type (I)

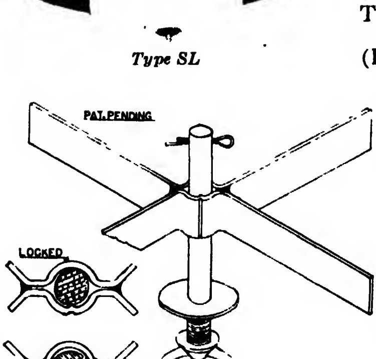
Type (S) General Applications



Flush ceiling model Type (F)

Typical Illustration

Type SL



The Demuth Air Distributor is a ceiling air diffuser of patented principles and design. One of the many outstanding characteristics of the Demuth Air Distributors which makes it different, is that it has as an integral part of the device air equalizing vanes. These equalizing vanes are fastened to a deflecting cone to form a nozzle whereby a series of air currents are equally divided, and discharged throughout the 360 deg opening in a circulatory motion.

A second deflecting cone extends beyond the edge of the principles cone, and spaced below, forms a tangential injection nozzle. Also, a vacuum is created at the outside end of the vane. Both of these actions together with action of the air streams discharged from the nozzle creates a vortex directly below the device. Consequently, the Demuth diffuser distributes a series of turbulent air streams resulting in a noiseless and positive 360 deg circulation and recirculation of the air without draft. Turbulent air streams, plus injection, provides complete diffusion and uniform temperature of the air. Manufactured in all sizes, Aluminum Polished or Satin finish, Steel painted, and Stainless Steel.

Typical Illustration—All models having the curved equalizing vanes as an integral part of the Demuth Air Distributor. Note spider collar ease of installation. Volume control types with collars & draw bolt installation.

Demuth Air Distributor with lighting fixture of Demuth design. Type (SL), also supplied with Types (F) & (I), as are types (FL) & (IL).

Pendant fixtures can be hung from types (S), (I), & (F).

The adjustment device used to control the volume of air is an entirely new Demuth principle—that is an integral part of the Distributor, whereby the equalizing vane cone assembly can be moved from completely closed (no cfm) to full open. This is accomplished by a ¼ turn, either in the right or the left hand direction, then locked in place by the same action. The other patented principles and design of the Demuth Air Distributor are not altered in any manner whatsoever, therefore retaining all the features that assure good distribution of air. The Demuth Volume Control types come with (I)—(F) & (S) Units, and are types (IVC), (FVC), & (SVC)

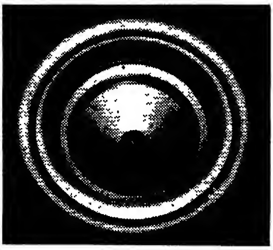
W. B. CONNOR ENGINEERING CORP.

**114 East 32nd Street,
New York 16, N. Y.**

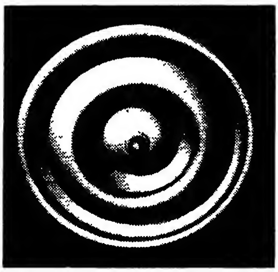
**Representatives in
All Principal Cities**

In Canada: Douglas Engineering Co., Ltd., Montreal, P. Q.

KNO-DRAFT *Adjustable* AIR DIFFUSERS



Type KDA for supply air. Aluminum, sizes 4 to 36 in. neck dia. Capacities 50 to 15,000 cfm per unit.



Type SRD for combination supply and return air. Sizes 6 to 24 in. supply neck dia. Capacities 50 to 2,500 cfm per unit, return neck area 75% of supply.

Diffuser and Cone separated

Diffuser and Cone assembled

CONTROLLED AIR DIFFUSION

Kno-Draft Adjustable Air Diffusers are designed to give accurate control of air distribution, plus installation and operation economies. With air direction and air volume adjustments on each diffuser, "custom-made" air patterns can be created which will insure draftless diffusion and equalized temperatures for comfort conditioning or specific patterns for industrial processes.

Installation, balancing and inspection are fast because of features like the Type HD quick-opening set-lock assembly, the self-contained inner unit and the sleeve-type damper.

System design problems are eased because Kno-Draft Diffusers are adjustable *after installation*. The often difficult and hazardous job of figuring everything about the air movement in advance is eliminated. And the air pattern in an area can be changed with the seasons or when processes, people or partitions are relocated.

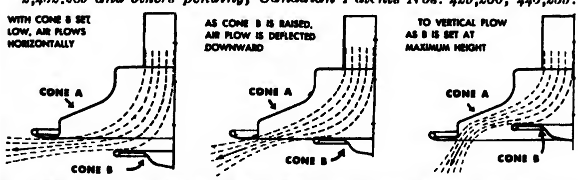
Kno-Draft Diffusers are geometrically proportional, size for size, insuring like resistance at like neck velocity for any size—a considerable advantage when selecting various size diffusers for a common system.

Designed for high or low ceilings or attachments to exposed ducts, these diffusers will effectively distribute large volumes of air and pre-mix room and supply air. They permit the use of high duct velocities—resulting in smaller ducts and lower costs. Duct designs are simplified.

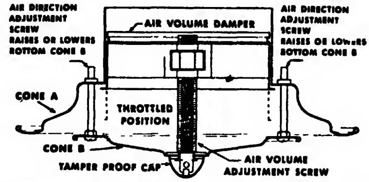
The simple, attractive design of the Kno-Draft Diffusers enables them to blend with either period or modern interiors. In their original aluminum, they create an interesting and unobtrusive decorative accent. Painted to match the ceiling, they become self-effacing.

Anti-Smudge Cone: Where exceptionally sooty or dusty air conditions are expected or where rough-textured ceilings are employed, the use of this accessory cone is recommended. It furnishes the additional control needed to provide the precise air separation which inhibits smudging.

Kno-Draft Diffusers are covered by U.S. Patents Nos. 2,365,867; 2,369,119; 2,432,289 and others pending; Canadian Patents Nos. 429,206; 443,235.



Any angle of air discharge needed to suit ceiling heights and heating, ventilating or cooling air patterns can be obtained by raising or lowering bottom Cone B.



Type D Air Volume Control operates independently of the air directional adjustment. It varies only the quantity, not the characteristic of the air distribution. It consists of a cylindrical, sliding, sleeve-type damper connected by a specially designed spider to a centrally operated screw. The shank of the screw extends through the lower cone of the diffuser and is concealed by a tamper-proof cap. With this damper on each unit, a series of diffusers can be quickly balanced.

Kno-Draft Features Speed Installation

The development of the self-contained, removable inner assembly alone (see Fig. 1) reduced installation time as much as 50 per cent. The addition of the Type HD set-lock assembly (see Fig. 2) reduced the time for that part of the installation to a matter of minutes. No tools are required. The B cone or inner element of the diffuser is secured to the combined suspension and adjustment screws by a springloaded catch which is kept in compression by a slotted washer. The holes in B cone pass over the bolt heads. All that is necessary is to press up on B cone and insert or remove the slotted washers. (See Fig. 3.) Even where ceilings already exist, the outer cone is easily attached to a duct or collar.

The air direction adjustment is also accomplished quickly. All that is needed is a screwdriver to adjust the suspension screws for any angle of air discharge from horizontal to vertical. (See Fig. 4.)

System balancing is fast and simple. The single annular air stream permits immediate and accurate velometer reading. (See Fig. 5.) Desired air supply ratios may be rapidly obtained by adjusting the volume control dampers. A twist of the wrist regulates the volume instantly. (See Fig. 6.)

Nation-wide Sales and Engineering Service

The W. B. Connor Engineering Corp. maintains a research laboratory with a staff of trained specialists and district representatives in leading cities. Their services are at the disposal of consulting engineers, architects, air conditioning dealers and plant engineers. They can assist you in getting the best possible performance from your air conditioning system by creating custom-made air patterns which will thoroughly mix room and supply air, eliminate drafts and maintain uniform temperature throughout an area.

Free Handbook on Air Diffusion

It contains the latest engineering data on air diffusion and is profusely illustrated with charts, photographs, sketches and dimension prints that simplify the selection, application, location, assembly, erection, testing and adjusting of Kno-Draft Adjustable Air Diffusers. It is designed to help you get top efficiency from an air conditioning system by creating "custom-made" air distribution patterns.

For your FREE copy, please write Dept. Y-29.

Fig. 1 Self-Contained Inner Unit

Fig. 2 Type HD Set-Lock Assembly

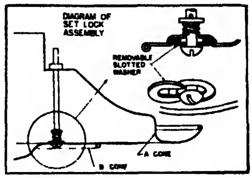


Fig. 3 Now Standard Equipment

Fig. 4 Air Direction Adjustment

Fig. 5 Balancing

Fig. 6 Air Volume Adjustment

(See pages 1110, 1111, 1112, and 1113 for data on Dorex Air Recovery and Air Purification Equipment.)

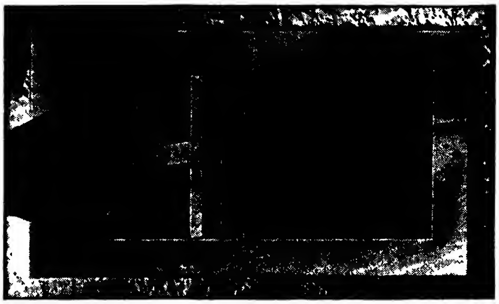
Hart & Cooley Manufacturing Co.

Established 1901

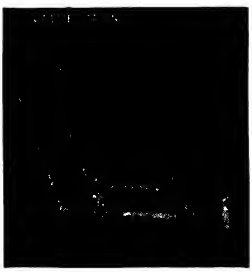
Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

Holland, Mich.

NO. 75 DESIGN—FLEXIBLE FIN TYPE with TURNING BLADE VALVE to provide DOUBLE DEFLECTION. Also without Valve as Grille or Intake



CONTROL OF AIR FLOW IN TWO PLANES



Instant Adjustment of Air Flow
(Up Straight or Down)

Is obtained by turning the regulator on the register face to the proper setting with a key furnished with each register. When the valve is opened, as shown at the left, the individual valve louvers automatically stop in position to provide the proper air flow—Up (Fig. 1) for cooling systems to avoid drafts; Straight (Fig. 2) for ventilating systems; Down (Fig. 3) for heating systems to prevent stratification. When the valve is closed, as shown at the left below, it completely stops the flow of air.

Air Flow Can be Quickly Adjusted Sideways

No. 75 Design has a flexible fin-type face. Each fin may be twisted individually with a wrench furnished, to provide any desired sideway deflection of air flow.

Greatly Reduced Turbulence and Resistance

Figs. 1, 2, and 3 show the air flow with No. 75 Design; Fig. 4, with the conventional register. Compare the turbulence in the stackhead of the latter with the smooth flow obtained with No. 75 Design.

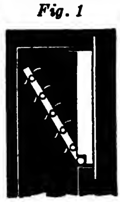


Fig. 1

Fig. 2

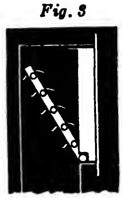
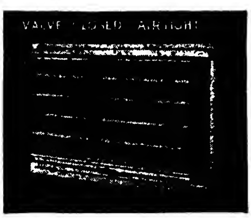
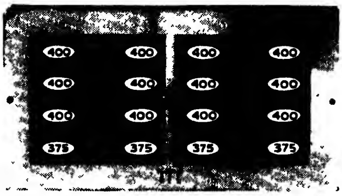


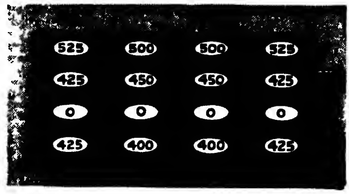
Fig. 3

Fig. 4





Velocities with No. 75 Design



Velocities with Conventional Register

EVEN DISTRIBUTION OF AIR OVER ENTIRE FACE

The turning blade valve distributes the air evenly with a uniform velocity over the entire face, as shown in Figs. 1, 2, and 3 on the preceding page. Note how the air rushes through the upper part of the face with a conventional register, as shown in Fig. 4. Since the entire face of No. 75 Design register is utilized for discharge of air, smaller and in some cases fewer registers can be used without causing excessive velocities.

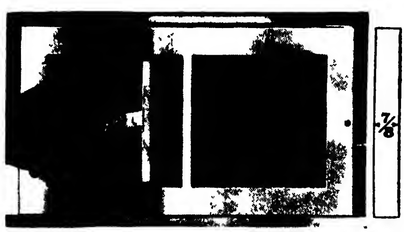
Prevention of Streaked Ceilings—With either UP, STRAIGHT, OR DOWN deflections the air does not strike the ceiling immediately in front of the register; streaked ceilings are thus avoided.

Excellent Concealment of Duct—The depth and close spacing of the vertical bars, combined with the valve, provide almost complete concealment of the duct, adding considerably to the pleasing appearance of the register face.

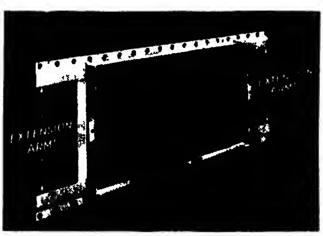
Special Settings—No. 75 Design functions equally well when located at the end of a horizontal duct or by installing it upside down, when the air is delivered to it from above.

AVAILABLE IN FOUR TYPES

With Turning Blade Valve—No. 751 Register (Left) has Sponge Rubber Gasket and 3/16 in. turndown. No. 754 Register (Right) is similar except has 7/8 in. projection and is furnished without gasket.



Without Valve—No. 750 Grille (Left) has 3/16 in. turndown. No. 757 Intake (Right) has 7/8 in. projection.



FOUR TYPES OF INSTALLATION FRAMES

No. 75 Design items can be used with or without installation frames. No. 3 Sidewall Stud Frame (illustrated), fastens directly to stud, forming a solid, streak-proof foundation for register. No. 8 Frame is similar for baseboard use. No. 5 Baseboard Stack Frame provides inexpensive, streak-proof installation. No. 2 Band Iron Frame provides for connecting register to stackhead.

CATALOG 49 showing the complete H & C line, available upon request.

Hendrick Manufacturing Company

48 Dundaff Street, **Carbondale, Pa.**

Sales offices in principal cities—consult telephone directories

Hendrick Bulators; Hendrick Perforated Metal Grilles; Hendrick Mitco Open Steel Flooring, Armorgrids, Shur-Site Treads

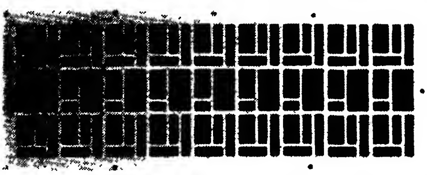
HENDRICK BULATOR*

Hendrick Bulator* is a practicable combination of an ornamental grille and a deflecting vane grille.

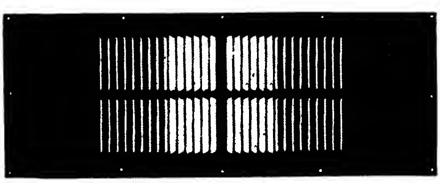
Properly placed behind the ornamental grille, the deflecting vanes of the Bulator give the air throw and spread specified by the engineer. The vanes are adjustable so that the air flow can be deflected to right or left, up or down, or in a combination of directions.

Although mounted just behind the ornamental metal grille, the deflecting grille is not noticeable, as will be seen from the illustration, made from an unretouched photograph.

Tests made at the Case Institute of Technology, Cleveland, under the direction of Professor G. L. Tuve, show that correctly designed ornamental metal grilles, properly mounted in front of the deflecting vanes, have little effect on the air stream pattern and air throw.



Photograph taken with deflecting vanes less than an inch behind grille, shows that vanes are not noticeable.



Vertical deflecting vanes, showing how the vanes may be set to produce any desired air stream pattern.

Using a 24 in. wide by 8 in. high combination of control louvers, vertical deflecting vanes, set 45° each way, and a metal grille with 65 per cent open area, the tests showed the following resistance pressure in inches of water:

Air flow volume cubic feet per minute				
400	500	800	1000	1200
Total throw of air stream in feet				
5.5	8.2	10.9	13.6	16.4
Resistance pressure with grille attached				
0.03″	0.08″	0.13″	0.21″	0.29″
Resistance pressure with grille removed				
0.02″	0.05″	0.09″	0.14″	0.20″

The report states: "In the series of tests, we found that at a given air volume the presence or absence of the 'Mosaic' (pattern) grille made very little difference on either the air stream pattern or the throw, but the resistance was changed by the presence of the grille, as shown by the table."

Thus the Bulator enables the architect to combine with the air-conditioning system of a building, ornamental grilles which harmonize with the decorative scheme without appreciably affecting the air flow.

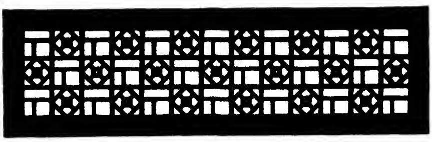
Hendrick's engineering department will be glad to cooperate with architects and engineers in the selection and installation of Bulator Grilles, incorporating any type of control desired.

*Beauty + Ventilator

HENDRICK PERFORATED METAL GRILLES

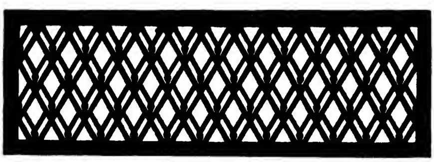
Hendrick decorative grilles are furnished in over a hundred patterns, and in a wide variety of overall dimensions, bar sizes, and number and size of perforations.

Many exclusive Hendrick designs, originally produced to meet an architect's specifications for some particular project, are now available as standard numbers, and facilities for making special designs to specifications make the Hendrick service even more complete. The wide range of patterns permits the choice of a grille that will harmonize with any style of architectural design or period construction.

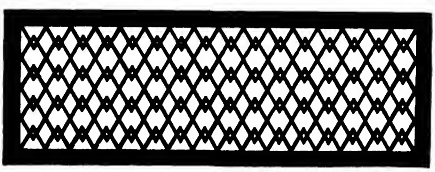


Arglin—63 per cent Open Area

Grilles are fabricated in heavy-gauge aluminum, bronze, copper, Monel, steel, stainless steel, and other commercially rolled metals. With ample open areas and accurate sizes, Hendrick grilles are characterized by clean-cut perforations, fine finish, and freedom from burrs and other imperfections.

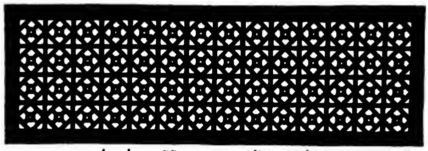


M-No. 2—57 per cent Open Area

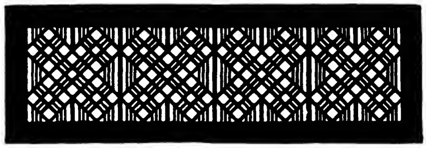


M-No. 9—67 per cent Open Area

They are easy to install, and always lie flat because of a special flattening operation in their manufacture.



Argive—60 per cent Open Area

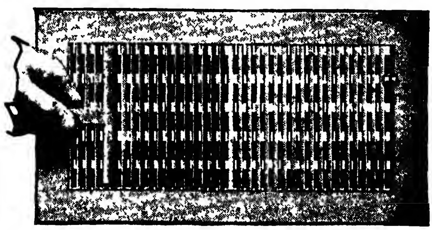


La Crosse—55 per cent Open Area

AIR CONDITIONING GRILLES AND REGISTERS

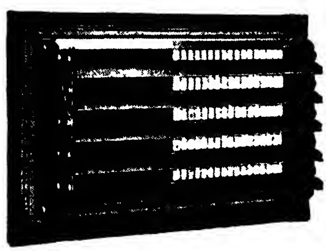
Hendrick air conditioning grilles and registers for directed air flow are made in various meshes, the standard having $\frac{3}{8}$ in. opening between face bars. All types are furnished with either horizontal or vertical directional bars.

The standard mesh can either be furnished with grille bars permanently set at the factory for straight or directional air flow, or furnished so that the grille bars may be individually adjusted on the job to direct air flow to any desired degree.



Front View

Rear View

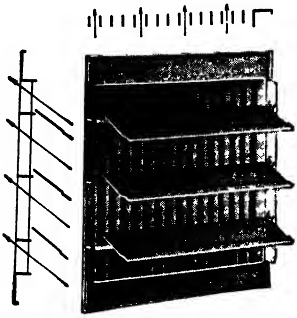


The Independent Register Co.

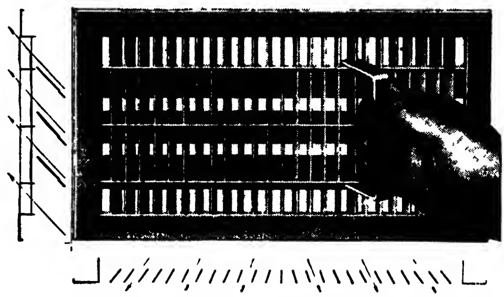
ESTABLISHED 1898

3747 East 93rd Street, **Cleveland, Ohio**

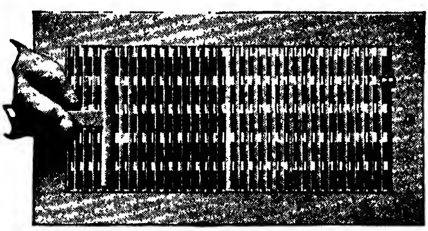
AIR CONDITIONING REGISTERS AND GRILLES



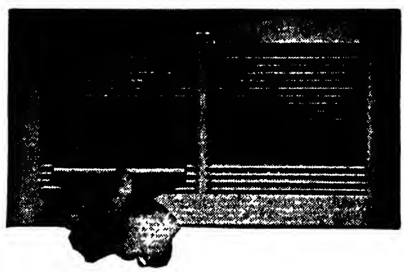
Rear View
Showing Adjustable Deflecting Vanes



No. 321A Grille with Deflecting Vanes—With vertical grille bars and horizontal deflecting vanes. The grille bars may be individually adjusted to direct air flows to right or left; and the vanes are made individually adjustable to deflect air flows up or down.



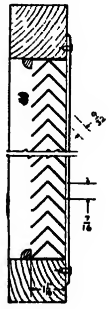
No. 238 Wrought Steel—4-way adjustable direction of air flow. Flexible vertical grille bars, multiple valves.



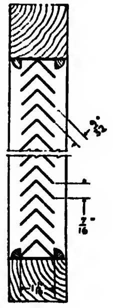
No. 139 Wrought Steel—Flexible horizontal grille bars, bendable for up, down or straight air flow. Single valves.

Independent No-Vision Grilles—No. 1312 for Doors, Walls and Partitions

The grille bars are "V" shaped; it is impossible to see through the grille from any viewpoint.



R



C

No. 1312R—With overlapping rim ⅝ in. wide, on all four sides.

No. 1312C—With grille core only, installed with moulding.

Etablissements NEU

47 rue Fourier—Lille

Nord **France**

Vega

Any of the three following functions may be obtained with **Vega** ventilation outlet, invented by J. Fourtier:

—delivery of a direct air vein,
—diffusion of the delivered air,
—no delivery at all.

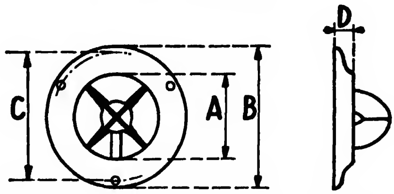
An adjustable device, formed by 4 semicircular blades set in a circular opening, may change the direction and the diffusion of air delivered by the ventilation outlet.

The **Vega** ventilation outlet is of a very reduced size, because the adjustable device is placed in the delivered air vein. Of very light weight, it is easily fitted up; it can be placed on a receiver supporting one or several devices, or on the ventilation duct itself, either vertical or horizontal. Its working is simple and rapid.

Ventilation outlet

Patent Applied for No. 665,689

Main Dimensions

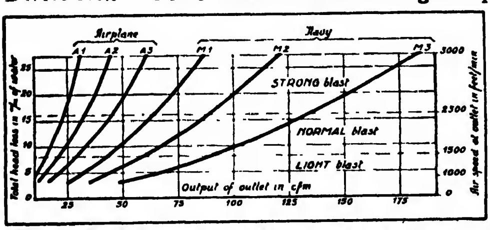


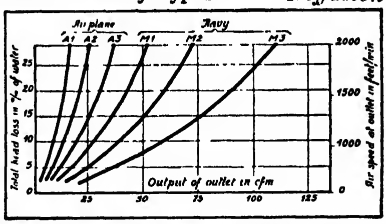
Type No.		Nominal Capacity cfm	A mm	B mm	C mm	D mm
AIR-PLANE	A1	20	57.5	80	70	10
	A2	30	45	80	70	10
	A3	40	53	80	70	10
NAVY	M1	60	65	132	116	17
	M2	90	75	132	116	17
	M3	135	90	132	116	17

weights: airplane: 110 gr. 3, oz 9—navy 240 gr. 8, 5oz

The special design and fitting of the device enable the quick adjustment of any outlet from one number to another.

Direct vein **Performance curves of Vega Airplane and Navy Types** *Diffusion*



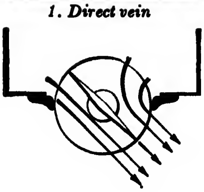


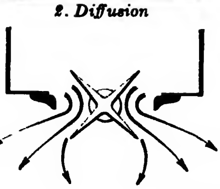
1. Direct vein

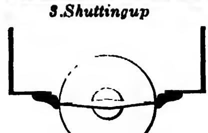
2. Diffusion

3. Shuttingup

Vega ventilation outlet is made of cast metal or cast material; its finish is made to suit all applications and to be in harmony with any kind of decoration.







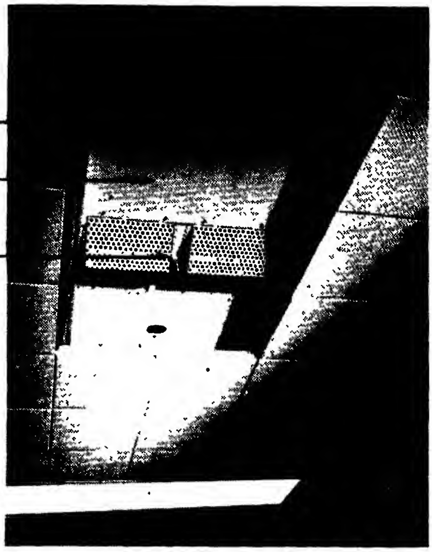
Main Position of Adjustable Device

The Pyle-National Company

Multi-Vent Division
1363-78 W. 37th St.
Chicago 9, Illinois

Sales Engineers and Agents
in Principal Cities of
U. S. and Canada

MULTI-VENT* *LOW VELOCITY* AIR DIFFUSION



Concealed Multi-Vent Panel exposed by removal of six squares of metal acoustical ceiling.

Panel Frame . . . installed in the bottom of air supply duct.

Control Plate . . . supporting one or more valves per panel, is hinged in panel frame providing ready access to duct above for cleaning.

Pressure Displacement Air Valve . . . single adjusting screw raises, and lowers a valve plate above opening in control plate to regulate volume of air flow from duct into dual V shaped primary distribution sections, the design of which insures even distribution of air over the entire perforated area below panel.

6 Outstanding Advantages

1. No Strong Air Streams to Direct: With Multi-Vent duct velocities are so radically reduced (within the diffuser itself) . . . diffusion is so rapid, thorough and wide-spread . . . that no air movement in excess of ASHVE comfort zone requirements exists more than six inches away from the perforated distribution plate.

2. No Deflection Problems to Restrict Location or Capacity of Outlet Panel: With Multi-Vent the location and the capacity of the diffuser can be determined solely by load considerations assuring maximum effectiveness and efficiency. The proximity of seating locations or the relative positions of partitions and lighting fixtures—which must be a major consideration in locating high velocity diffusers to avoid drafts—need not be considered with Multi-Vent regardless of ceiling heights.

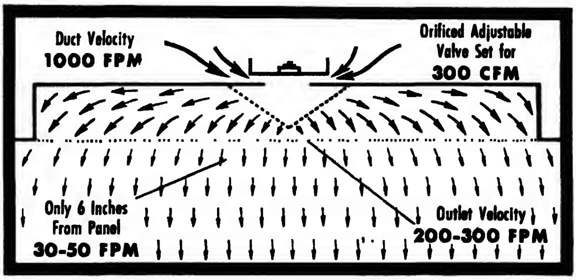
3. No Change in Air Diffusion Patterns When Desired Volume of Air Delivered is Varied: Multi-Vent's adjustable pressure displacement valve can be easily set for delivery of various amounts of air without disturbing the balance of the overall system. Neither single panel adjustments to suit occupants special requirements nor substantial reduction or increase of air capacity at source to meet seasonal demands will in any way affect the desired air flow pattern.

4. 40 Per Cent Higher DTD Will Meet Comfort Zone Requirements: Multi-Vent will permit raising the usual 15 deg Diffusion Temperature Differential to as high as 25 deg (with an 8 ft ceiling for

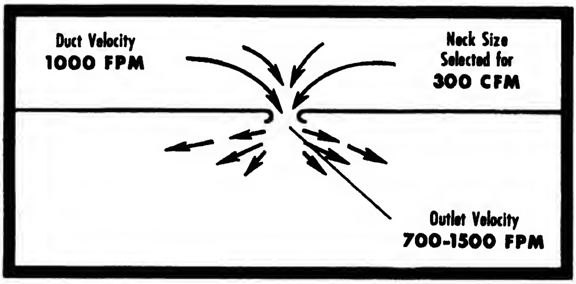
**This application of low velocity, pressure displacement air diffusion is fully protected by patents. Only with Multi-Vent can you enjoy its benefits. Multi-Vent is the registered trade mark of the Pyle-National Co., Chicago, for air distribution systems and parts thereof.*

THE PYLE-NATIONAL COMPANY
Multi-Vent Division

MUTI-VENT*-LOW VELOCITY



OTHER HIGH VELOCITY DIFFUSERS



example). Thus 40 per cent less air need be used to handle a given load making possible substantial economies in ducts, fans, filters and coils.

5. No Protruding Outlet Fixtures to Mar the Beauty of Modern Interiors: Multi-Vent can be completely concealed above the square perforated pans in a metal acoustical ceiling. Multi-Vent installed flush in all other type ceilings is less conspicuous than diffusers of any other make.

6. Uniformity of Room Temperature and Humidity: Multi-Vent can achieve a temperature differential of as little as 1 deg within the comfort zone in all seasons . . . and 2 deg is guaranteed. This insures *true* air conditioning comfort and will meet the most exacting air conditioning requirements for scientific research and industrial processing.

The Standard sizes of Types MVAR and MVAMC are given. Overall sizes are approximate.

Type	Width In.	Length Ft.	Gross Area Sq. Ft.	Net Free Area Sq. Ft.
MVAR-244	24	4	8	1.04
MVAR-245	24	5	10	1.33
MVAR-246	24	6	12	1.61
MVAR-364	36	4	12	1.64
MVAR-365-1	36	5	15	2.08
MVAR-365-2	36	5	15	2.08
MVAR-366-1	36	6	18	2.52
MVAR-366-2	36	6	18	2.52

Note: Types 365 and 366 are available with one or two valves.
Special sizes to accommodate lighting or ceiling decoration are available.

For use with acoustical metal ceiling pans installed below Multi-Vent:

Type	Width In.	Length Ft.	Gross Area Sq. Ft	Net Free Area Sq. Ft
MVAMC-244	24	4	8	1.04
MVAMC-245	24	5	10	1.33
MVAMC-246	24	6	12	1.61
MVAMC-364	36	4	12	1.64
MVAMC-365-1	36	5	15	2.08
MVAMC-365-2	36	5	15	2.08
MVAMC-366-1	36	6	18	2.52
MVAMC-366-2	36	6	18	2.52

Multi-Vent installations are simple, quick to balance and easy to clean. They have been applied with remarkable results to almost every type of building, new or old, and are particularly well adapted to the lower ceilings of modern architecture.

* See foot note on page 1204.

Application data will be furnished on request.

Register & Grille Mfg. Co.

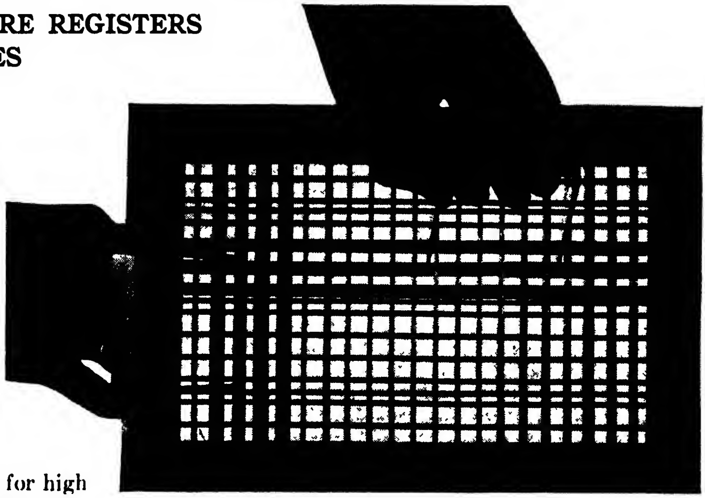
Incorporated

70 Berry Street, **Brooklyn 11, N. Y.**

Headquarters for all types of Registers and Grilles

RESIDENTIAL AND COMMERCIAL

DOUBLE CORE REGISTERS AND GRILLES



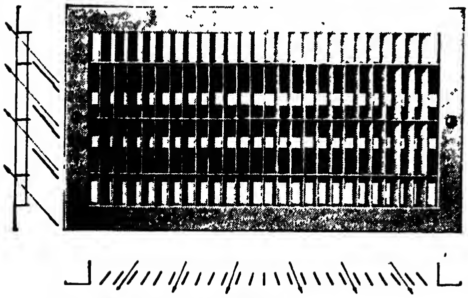
Style 620—Six-Way Adjustable Deflection

Designed only for high grade work where appearance and performance are the first consideration.

When used as a grille only, style 620 gives complete control of directional throw, and, when fitted with one of our many types of shutter, permits control of volume as well.

Can also be had with outside adjustable bars running horizontally (Style 610).

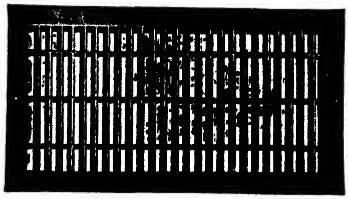
FOUR-WAY ADJUSTABLE DEFLECTION



Style 20 Grille and HMV deflecting vanes

Front bars vertically adjustable, rear vanes horizontally adjustable; *or* Front bars horizontally adjustable, rear vanes vertically adjustable.

TWO-WAY ADJUSTABLE DEFLECTION



Use No. 20 Grille for adjustable right and left deflection. Style 10 has horizontal adjustable bars for up and down deflection.

Ask for our catalog which shows other types of air controls; also 81 different Stamped Metal designs.

Rock Island Register Co.

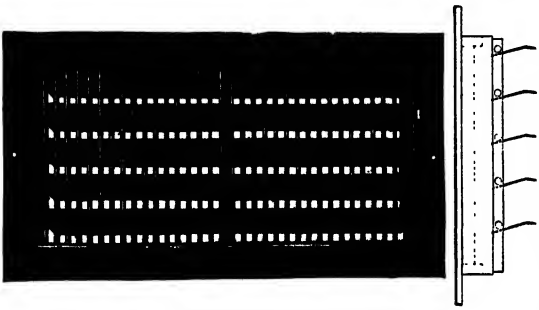
2435 Fifth Ave. Rock Island, Ill.

Air Conditioning Registers & Grilles

Specify ROCK ISLAND Registers

Designed for maximum efficiency in new and old homes and prefabricated construction. All combinations of air deflection possible.

The 800 Series

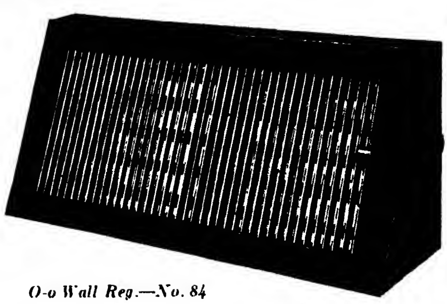


Air Vane Wall Reg — No 802M

800 Series are of fabricated construction and feature rigid assembly of vanes in groups permitting simple and secure adjustment to any angle from 0 deg to 45 deg, left or right, and are available in either horizontal or vertical styles. Multi-shutter valve assembly along with the adjustable grille make four-way deflection adjustment possible. Multi-louvres allow 22 deg upward and downward air flow. Multi-valve assembly is shallow, extending back into head sufficiently to give directional air flow without restricting air capacity.

The 80 Series

The Out-o-Wall Registers and Intakes were designed for installations where there is a girder, sill, or wall under the partition and where a standard register and head cannot be used. They are small, neat, and compact in design. Registers are complete with head, assembled as one unit. The Out-o-Walls are easy to install.



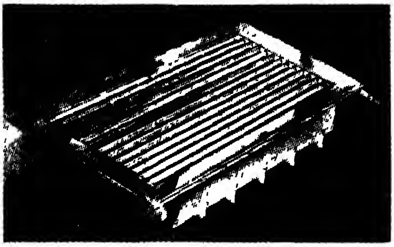
O-o Wall Reg.—No. 84

Out-o-Wall Registers and Intakes are manufactured in both 800 and 900 Series designs to match corresponding forced-air registers. Each register is shipped and priced complete with head—ready to install, fastening to the baseboard with two screws. Separate heads can be purchased for use with cold-air Intakes.

Standard Stamping & Perforating Co.

3111 W. 49th Place, **Chicago 32, Illinois**

Air Conditioning Registers and Grilles—Cold Air Faces
Perforated Metals for all Purposes

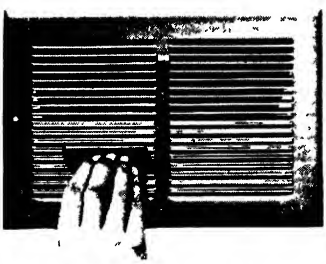


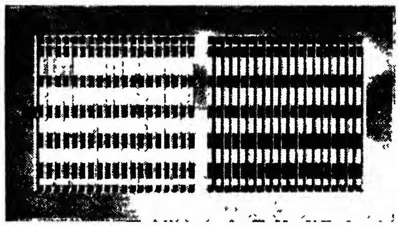
←No. 101 Floor Register

This model floor register features beveled edges, solid construction and constructed of fabricated metal. There are no loose parts. The finish is oak . . . an ultra strong baked-on finish for longer wear and it will match flooring. There is a matching cold air face available.

No. 41H Sidewall Register (for forced air installations)

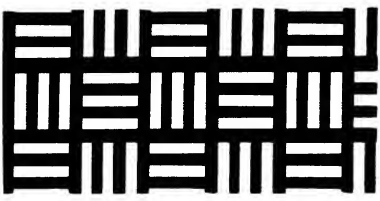
This model is smartly styled, efficiently designed for sidewall and baseboard installations. It has a single damper and $\frac{3}{16}$ in. turned down edge. Sponge rubber gaskets are attached. The illustration shows how easy it is to set the bars with a "Bend-Esy" tool. ↓



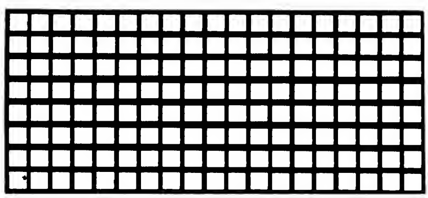


↑No. 331 Horizontal Multiple Valve Louvres Attached to Vertical "Bend-Ezy" Faces

The No. 331 Sidewall Register has Horizontal Multiple Valve Louvres and a $\frac{3}{16}$ in. turned down edge for flush sidewall installation. These are attached to vertical "Bend-Ezy" faces. This allows adjustable four-way air deflection. The attached sponge rubber gaskets are an extra feature that prevents the escape of air and chipping of wall Surface.



Design "N" *60 per cent free area*
1½ in. x 1½ in. unit *3/16 in. bar*



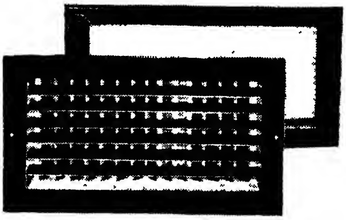
Plain Lattice Design

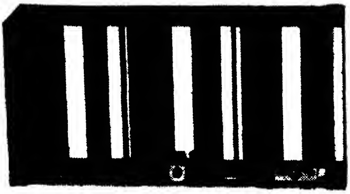
All Stanforated Grilles and ornamental designs are available in steel from 16 gage to ½ in. thicknesses. They can also be furnished in non-ferrous metals, such as aluminum, brass bronze and stainless steel, and in varying thickness according to the physical properties of each metal.

A complete line of registers, grilles and scoops for all types of industrial and residential air conditioning applications.

Stewart presents 54 styles in 2140 sizes for engineer or contractor with special as well as standard specification requirements. Engineering literature available on all technical and price information. Data conveniently arranged for estimating large or small installations. Standard material is cold rolled steel, but stainless steel, aluminum and brass are also available for all products.

Style DDH, ALL-WAY SELECTIVE AIR-THROW, has two banks of individually adjustable fins—front bank horizontal, rear bank vertical. Fins are 14 gauge steel, $\frac{3}{4}$ in. deep, spaced on $\frac{3}{4}$ in. centers. Bank of lever- or key-operated 1 in. multiple valves can be furnished as addition or substitution. Valve blades overlap when closed and open through 110 deg arc. Overall size of face $1\frac{3}{8}$ in. greater than duct opening. PLASTER FRAME allows for removal of grille when desired without damage to surrounding painted areas. Holes supplied for attaching plaster frame to duct, and lintel frame or stud. Usable with composition duct, furnishing base to which register is screwed. If used with composition duct, please specify.





BALDAMPER is designed to limit volume and direct air at 90 deg to rear of supply outlet. Even distribution across entire face results. Fins operate simultaneously, in pairs. Adjustment made with removable key (or knob, not shown). The maximum-volume-positioner in lower right corner is supplied with knob-type only.
Key-type is for permanent setting as balanced. Knob-type permits full closing, or opening to the point determined by maximum-volume-positioner.
Available as a damper but generally supplied complete with grille face as one unit. Any Stewart grille face is available.

Style 94, MULTIPLE VALVE DAMPER WALL REGISTER, is a unit of the Stewart residential line. Horizontal face bars $\frac{1}{4}$ in. center to center and $\frac{1}{4}$ in. deep, deflect air downwards. Multiple valve damper is lever operated. 1 in. valves overlap when closed and open on 110 deg arc. Face bars can be adjusted with removable key furnished. Overall size of register $1\frac{3}{8}$ in. greater than duct opening. All grilles and registers have zinc chromate prime coat finish and sturdy rubber gasket is cemented on at factory.

Technical and price information available through representatives in all principal cities, or at factory headquarters.

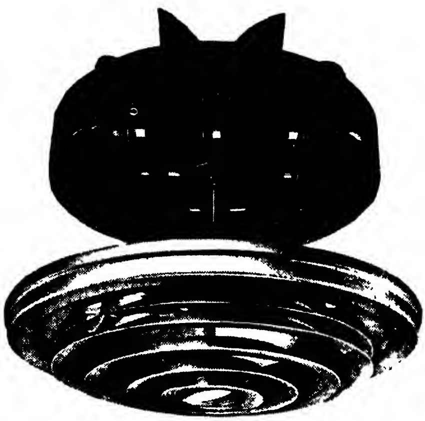
IN CANADA: DOUGLAS ENGINEERING CO. LTD, 190 MURRAY ST, MONTREAL 3, P.Q.

AEROFUSE

Diffuser

and

Damper

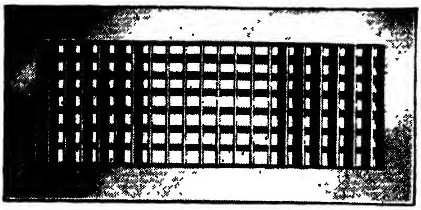


The efficient performance of an air conditioning installation depends largely upon the design of the distributing units. Tuttle & Bailey Type EAC Aerofuse Diffusers, with Effective Area Control, an exclusive Tuttle & Bailey feature, provide complete adjustability in the field . . . positive, job-tailored control at the vital point of air delivery . . . assure the correct air quantity, delivered as you want it, where you want it, evenly distributed and without drafts.

Aerofuse Diffuser—The new Type EAC Aerofuse is equipped with an auxiliary cone, operated from the face, which provides a simple method of varying the performance of the diffuser to meet specific job requirements.

Aerofuse Damper No. 4—An improved multi-louvre damper with exclusive designed-in features that provide positive volume control. Adds real efficiency when installed with Aerofuse Diffusers. Minute adjustment of volume for accurate balancing . . . multi-louvres divide supply stream, mean minimum turbulence and quiet operation. Synchronized deflector vanes assure even distribution regardless of volume. In open position, damper provides effective area greater than that of corresponding size diffuser . . . closed, it assures 100 per cent shut-off. Tamper-proof, louvres may be locked in any position.

Tri-Flex SUPPLY AIR GRILLES AND REGISTERS



26

STANDARD SIZES

8 x 4
10 x 4
10 x 6
12 x 4
12 x 5
12 x 6
14 x 4
14 x 5
14 x 6
16 x 5
16 x 6
20 x 5
20 x 6
20 x 8
24 x 5
24 x 6
24 x 8
24 x 10
24 x 12
30 x 6
30 x 8
30 x 10
30 x 12
36 x 8
36 x 10
36 x 12

GRILLES—Provide maximum control of air deflection. Individually adjustable face bars, pivoted at front. Available with horizontal or vertical bars.

DOUBLE DEFLECTION GRILLES—Provide adjustability of both vertical and horizontal direction of air path. Available with vertical face bars and horizontal rear bars . . . or with horizontal face bars and vertical rear bars.

MULTI-SHUTTER REGISTERS—Combine grille and multi-shutter unit for complete air shut-off. Available with vertical face bars and horizontal back blades . . . or with horizontal face bars and horizontal back blades.

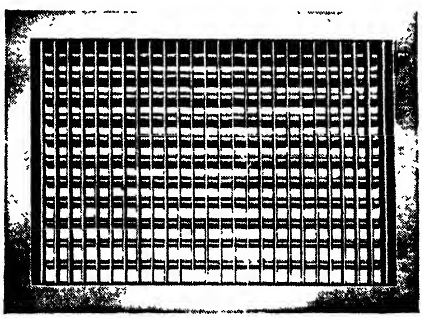
DOUBLE DEFLECTION MULTI-SHUTTER REGISTERS—Provide maximum directional and volume control. Available with vertical face bars, horizontal rear bars and horizontal back blades . . . or with horizontal face bars, vertical rear bars and horizontal back blades.

Aerovane RETURN AIR GRILLES AND REGISTERS

10 x 6
10 x 8
12 x 6
12 x 8
12 x 12
18 x 6
18 x 12
18 x 18
24 x 12
24 x 18
24 x 24
30 x 12
30 x 18
30 x 24
36 x 18
36 x 24
36 x 30
48 x 24
48 x 30
48 x 36

20

STANDARD SIZES



GRILLES—Furnished as standard with vertical bars set straight or horizontal bars set at angle of 35 deg.

REGISTERS—Furnished as standard with vertical or horizontal bars set straight... horizontal multi-shutter blades.

SANTROLS

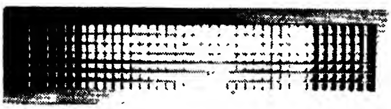
A flexible device that provides positive control of air volume as well as uniform distribution over the entire supply outlet. Blades fully adjustable. Furnished as standard in same 26 sizes as listed for TRI-FLEX.

For engineering details write Tuttle & Bailey, Inc., New Britain, Conn.

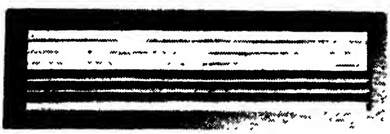
Titus Mfg. Corp.

WATERLOO, IOWA

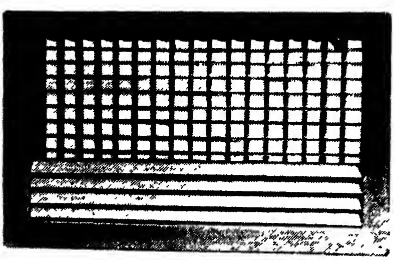
Titus Mfg. Corp. are designers and manufacturers of Airfoil Grilles featuring the Airfoil *Louvre*—patterned after the airfoil section of an airplane. The following distinctive features identify Titus Airfoil Louvres. (1) Smooth as glass streamlined surface (2) Solid construction (3) Noiseless performance (4) Minimum turbulence.



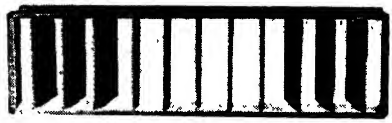
L-2 4-Way Directional Titus Airfoil Grille features horizontal front louvres and vertical rear louvres. Frame is constructed in one piece regardless of size. Custom built for individual requirements. Designed with streamlined Titus Airfoil louvres. Available as S-2 front louvres vertical, rear louvres horizontal.



L-1 Airfoil Grille features single set of horizontal Titus Airfoil louvres set on ¾ in. centers. Solid one piece frame construction. Easy blade adjustment for long or short throw. Available as S-1 with louvres vertical.



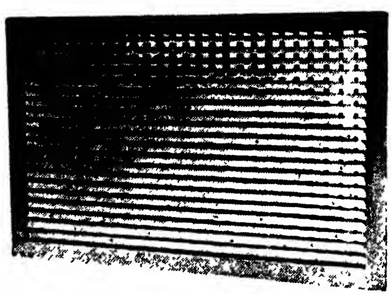
L-4 Airfoil Multi-Shutter Register—Lever operation gives positive shut off of blades. Opens uniformly. Front individually adjustable airfoil louvres spaced on ¾ in. centers.



A-G-25 Volume Controllers provide positive control of air volume. Blades individually adjustable. Sponge rubber gasket holds unit firmly in duct preventing leakage. No screws necessary. Dull black lacquer finish.

R-L-21 Return Air Grilles—blades on ⅝ in. centers. Parallel to long dimension. Easy to clean.

R-L-23 Return Air Grille—blades on ¾ in. centers. Parallel to long dimension. Deep fins. Standard 1¼ in. beveled border. Grey lacquer primer finish. Any size grille can be furnished. Special matching finishes on request.



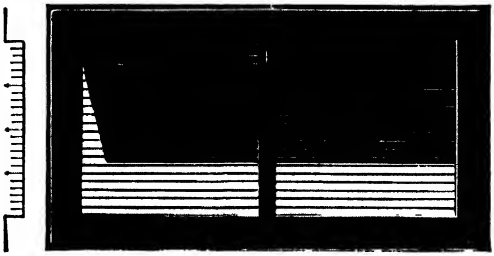
R-L-24 Return Air Register—Features rear multi-shutter louvres—quick shut off. Rear blades are deep-spaced on 1 in. centers.

United States Register Company

General Offices: Battle Creek, Mich., U.S.A.

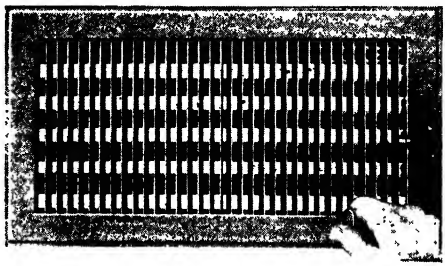
Branches: Minneapolis, Minn., Kansas City, Mo., Albany, N. Y.

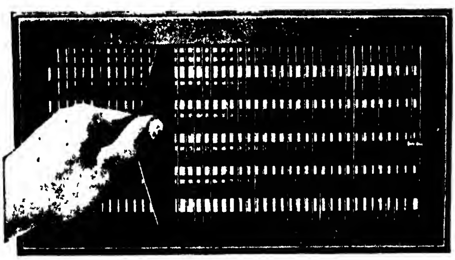
Air Conditioning Registers, Vents and Grilles



No. 153—Single-Valve Air Conditioning Register-Bars ¼ in. deep —Spaced 4 openings to the inch affords Non-Vision. Can be supplied in Directional Flow in either Horizontal or Vertical Bar Styles. Can be furnished with all styles of Setting Frames.

No. 249—Multiple-Valve Air Conditioning Register. Gives complete Air Control. Vertical Front Bars—Key-pin adjusted to provide 45 deg Right and Left or Two-way Side Flow. Lever operated Horizontal Back-valves give from Full Closed to any degree of Upflow and to 45 deg Downflow. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.



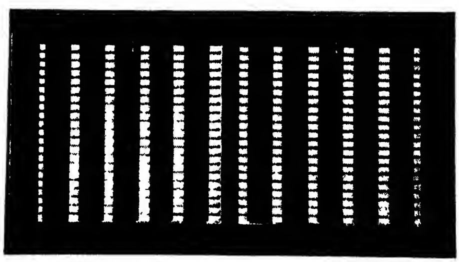


No. 256—Multiple-Valve Flex-bar-Air Conditioning Register. Vertical Front Bars set 22 deg Right and Left. Side Flow Deflection attained by setting of Grille Bars with bending wrench to accommodate room condition. Back-valves give same Up and Down control of air flow as No. 249 above. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.

All of above Styles can be supplied with either Lever or Individually adjusted Multiple Valves or Louvers. *i.e.* 177VVI—Vertical Valves Individually adjusted. 145VVL—Lever operated Vertical Valves.

Grilles and Vents in Matching designs are available.

For Complete Information Write for Latest Catalogs with Engineering Data.



No. 153-VVL Horizontal Bar Non-Vision Design-Vertical lever operated rear valves.

Complete Gravity and Air Conditioning Register, as well as Fitting Catalogs furnished on request.

Young Regulator Company

5209 Euclid Avenue, **Cleveland 3, Ohio**

DAMPER REGULATORS; REMOTE CONTROLS; VOLUME CONTROL GRILLES; DAMPERS

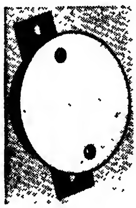
Manually and Automatically Controlled

Sales Representatives in Principal Cities

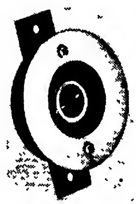
Young Regulators meet practically every condition where damper regulators are required for controlling air volume.

No. 1A—For installation on finished wall. May be locked in any position. Size ⅜ in.
No. 301A and 201A—For imbedding in plaster. Cover plate flush with finished wall.
No. 403A—For mounting on duct, lever adjustment. May be locked in any position. Sizes ¼, 5/16 and ⅜ in.
No. 401A—For mounting on duct. Key adjustment. May be locked in any position. Sizes ¼, 5/16 and ⅜ in.
No. 900A—For operating a splitter damper or deflecting vanes.
No. 656 —End bearing provides a bearing for the damper rod at side away from regulator.

No. 1 A



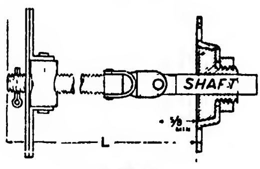
No. 301 A



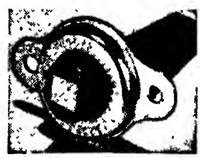
No 201 A

No. 403 A

No. 401 A



No. 900 A



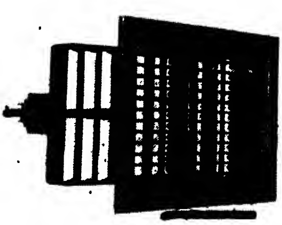
No. 656

Young Remote Control and Dampers.

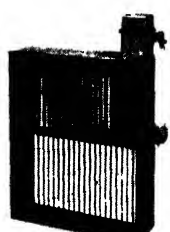
No. 700A—For remote control of one or more dampers at distances up to 250 ft or more.
No. 704 —Corner pulley eliminates friction on long rung or where there are many turns.
No. 915 —Volume control grille gives equal distribution of air over entire grille and directional flow.
No. 815 —Relief damper controls temperature in individual rooms. Remote bulb thermostat regulates dampers to give desired temperature in each room.

No. 700 A

No. 704



No. 915



No. 815

United States Steel Corporation Subsidiaries

Carnegie-Illinois Steel Corporation, Pittsburgh and Chicago
Columbia Steel Company, San Francisco
Tennessee Coal, Iron & Railroad Company, Birmingham
United States Steel Export Company, New York
District Offices in all Principal Cities

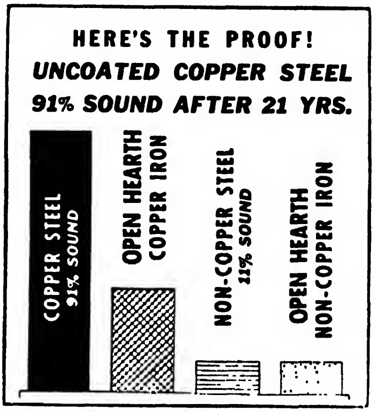
U•S•S COPPER STEEL
For Superior Rust Resistance at Low Cost

Corrosion resistance and cost are two determining factors of the type of metal to be used for various air-conditioning jobs.

Copper Steel has 2 to 3 times the atmospheric corrosion resistance of plain steel or pure iron as shown in the results of unbiased tests made at Pittsburgh, Ft. Sheridan and Annapolis by the American Society for Testing Materials.

The cost of U·S·S Copper Steel is less than that of pure iron or copper-bearing pure iron and only slightly more than plain steel. Thus there often is a dividend of 200 per cent to 300 per cent longer life and a saving in the first cost as well.

When galvanized, U·S·S Copper Steel produces a sheet that is rust resistant all the way through—not just on the surface. It should be used for all ducts carrying humidified air or placed in damp locations such as basements, shower rooms, etc.



Corrosion test of A.S.T.M. on 22 gage black sheets exposed at Annapolis, Md., October, 1916. The copper steel sheets outlasted all others in the test.

U•S•S PAINTBOND

U•S•S PaintBond should be used whenever galvanized steel is to be painted. This special Bonderized sheet can be painted immediately, offers a much better surface for painting, lessens danger of the paint flaking and retards corrosion. It is used for ductwork, furnace housings and outdoor metal work. Produced in Chicago district.

Send for our PaintBond booklet.

U•S•S GALVANNEALED

U•S•S Galvannealed has a tight, matte-surface zinc coating that withstands severe forming operations without peeling or cracking. Suitable for immediate painting without aging or other surface preparation. Produced in Chicago district.

U•S•S DUL-KOTE

U•S•S Dul-Kote is a specially treated non-spangled galvanized sheet which also can be painted immediately without aging or otherwise preparing the surface. Produced in Birmingham district.

Send for our Dul-Kote booklet.

OTHER U•S•S PRODUCTS INCLUDE:

Black Sheets—All grades, hot rolled, cold rolled in a number of different finishes.

Stainless—Heat resisting steel for various uses where temperatures are high and corrosion severe.

Cor-Ten—High strength steel—greater strength, greater atmospheric corrosion resistance for smokestacks, hoods, etc.

Chicago Metal Hose Corporation

EXPANSION JOINT DIVISION

Maywood, Illinois

PLANTS

Maywood, Elgin and Rock Falls, Ill.

District Offices

Atlanta Boston Chicago Cleveland Detroit Ft. Worth Los Angeles New York Philadelphia Pittsburgh San Francisco

In Canada: Canadian Metal Hose Company, Ltd., Brampton, Ontario

CMH CONTROLLED-FLEXING EXPANSION JOINTS

For pressures up to 300 psi—copper liners for temperatures to 400 F, stainless steel liners for temperatures to 1400 F (under certain conditions).

CMH Controlled-Flexing Expansion Joints combine CMH "engineered curvature of corrugations" with precision designed control rings. The corrugation-mated control rings provide guided flexing of the joint and prevent any permanent deformation of corrugations. Control rings are firmly anchored into corrugations to close working dimension tolerances. Expansion travel up to $6\frac{3}{4}$ in. may be secured with a single CMH Expansion Joint.

Standard sizes from 4 in. to 24 in. I.D. in either stainless steel or copper. Available with or without stainless steel or Monel internal sleeves.

Cut-away view of CMH Controlled-Flexing Expansion Joint with flanged ends.

CMH Controlled-Flexing Expansion Joint with Welding Ends.

CMH FREE-FLEXING EXPANSION JOINTS

. . . for pressures up to 30 psi . . . Stainless Steel or copper . . . temperatures to 1400 F for Stainless Steel units (under certain conditions) and 400 F for copper.

CMH Free-Flexing Expansion Joints are made with single or multiple corrugations. Designed for expansion travel up to 1 in. per unit. Used in low to moderate pressure systems for controlling expansion up to $\frac{5}{16}$ in. with each expansion joint. Additionally, misalignment correction or offset motion is calculated at $\frac{1}{16}$ in. per corrugation when two or more corrugations are used. When more than four corrugations are used a greater lateral motion may be allowed, depending on service requirements.

Standard sizes from 4 in. to 24 in. I.D. in either stainless steel or copper. Available with or without stainless steel or Monel internal sleeves.

Left—CMH Free-Flexing Expansion Joints with multiple-corrugations require less force to compress than single-corrugation joints. Force necessary to compress multiple-corrugation joints may be calculated for copper by multiplying inside diameter in inches by 100; for stainless steel, 150 times diameter in inches. Right—CMH single-corrugation Expansion Joint for travel up to $\frac{1}{2}$ in. Force in pounds necessary to compress joint $\frac{1}{2}$ in. may be calculated for copper by multiplying 200 by inside diameter of pipe in inches; for stainless steel, multiply pipe diameter in inches by 300.

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence 1, R. I.**

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PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Flared Tube Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Piping for acids and other special materials.

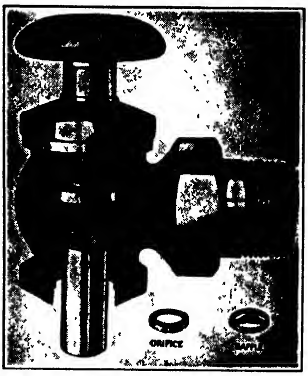
Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

Valves: Check, Diaphragm, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance. This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

Equiflo Data Book sent to interested parties.

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers. Their time and trouble-saving qualities during installation are equally exceptional. Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers. Send for Hanger Catalogue showing complete line.

Adjustable Swivel Rings (Patented)

Fig. No. 101
Solid Ring

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The Split Ring permits adjustment either before or after Ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

Fig. No. 104
Split Ring

Adjustable Swivel Pipe Rolls (Patented)

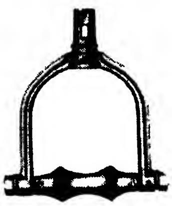


Fig. No. 174
Swivel Pipe Roll

An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

CB-Universal Concrete Inserts (Patented)

Made of malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in., ⅝ in., ¾ in. or ⅞ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

Fig. No. 282
CB-Universal Insert

GRINNELL WELDING FITTINGS

90° Elbow, Long Radius

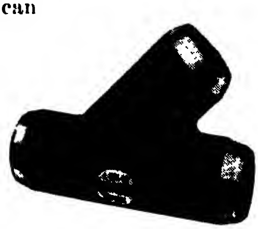
Grinnell Welding Fittings are made from Seamless Steel Pipe or tubing and possess the same physical characteristics as standard, extra strong and o.d. steel pipe or seamless steel pipe of comparable size. They can be used under the same conditions, pressures and temperatures as the pipe itself.

All Grinnell Welding Fittings have welding faces for all plain circumferential butt welds scarfed or beveled as follows: For wall thicknesses 3/16 to ¾ inch inclusive, 37½ deg. ± 2½ deg., straight bevel. Angles of bevel other than 37½ deg. can be furnished on special order.

45° Elbow, Long Radius

Welding Tee

Lap-Joint Stub End with Lap Flange attached



Welding Lateral

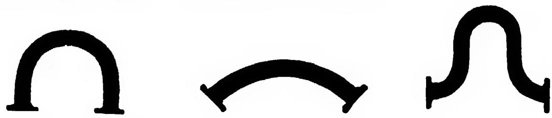
Arthur Harris & Co.

210–218 N. Aberdeen Street **Chicago 7, Ill.**

ENGINEERS AND BRONZE FOUNDERS—FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

Metals Fabricated—Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel, Inconel, Stainless Steel and KA2 SMO. Bulletin on request.

Bends



We make bends in every shape from all sizes of copper tube, pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U-bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing.

Non-Ferrous Castings—"Dairywhite" nickel silver for Process Industries Equipment. Suitable for milk and food products machinery. Castings also of 88-10-2. 80-10-10, 85-5-5-5, silicon bronze and manganese bronze, and special mixtures.

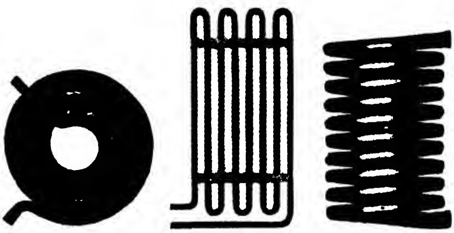
Copper Expansion Joints

For low pressure and vacuum. Made in two styles—convex and concave. Sizes 4 in. to 60 in. diameter. Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered: B-290 available only in sizes 4 in. to 15 in. inclusive.

B-280 Convex

B-290 Convex

B-281 Concave



Coils

For heating, cooling and condensing. All shapes made from any size pipe or tube—standard or special connections, of copper, brass, aluminum, stainless steel, KA2 SMO, monel, inconel, nickel, block tin, and Everdur.

Metal Floats

Column

Ball

Flat Cylindrical

Cylindrical

Cylindrical

Made of copper, plain steel, copper plated steel, stainless steel, KA2 SMO, aluminum, brass, Monel, pure nickel, Admiralty and Everdur, for open tank and all pressures.

Seamless copper ball floats carriedi n stock in diameters of 3 in., 4 in., 5 in., 6 in., 7 in., 8 in., 10 in., 12 in. for open tank and pressures of 25, 50, 100 and 150 lb. Floats in special sizes and pressures—made to order. Stainless steel ball floats 2½ in. to 12 in. for high pressure and corrosion carried in stock—special stainless steel floats made to order—stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request.

Alco Valve Company

ENGINEERED REFRIGERENT CONTROLS

851 Kingsland Avenue, St. Louis 5, Mo.

NEW YORK OFFICE: 122 EAST 42ND STREET

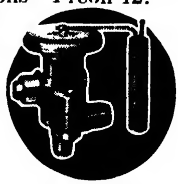
THE COMPLETE LINE OF REFRIGERANT CONTROLS

ALCO THERMO EXPANSION VALVES

THERMO EXPANSION VALVES: for automatic control of liquid refrigerant on all types of refrigeration and air conditioning systems. Capacities—from fractional tonnage to 100 tons Methyl Chloride, 50 tons "Freon-12."

Thermo-Limit—with pressure limiting feature and capacity change.

Type TK—"3 valves in 1"



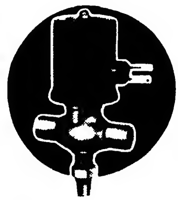
Type TCL

Type TR—Multi-Outlet

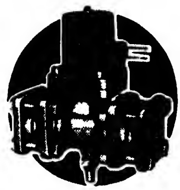
ALCO SOLENOID VALVES

SOLENOID VALVES: for all types of service. For Liquid: "Freon"—up to 75 tons. Methyl Chloride—up to 150 tons. For Suction: "Freon"—up to 8.8 tons. Methyl Chloride—up to 17 tons.

Type S1



Type M3

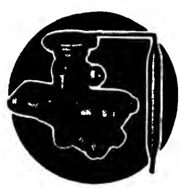


Type R2

ALCO AMMONIA CONTROLS

AMMONIA CONTROLS: Solenoid Liquid Valves—up to 172 tons. Solenoid Suction Valves—up to 28 tons. Thermo Expansion Valves—from fractional tonnage to 60 tons.

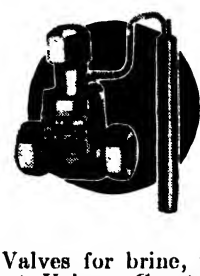
Type M9F



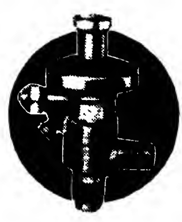
Type TG

ALCO SUCTION LINE CONTROLS

Type EPR13—for all refrigerants, with connection sizes up to 6 in.



Type 732 SNAP ACTION SUCTION VALVE—Temperature—½ ton, "Freon-12" — 1 ton, Methyl Chloride.



Type 760 "EVAPO-TROL"—Pressure — ½ ton, "Freon-12" — ¾ ton, Methyl Chloride.

ALCO also makes: Solenoid Valves for brine, water, gas, air and steam—Float Switches—High Pressure Float Valves—Constant Pressure Expansion Valves—Liquid and Suction Line Strainers.

Detroit Lubricator Company

Division of AMERICAN RADIATOR & Standard Sanitary CORPORATION

General Offices 5900 Trumbull Ave., Detroit 8, Michigan, U. S. A.

NEW YORK 18, N. Y., 40 West 40th Street — CHICAGO 5, ILL., 816 S. Michigan Avenue

Export Department—Box 218, RIDGEFIELD, N. J.

Canadian Representatives:

RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg

Automatic Controls

No. 411

Detroit Lubricator Company manufactures a very complete line of electrical controls, designed to open or close an electrical circuit with changes of temperature or pressure. The No. 411 Thermostat (illustrated) is a low voltage model and is made in plain and Day and Night types. All No. 411 Thermostats are available with heat compensation to provide smooth, accurate temperature control. The No. 855 Mercoid Room Thermostat (illustrated) is a line voltage type—available in heating, air conditioning and refrigeration ranges.

No. 855

For industrial use the No. 250 and No. 450 line of pressure and temperature controls is available, in pressure ranges from 20 in. vac. to 350 lbs, and in temperature ranges from −30° F to 495° F. Write for Catalog No. 100-C.

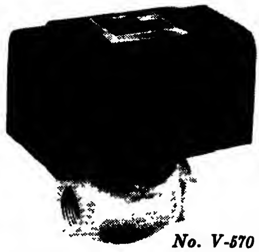
Furnace Controls

Also available is a full line of blower, limit, combination blower and limit controls, such as the CA-815 illustrated.

There is a "Detroit" control available for practically every application where a dependable device is required to open or close an electrical circuit with changes of pressure or temperature. Write for complete information. Our Engineering Department is always ready to make recommendations on any specific problem. Write for Catalog No. 100-C.

No. CA815

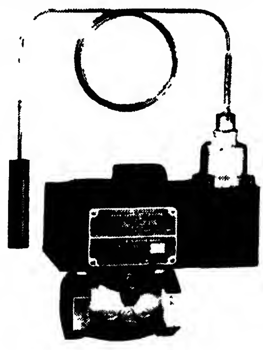
Gas Valves



No. V-570

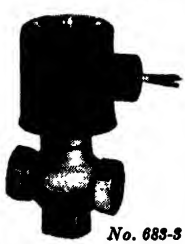
The No. V-570 Electric Gas Valve is an electrically operated valve for control of gas lines from ½ to 1¼ in. It provides partial opening upon initial operation, permitting quiet ignition of gas. Inexpensive, compact, attractive in appearance, and easily serviceable. Write for Catalog No. 300.

No. V-574 Electric Gas Valve has all the features of No. V-570, with the addition of mechanical limit control. If pressure or temperature exceeds the limit setting of the valve, the valve will close and remain closed until normal operating conditions are restored. No. V-574 is available with pressure type mechanical limit control for steam (V-574-S) or with temperature type mechanical limit control for warm air or hot water, (V-574-FW). Operation is quiet and the valve functions independently of gas pressure. Ask for Catalog 300.



No. V-574

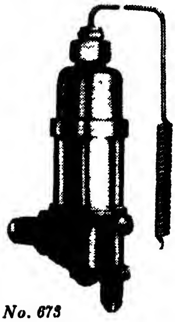
Solenoid Valves



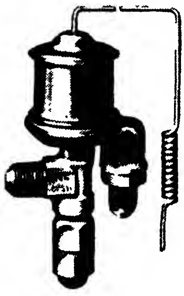
No. 683-3

"Detroit" Solenoid Valves for control of water, air, oil, gas, or refrigerant, embody many desirable features. A.C. hum is minimized and they will open against high pressures. Available in all standard voltages A.C. or D.C. No. 683-3 (illustrated) is a small size

valve with $\frac{3}{8}$ in. connections. No. 681 is a pilot operated, intermediate size valve, and the No. 686 is a large pilot operated valve with capacity up to 17 tons Freon-12. No. 686 valve available with flanged connections. No. 681 and 686 are furnished with manual opening feature to permit opening in case of current failure. All models may be taken apart and cleaned in the field without removing from pipe lines. Write for Bulletin No. 199.



No. 673



No. 573

"Detroit" Thermostatic Expansion Valves

"Detroit" Thermostatic Expansion Valves are designed to keep the evaporator in a refrigerating system completely refrigerated. Power elements are "gas charged" to a definite pressure, for motor overload protection; providing quicker response and more sensitive control. Capacities from $\frac{1}{2}$ to 20 tons Freon-12. All needles and seats are made of Delubaloy, a very hard, corrosion resistant alloy to insure long, trouble-free service.

No. 673, the "standard of the industry," has a long record of dependability. It is sensitive and accurate in operation. No. 573 has the quality and operating characteristics of the 673 for smaller installations. It has a *single* power element with double diaphragm. Write for catalog No. 200B for complete expansion valves listings.

No. 893

"Detroit" Single Diaphragm Expansion Valves

These valves are of single diaphragm construction. The diaphragm is made of special alloy, carefully tested for flexibility to assure uniform capacities and smooth operation. Power elements are gas charged for protection against motor overload and for maximum sensitivity. Needles and seats are hard, corrosion resistant Delubaloy.

Automatic and thermostatic types in capacities from $\frac{1}{2}$ to 20 tons Freon-12 and .9 to 36 tons methyl chloride. Write for Catalog No. 200-B.

Refrigerant Distributors

"Detroit" No. 790 Refrigerant Distributors insure a uniform supply of refrigerant to all sections of multiple circuit evaporators and are attached directly to the expansion valve by means of a flanged or union connection. These corrosion-resistant distributors are available for "Detroit" Thermostatic Expansion Valves No. 786, No. 787, No. 788 and No. 899.

They are made in five types of 2 to 18 passes for the above valves.

All distributors are designed for use with $\frac{1}{4}$ in. O.D. copper outlet tubes.

For complete capacity tables write for Bulletin No. 207.

"Detroit" Float Valves For Vaporizing Oil Burners

CRC 239 Single Metering Float Valve

Used On Oil Burning Water Heaters, Space Heaters, Furnaces, Ranges.

"Detroit" Float Valves provide the best, most reliable control for vaporizing oil burners. Easily cleaned, maintenance is never a problem. Full temperature compensation assures even flow of fuel regardless of oil temperature.

Model shown is for manual operation. Models are available for thermostat control—with draft fan control—and with temperature control (for water heaters, etc.).

Write for Catalog 400.

GENERAL CONTROLS

801 ALLEN AVENUE **GLENDALE 1, CALIF.**

Manufacturers of Automatic Pressure, Temperature, Level & Flow Controls

FACTORY BRANCHES: BIRMINGHAM (3), BOSTON (16), CHICAGO (5), CLEVELAND (15), DALLAS (2), DENVER (10), DETROIT (8), HOUSTON (2), KANSAS CITY (2), NEW YORK (17), PHILADELPHIA (40), PITTSBURGH (22), SAN FRANCISCO (7), SEATTLE (1) OKLAHOMA CITY (1) • **DISTRIBUTORS IN PRINCIPAL CITIES**

THERMOSTATS

Type T-70

Compact, snap-action, T-70 Metrotherm Thermostat. Functional beauty for accurate, remote control of desired temperature. Extends only ⅞" from wall. Streamlined stainless cover, sensitive to slightest temperature change, ivory plastic base.

MAGNETIC GAS VALVES

Type K-3B

Versatile, two-wire, straight magnetic current-failure valve. Packless. Insures tight shutoff indefinitely. Humless. Size range, ⅞ in. to 6 in. I.P.S. Operating pressures up to 5 lb. Voltages and frequencies A.C. or D.C. Quiet, positive, trouble-free. Available in explosion-proof housing.

SLOW OPENING GAS VALVES

Type B-55

For industrial and commercial burners and furnaces. Adjustable opening time, 5 to 50 secs. Wide pressure range up to 5 lb. Size ¾ in. to 6 in. I.P.S. Ample power for louvre control. Damper arm easily rotated. Packless.

MAGNETIC VALVES

Type K-10

Provides six times more power than ordinary solenoid valves. Controls air, gas, water, light, heavy oils, steam. Positive opening, complete shut-off, packless, humfree. Available for any voltage, A.C. or D.C., in sizes up to 1¼ in. I.P.S., port sizes up to ⅝ in. K-20 Series is designed for applications where single needle port sizes provide sufficient flow capacity. W-3 Series are magnetic 3-way valves for controlling fluid to piston and diaphragm operators on valves, doors, gates, etc.

B-60 GAS ACTUATED PACKAGE SETS

Everything needed in a convenient package for remote gas control—B-60 Valve, Trimtherm Thermostat, Pilot Burner Generator and 25 ft of wire. No outside current required. Safe, quiet, dependable. Applicable to gas furnaces, floor furnaces, boilers, circulators, gas radiators. Operates on all types of gases.

*hi-g MAGNETIC VALVES

Type PV (AV)-3

Designed for positive operation on aircraft, trucks, tractors, tanks, graders, ships, and other moving equipment. Handle all fluids, vapors and gases on anything that rolls, floats or flies at pressures up to 3000 lb or more. Packless, two-wire, current - failure type, available normally open, normally closed for intermittent or continuous duty.

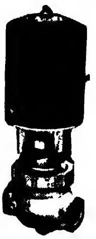
*Trade Mark—"hi-g" indicates positive ability to function in any position, regardless of vibration, change of motion or acceleration.

REFRIGERANT CONTROLS

Type K-15

Magnetic piloted, two - wire, current failure, high pressure, packless. Handle large capacities with minimum pressure drop and loss. Tight shut-off. Operates on wide variety of fluids and gases.

HYDRAMOTOR VALVES



Type G-1-7

Simplify valve control installations. Two-wire, current failure, electric - hydraulic operation. Ample motor - driven power, slow opening and closing movement. Operator drives against a spring in one direction; power failure or opening of circuit causes spring to operate in other direction.

MANUAL RESET VALVES

Type V-110

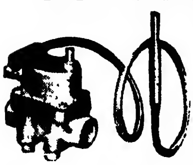
Equipped with manually-reset electromagnetically-held valve operator. Current flowing to operator permits manual opening by turning valve wheel at side. Current failure releases operator allowing valve to close. Trip-free mechanism cannot be opened under unsafe conditions. Once closed, valve is re-opened manually because applying current has no effect.

STRAINERS

Type S-5-1

Actual tests prove importance of strainers in prolonging operating life and reducing leakage of flow controls. S-5 Series STRAINERS come in 8 types; meshes 3/16 in. diam. to 120-per-inch.

THERMOPILOT (Valve Model)



Type MR-2

Manually-reset, electromagnetically-held-open valve with current generated by single couple subject to heat of pilot flame. Available $^3/_8$ in. to $1^1/_2$ in. I.P.S.

THERMOPILOTS

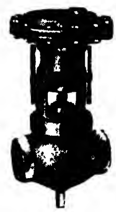
Type A-100

Proven principles of operation insure dependability. Flame applied to thermocouple maintains electrical contact, allowing electrical gas valve to open. When flame fails, Thermopilot opens circuit to main gas valve. Flexible, armored, asbestos-covered cable detachable from relay. 2 or 3-wire control. Electrical rating, 2 amp., 24 volt; 1 amp., 115 volt; 0.5 mp., 230 volt.

THERMAL EXPANSION VALVES

Type V-200 with new selective capacity cartridge provides instant sizing adjustment. Only one valve required for full capacity range in each body size at all back pressure or suction temperature ranges. For Freon, Methyl Chloride or Sulphur Dioxide.

GAS FUEL GOVERNORS



Type V-25-6

Throttle gas lines according to boiler pressure applied to diaphragm. Ball bearing thrust adjustment, ground and polished non-corrosive stems, low friction packing gland seal, multiple calibrated springs, high lift for maximum capacity. Suitable for butane, natural or manufactured gas. Available $^3/_8$ in. to 3 in., I.P.S.

RELAYS AND TRANSFORMERS

Type RS-100

Type RS-100 handles single phase motor loads up to 1 hp or heating loads up to 1.1 kw. Combines double-break relay and integral transformer. Normally open; large double - break contacts. Two-wire control circuit; maximum holding current 0.4 amps. Furnished with $^1/_2$ in. conduit connections and low voltage outlet. A.C. only.

TANK THERMOSTATS

Type L-61

Averages 6 to 10 degrees effective differential on water heater. Saves fuel by eliminating over-heating due to large differentials. Temperature range, 60 to 170 F. Also to 230 F.

LOW PRESSURE GAS REGULATORS

Type V-300

New V-300 Series are reliable, trouble-free valves with high capacity, close regulation, yet small and compact. Regulator size range from $^3/_8$ to 2 in. I.P.S. Internal parts, corrosion-resistant. Cast iron regulator bodies, long life calibrated springs, properly fitted all-metal valves.

GENERAL CONTROLS manufactures a complete line of Automatic Pressure, Temperature, Level & Flow Controls. For complete specifications request Catalog.

The Electric Auto-Lite Company

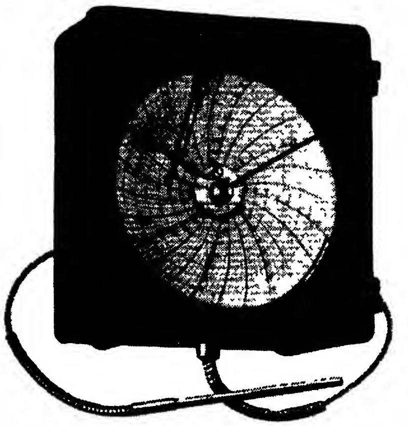
INSTRUMENT AND GAUGE DIVISION

AUTO-LITE

TOLEDO 1, OHIO

NEW YORK • CHICAGO • SARNIA, ONTARIO

TEMPERATURE INDICATING & RECORDING THERMOMETERS



Model 500 Recording Thermometer

Temperature cycles are permanently charted by the Auto-Lite Model 500 Recorder. Precision-engineered for accuracy, it has legible 6 in. chart. Uniformly spaced subdivisions insure accurate readings. The movement is liquid-filled and responsive to changes throughout the temperature range. Head is compensated for temperature changes.

All recorders are enclosed in dustproof and moistureproof aluminum cast cases. Movements and all actuating parts made of non-ferrous metals. With double braided flexible armored capillary tubing of bronze composition or $\frac{3}{16}$ in. semi-rigid copper tubing where capillary is to be immersed in liquids. Standard chart ranges between minus 40°F and plus 550°F.

Choice of 24-hour or 7-day mechanical clock movement. Complete with 100 charts, bottle of recorder ink, ink dropper. Wide choice of temperature ranges.

Model 500 is made in 3 standard types: WALL MOUNTING, with brackets for mounting; bottom connection. PORTABLE with spool-wound capillary, and strap handle. PORTABLE, SELF CONTAINED, with strap handle.

Model F-1 Indicating Thermometer

This thermometer is designed to facilitate systematic temperature observation for air conditioning, refrigeration or heating applications. It has large, easily read dial, evenly calibrated and fully compensated for temperature changes at the indicating head. Choice of temperature ranges between minus 40°F to plus 750°F.

Equipped with flexible capillary tubing for distance reading, or with rigid stem for direct mounting.

Auto-Lite's perfected one-to-one liquid filled movement eliminates delicate parts. Due to its strong construction, these instruments are particularly suited for installation on equipment where vibration is a factor.

Available with adjustable, electric alarm contact at small added cost.

Model F-1 — 5¼ in. dial width

Model F-2 — 4 in. dial width

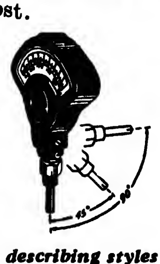


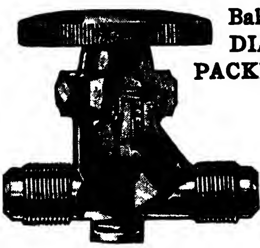
Diagram at right shows 3 positions at which Auto-Lite Indicating Thermometers may be mounted by screw adjustment.

Send for illustrated Catalog describing styles and types of Auto-Lite Thermometers, including detailed information on dial and chart ranges available.

Henry Valve Company

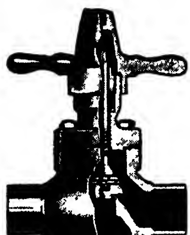
MELROSE PARK, ILLINOIS
(A Chicago Suburb)

HENRY PRODUCTS FOR REFRIGERATION, AIR CONDITIONING, AND INDUSTRIAL APPLICATIONS: CONTROL DEVICES, VALVES, STRAINERS, DRIERS, FITTINGS, AND ACCESSORIES.



Balanced-Action DIAPHRAGM PACKLESS VALVES

Line shut-off, branch shut-off, and angle types; flare, solder, or pipe connections. Hand expansion types also available. Forged brass body and bonnet, ports-in-line, non-directional stainless steel diaphragm. Stock sizes ¼″ to ⅝″ S.A.E., ¼″ to 1⅝″ O.D.S., ¼″ to 1″ F.P.T.



WING CAP PACKED VALVES

Globe and angle types, bronze with solder connections, semi-steel with solder and pipe connections. Floating, replaceable stem disc, backseating to permit repacking under pressure, wrench socket in top of wing cap operates valve. For "Freon" or methyl chloride. Sizes: Bronze globe, ⅞″ to 4⅛″ solder; bronze angle, ⅞″ to 3⅛″ solder; semi-steel, globe ½″ to 2″; semi-steel, angle ½″ to 2″; flanged semi-steel globe with tailpieces for 1⅝″ to 5⅛″ solder and for 1½″ to 5″ I.P.S. butt weld.

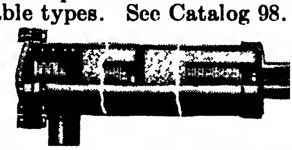
RELIEF VALVES

Diaphragm Type (Illustrated): For low-pressure refrigeration. Approved under safety codes. Diaphragm actuated. Sizes, ½″ F.P.T., ⅞″ solder. Solder extensions provided to protect internal parts from heat. Smaller relief valves also available. See Catalog 98.

Ferrous Type: For ammonia, "Freon," or methyl chloride. Approved under safety codes. Sizes ⅜″ to 2″ F.P.T. See catalog for illustrations and details.

DRIERS

Henry Abso-Dry Driers are charged with dehydrated air. When seal caps are opened at time of installation, hiss of escaping air proves drier is free of moisture or leaks. Perforated dispersion tube prevents "channelling." Brass shells and forged brass end caps. The 50 mesh and 30 x 250 Dutch Weave monel screen outlet filter will not collapse under any system pressure. Refillable and non-refillable types. See Catalog 98.

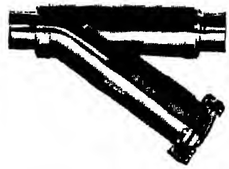


Type 756 (Illustrated): Cartridge-type drier with side outlet, compression spring, dispersion tube, safety cylinder and distortion-proof access flange. Dehydrant volume capacity 12 to 500 cu in. Sizes ⅜″ to 2⅛″ O.D.S.

STRAINERS

Henry strainers feature large capacity and easy cleaning brass shell with forged end caps, silver brazed construction. See Catalog 98.

Type 866: Angle strainer with solder connections, reinforced 100 mesh monel metal screen, drawn brass shell, distortion-proof access flange. Screen area 23 to 105 sq in. Sizes ⅜″ to 2⅝″ O.D.S.



Type 895: "Y" strainers have steel welded shells with forged brass connections for solder lines and steel connections for F.P.T. applications. Brass plated. Welded construction, distortion-proof access flange, protective baffle on inlet. Screen area 23 to 175 sq in. Capacity and connections for ⅝″ to 4⅛″ O.D.S.; 1″ to 3″ pipe.

Sold and Recommended by Leading Refrigeration Jobbers in 130 Cities

Hubbell Corporation

319 N. Albany Ave.
Chicago 12, Ill.

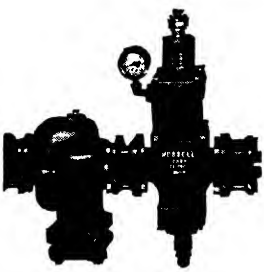
Designers and Manufacturers of Automatic Control Valves For All Refrigerants

Type SA-5, SF-5

Type SA-5, SA-6, SF-5 and SF-6 Back Pressure Regulating Valves are of the conventional type used to maintain a constant evaporator pressure.

Type SA-7, SA-8, SF-7 and SF-8 Combination Back Pressure Regulator and Stop Valves. This regulator is of the conventional type used to maintain a constant evaporator pressure and the addition of a small electric pilot valve built into the head, makes it a suction stop valve.

The DSA-9 and DSF-9 is a dual regulator which will control evaporators with two load conditions requiring different refrigerant temperatures. The diaphragms in the dual head may be set for any two evaporator pressures and will automatically change from one to the other by the opening or closing of the electric pilot valve which is built into the head.

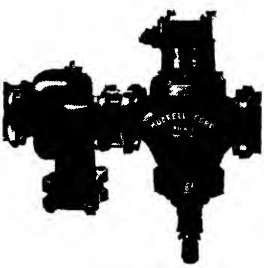


Type SA-6, SF-6

The SAC-6 and SFC-6 valves are of the compensating type and are used where a constant temperature is desired in the medium being cooled. These valves will increase or decrease the evaporator pressure to compensate for the increase or decrease of the cooling load. These valves are made to operate with air or electricity.

The Type "T" suction stop valve is used on installations where automatic suction line control is required. It is operated by high pressure gas and its construction makes a tight closing valve and its dependability far surpasses the conventional magnetic stop valve.

All of the valves have built-in opening stems, which eliminates the by-passes usually used for manual operation. Available with either screwed, welding type flanges or copper tube connections, in sizes from $^3/_4$ in. to 8 in. inclusive.

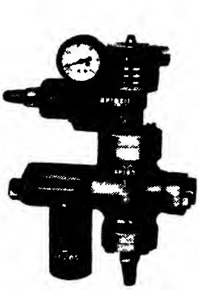


Type "T"

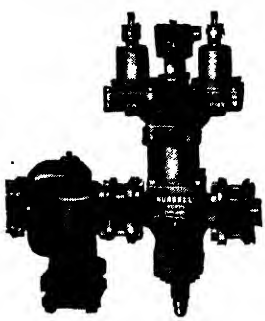
Solenoid Valves from $^3/_8$ in. to 2 in. inclusive for liquids and gases with composition seat discs readily renewable, coils for any electrical characteristics, built-in lifting stem standard, available with either screwed, welding flanges or copper tube connections.

Strainers are available in all sizes for liquid and gas with very large screen areas and arranged to bolt directly to valve or with screwed, welding flange or copper tube connections for installation wherever strainer is necessary.

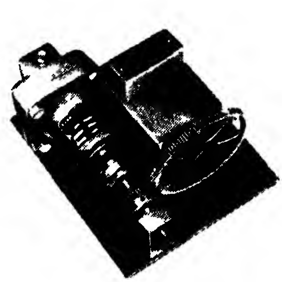
Write for complete information on these and our many additional Refrigeration Controls and Accessories.



Type SA-7, SF-7



Type DSA-9, DSF-9



Type SAC-6, SFC-6

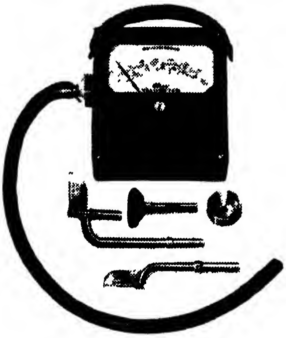
Illinois Testing Laboratories, Inc.

Room 516, 420 N. La Salle St., **Chicago 10, Ill.**

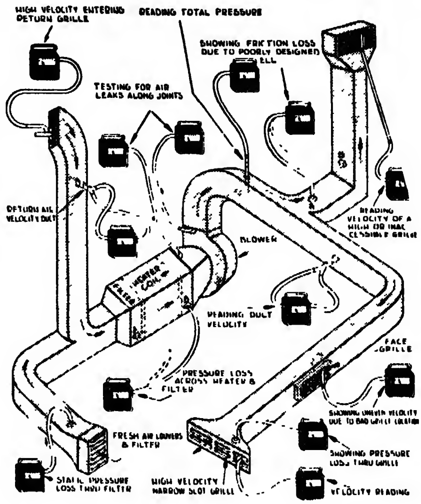
Alnor

Precision Instruments for *Every* Industry

The direct-reading ALNOR VELOMETER

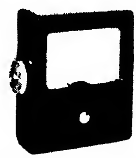


TYPE 4-F Standard minimum set for heating and air conditioning air velocity measurements.

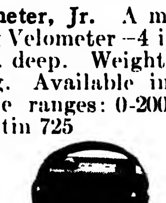


The Alnor Velometer is an instantaneous, direct-reading air velocity meter designed for convenient, rapid determination of air velocities in air conditioning, heating and ventilating, and exhaust systems. It gives instantaneous direct readings in feet per minute, without timing, calculations, or reference to tables or charts. Accurate information on performance of equipment, duct systems, etc., can be obtained with a few moments' inspection with the VELOMETER. It can be effectively used to locate drafts and leaks around windows and doors, or in duct systems.

The Alnor Velometer is built in several standard ranges from 20 fpm to 6000 fpm, and up to 3 in. static or total pressure. Special ranges available as low as 10 fpm and up to 25,000 fpm velocity and 20 in. pressure.



Alnor Velometer, Jr. A miniature, direct reading Velometer—4 in. high, 3 in. wide, 1-½ in. deep. Weight, 8 oz. Accurate, strong. Available in single and double scale ranges: 0-2000 to 0-2500 fpm. Bulletin 725



Alnor Thermo-Anemometer. For accurate measurement of low air velocity. Compact, direct-reading, battery operated, self-contained, portable. Scale 6 in. Meter range, 0-600 fpm. Accurate readings as low as 5 fpm. Temporary Bulletin 913-A.

Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Office and Factory

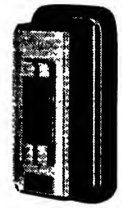
Milwaukee, Wis.

Direct Branch Offices in Principal Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 570 QUEEN ST., E., TORONTO, ONT.
HALIFAX, N. S. MONTREAL, QUE. WINNIPEG, MAN. CALGARY, ALTA. VANCOUVER, B. C.

PRODUCTS AND SERVICES

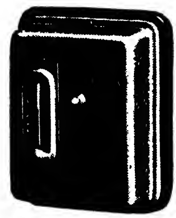
Manufacturers, Engineers and Contractors for automatic temperature and humidity control systems applied to all types of heating, cooling, ventilating, air conditioning and industrial processing installations.



Single Room Thermostat T-401

Space Control—Automatic control of room temperatures and humidities, applied to radiators, radiant heating, unit ventilators, unit heaters and heat delivery ducts. Johnson "Duo-Stats" maintain proper relationship between outdoor and heating system temperatures for groups of radiators, or "heating zones." A complete line of controllers for air conditioning systems, heating, cooling, humidifying, dehumidifying.

Process Control—Automatic temperature and humidity control for every range required in manufacturing and industrial processing. Thermostats, valves and dampers applied to tanks, dryers, vats, kettles, curing rooms, coolers, kilns, etc., in textile, rubber, pulp and paper, petroleum refining, meat packing, dairying, baking, sugar refining, brewing and distilling, tanning, candy making and other industries.



"Dual" Room Thermostat T-460

Nation-wide Service—Johnson sales engineers, and trained installation men available at all branches listed above. None is an agent, jobber, or part-time representative. All are salaried employees, devoting their efforts to the interests of the Johnson Service Company and its customers.

Send for Bulletins describing Johnson controllers.

JOHNSON THERMOSTATS

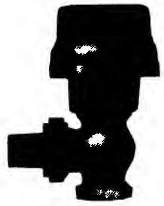
Room Thermostats—Proportional (gradual) or two-position (positive) action, maintaining temperatures within one degree above or below point of setting. Various covers, allowing wide selection of adjusting features, guards and method of mounting. Red-reading thermometers with magnifying tube attached to covers.

Room Humidostat H-107

Insertion and Immersion Thermostats—Rigid stem or capillary. Liquid or vapor-filled capillary systems for temperatures which are measured at point remote from location of operating mechanism. Various types of bulbs. Standard connecting tubing lengths: 8, 15 or 25 ft. Up to 50 ft on special order.

Thermometers—High grade insertion or immersion thermometers to measure temperatures in ducts, tanks, etc., with red-reading mercury column in heavy lens glass tube and 9-in. scale. Insertion thermometers have patented adjustable tilting feature.

Special Controllers—For applications encountered in industrial processes. "Record-O-Stat," combination capillary temperature controller and recorder. 12-in. chart and liquid or vapor-filled capillary systems. Single or duplex type, the latter controlling and recording wet and dry bulb temperatures . . . **Pressure Regulators**—Pressure ranges 27 in. of vacuum to 150 lb pressure. Bourdon tubes of types and sizes for required pressure range and for medium to be controlled: Air, water, steam or freon. Other regulators for pressures between ½ lb and 200 lb . . . **Liquid Regulators** (Float type)—Control within extremely close limits. Mounted through wall of containing vessel by stem with 1 in. pipe thread. Floats of copper, stainless steel or special alloys . . . **Static Pressure Regulator**—Measures variations in pressure from .009 in. to 3 in. of water. Also used as differential regulator, measuring difference in pressure between two chambers.



"Sylphon" Radiator Valve V-111

Rigid Steam Insertion Thermostat T-802

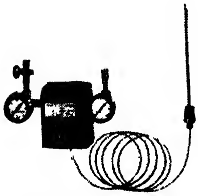
Sub-Master Thermostats—An important development for industrial applications and air conditioning. Applied to various controllers where readjustment must be made from a remote point.

Johnson Sensitivity Adjustment—A distinctive feature affording convenient means of adjusting the sensitivity of thermostats and humidostats, on the job, balancing "time-lag" with respect to capacity of conditioning apparatus. "Hunting" and temperature fluctuations prevented. Available on Johnson proportional action insertion and immersion thermostats, insertion humidostats, T-800 and T-900 capillary thermostats and certain room thermostats and humidostats.

JOHNSON HUMIDITY CONTROL

Johnson Humidostats—Automatically control supply of moisture delivered to air by a humidifier or other means, maintaining constant relative humidity. Available in room and insertion patterns with various elements, the most sensitive controlling within 1 per cent at relative humidities as high as 95 per cent at 100 F. Humidostatic elements are wood cylinder, by-wood strip, bow-wood, horn, hair or animal membrane.

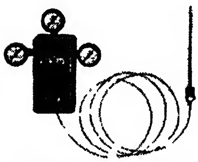
Johnson Humidifiers—"Steam grid" type (perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass heating coils and float control.



Capillary Thermostat T-800

JOHNSON VALVES

Johnson Diaphragm Valves—Simple and rugged. Diaphragms of special molded rubber, resistant to age and oxidation, operate valve stems against pressure of dependable springs. Available also with Sylphon seamless metal bellows. Made in standard sizes and patterns. Normally open (direct acting) or normally closed (reverse acting). Three-way mixing and by-pass valves, for steam, water, brine and other gases and liquids.



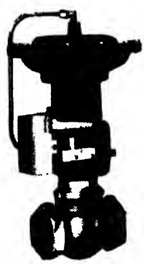
Sub-Master Capillary Thermostat T-901

Johnson "Streamline" Diaphragm Valves—With modulating discs and special internal construction. Superior proportional control . . . Where maximum power is required for repositioning at slightest demand of controlling instruments. Johnson molded rubber diaphragm valves are fitted with pilot positioner, independent of friction and pressure variations.

JOHNSON DAMPERS AND SWITCHES

Standard Johnson Dampers—Steel blades in flat steel frames with adequate bracing to form rigid assembly. Black lacquer or special corrosion-resisting finishes. Angle iron frames optional.

Special Dampers—Galvanized iron, monel metal, aluminum, copper, rust-resisting steel, etc. Brass pins in steel bearings or ball bearings.

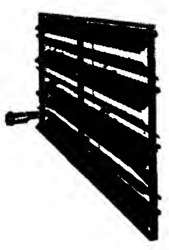


Globe Valve with Pilot Positioner V-103

Johnson Damper Operators—Similar in principle to valves. Seamless metal or specially molded rubber diaphragm operates damper through suitable linkage. Johnson "Piston" damper operators afford long travel at full power. With or without pilot mechanism, as described for "Valves."

Johnson Pneumatic Switches—For operation of dampers and to place controllers in and out of service, from remote points. Oiled slate is standard. Ebony asbestos, polished oak, and genuine or imitation marble on order. Various apparatus, at central control points, is mounted on special switchboards, including lever type switches, gradual and multiple-step switches, clocks, air pressure gauges, recording gauges, etc.

3-Way Mixing Valve (Rubber Diaphragm) V-108



Proportioning Louver Damper D-225

Piston Damper Operator D-251

The Mercoid Corporation

Main Office and Factory, 4201 Belmont Avenue, Chicago 41, Illinois
New York Office, 205 East 42nd St. Philadelphia Office, 3137 N. Broad St.

AUTOMATIC CONTROLS FOR HEATING, AIR CONDITIONING, REFRIGERATION AND VARIOUS INDUSTRIAL APPLICATIONS

THE 100% MERCURY SWITCH EQUIPPED CONTROLS

The distinguishing feature of Mercoid Controls is the exclusive use of Mercoid hermetically sealed mercury switches. These switches are not subject to dust, dirt or corrosion, thereby assuring better performance and longer control life. The items shown below are but a few miscellaneous items. See Catalog No. 700 for complete line.

FOR PRESSURE & TEMPERATURE

These controls have a wide range of applications. They are noted for their accuracy and dependable performance. The outside double adjustment feature and visible dial eliminate all guesswork when setting the operating range.

VISAFLAME

The Mechanical Eye Actuated by Light. A positive safety control system for domestic and industrial oil burners. Operates direct from the light of the flame instead of from heat in the stack. Tried and proven method for dependable oil burner performance.

OIL BURNER SAFETY CONTROLS

Type JMI provides positive protection against flame or ignition failure on intermittent ignition oil burners. This control insures having ignition circuit closed before every starting operation of burner. Type JM is used for constant ignition burners.

MERCOID THERMOSTATS

Mercoid low voltage room thermostats are known by their trade name Sensatherm. They operate on a total differential of 1 degree F. Type H is the regular popular room thermostat. Type DNH is a hand wound day and night Sensatherm. Type HBH is the two-stage type for control of high-low oil or gas burners. Recommended also for stokers. Type 855 is designed for direct line voltage applications recommended for unit heaters, air conditioning, etc.

LOW WATER CONTROLS

Available also as a combined low water and pressure control to prevent firing into a dry boiler or building up excess pressure. There are a number of different types for various requirements. The illustration shows type provided with quick-hook-up fittings designed in accordance with the *A.S.M.E.* code.

STOKER TIMER CONTROLS

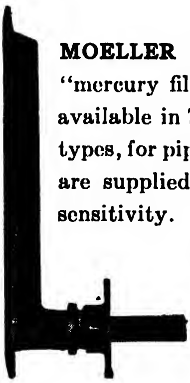
Type TV2 Stok-A-Timer combines a Mercoid Transformer-Relay and a synchronous motor timer mechanism for maintaining the stoker fire during periods when thermostat is not calling for heat. Interval adjustment can be set for ½ hour or 1 hour merely by moving a lever.

Moeller Instrument Company

132nd St. and 89th Ave., Richmond Hill 18, New York

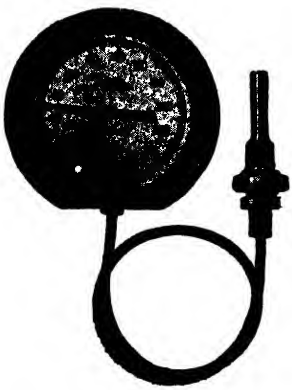
Representatives in Principal Cities

INSTRUMENTS FOR HEATING AND VENTILATING



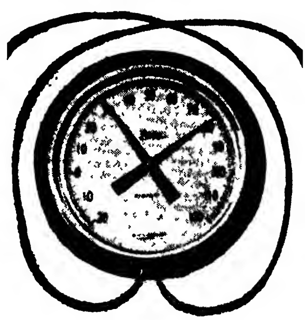
MOELLER BRASS CASE INDUSTRIAL THERMOMETERS "mercury filled" with MOELLER glass red reading column are available in 7, 9 and 12 in. cases, fixed, union or separable socket types, for pipe line and tank applications. Air duct thermometers are supplied with flanged union connection and bare bulb for sensitivity.

MOELLER DIAL AND RECORDING THERMOMETERS are mercury actuated, built for accuracy and long life. Dials are available in 4½, 6 and 8 in. sizes, single or duplex type. Recorders in 10 and 12 in. sizes, in 1, 2, 3 and 4 pen types.



Remote Reading Type, mercury actuated with separable socket connection. Back connection or flush mounting.

Single and multiple pen, Remote Reading types in rectangular or round cases.



Duplex mercury actuated, plain bulb type indicates temperatures at two separate points.

Send for catalogs and literature on

INDUSTRIAL ● LABORATORY AND RECORDING THERMOMETERS
THERMOSTATS ● HYGROMETERS ● HYDROMETERS
PSYCHROMETERS and MARINE SPECIALTIES

MINNEAPOLIS-HONEYWELL REGULATOR CO.

2958 Fourth Ave., S., Minneapolis 8, Minn. Cable Address: MINNREG, MINNEAPOLIS

HONEYWELL CONTROLS for Heating, Ventilating, and Air Conditioning

BROWN INSTRUMENTS for Indicating, Recording, and Controlling

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AT YOUR SERVICE with automatic controls for every application. Minneapolis-Honeywell manufactures a complete line of electric, pneumatic, electronic, and self-contained controls and regulators for every type of heating, ventilating, and air conditioning installation. In addition, the Brown Instrument Division of Honeywell manufactures a specialized line of indicators, recorders, and controllers. This means that you can rely on a single responsible manufacturer for all of your control needs. It eliminates the difficulties and misunderstandings that often result from split responsibility when controls are purchased from more than one source.

Each Honeywell branch office maintains a staff of experienced engineers who are qualified to give unbiased advice on control applications and to install and service all types of control equipment. They are prepared to assist in the writing of specifications and to furnish control layouts and cost estimates without charge.

Modutrol Valve

ELECTRIC CONTROL

Honeywell electric controls are noted for variety, versatility, dependability and precision operation. The trade mark "Modutrol" is used to designate Honeywell electric control systems designed for air conditioning or heating applications (other than domestic). It is your guarantee of Honeywell quality. A wide variety of both modulating and two-position motors, controllers and valves provide a flexible selection of control equipment.

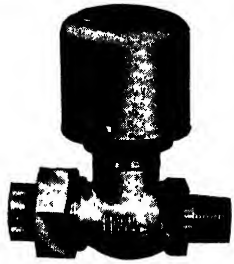
Electronic Thermostat

ELECTRONIC CONTROL

Now, because of the many special features of Honeywell Electronic Control, it is easy to achieve results which previously would have been extremely difficult or impossible. Electronic controls are super-sensitive and accurate. They are simple in operation and very flexible in application. A single electronic thermostat having no moving parts can be used to control both heating and cooling. The control settings can be maintained constantly or may be reset automatically. An almost unlimited number of averaging and compensating controls can be used to obtain practically any desired result.

PNEUMATIC CONTROL

The "Gradutrol System" designation is applied to any combination of Minneapolis-Honeywell automatic pneumatic controls used to govern the operation of air conditioning or heating systems. This equipment may be used to obtain either two-position or modulating control in any desired sequence. Such features as infinite positioning under full power with the Gradutrol Relay and accurate graduation of valve and damper motor position make the Gradutrol System a truly remarkable advance in pneumatic control



Radiator Valve

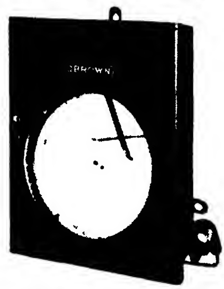
COMBINATION SYSTEMS

The outstanding advantages of the Honeywell pneumatic Gradutrol System, the electric Modutrol System, and the Electronic Control System may be combined in a single installation. Thus, maximum flexibility, low cost installation and dependable control results can be obtained. Honeywell can furnish controls for any particular type of system or for any combination of systems. This is your guarantee of fair and unprejudiced engineering advice as to the type of control best suited to your needs.

Electric-Pneumatic Relay

BROWN INSTRUMENTS

To obtain best results from modern heating and air conditioning equipment, it is necessary that the engineer in charge have a visual picture of actual conditions at all times. Brown instruments are designed to supply this information and to help operate the system at its highest efficiency. A complete line of Brown instruments is available for measuring, controlling, and recording temperature, humidity, flow, pressure, rpm, liquid level, pH, and fire safety.



Recording Thermometer

FREE HONEYWELL LITERATURE

Following is a list of free pamphlets on various applications of Honeywell controls. This list does not cover the complete Honeywell line but rather is limited to general applications. If you have a special control problem or require information on a specific type of instrument, please write, stating your needs, and we will try to furnish the information required. Remember, too, that friendly Honeywell representatives in offices throughout the nation await your call and will be glad to help you at any time.

ZONE CONTROL

UNIT HEATER CONTROL

RADIANT PANEL HEATING

CONTROLS FOR SCHOOLS

CONTROLS FOR HOSPITALS

ZONE CONTROL FOR HOMES

SPRING-RETURN VALVES

ELECTRONIC MODUFLOW FOR HOMES

ELECTRONIC AIR COND. CONTROL

CONTROLS FOR APARTMENTS

WEATHERSTAT OUTDOOR CONTROL

AQUATROL FOR HOT WATER SYSTEMS

HEATING AND AIR COND. CONTROLS

BROWN INDUSTRIAL INSTRUMENTS

Send your request for literature to the Minneapolis-Honeywell home office at Minneapolis or to your nearest branch office.

Penn Electric Switch Co.

Main Office—Goshen, Indiana; In Canada—Penn Controls, Ltd.

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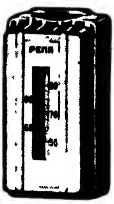
Atlanta; Berkely; Chicago; Cleveland; Dallas; Dayton; Denver; Detroit; Los Angeles; Minneapolis; Milwaukee; Moline; Newton, Mass.; New York; Philadelphia; Rochester; Salt Lake City; St. Louis; Seattle; Export—13 E. 40th St., New York 16, New York

Distributors and Jobbers in All Principal Cities

Automatic Controls for Heating, Refrigeration, Air Conditioning, Engines, Pumps and Air Compressors

Oil Burner Stack Switches



Thermostats

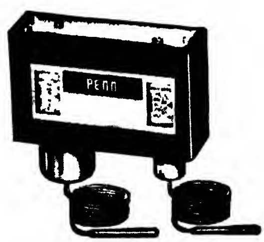
HEATING CONTROLS

A wide selection of controls is available for automatic heating service on steam, vapor, hot water, or warm air systems . . . gas, oil or coal-fired. Typical controls in Penn's complete line are: Low and Line Voltage Room Thermostats —Stoker Timers—Oil Burner Stack Switches —Solenoid Gas Valves—Hot Water and Warm Air Temperature Controls (including a new line of Liquid Expansion Fan and Limit Controls)—Vapor and Steam Pressure Controls—Relays and Relay-Transformers—Humidistats —Day-Nite Tem-Clocks —Damper Motor Controls—and Solenoid Valves. For complete descriptions and specifications, write for free catalogs.

New Warm Air Fan and Limit Controls

Stoker Timers

New Series "270" Low Side Pressure Control

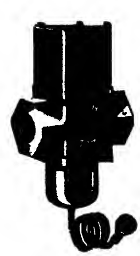


New Series "271" Dual Pressure Control

REFRIGERATION CONTROLS

A complete line of automatic commercial refrigeration controls . . . in a wide choice of pressure and temperature models to fit every need. New Series 270 Heavy Duty Refrigeration Control features two-pole, load-carrying contact structure and direct-reading calibrated scales. Other controls in the refrigeration line include: Cooling Room Thermostats—Humidistats—Relays—Solenoid Valves—Line Starters and Motor Contactors in Size 0, 1, and 1½—and Water Regulating Valves in sizes up to 2½ in. I.P.T. Write for free catalogs which give complete descriptions and specifications.

Motor Starters and Contactors



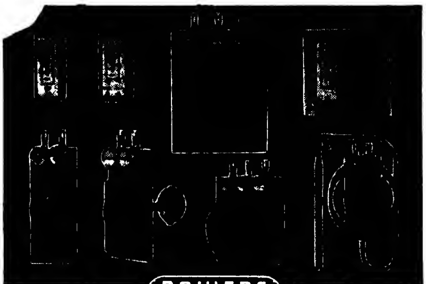
Water Regulators

THE POWERS REGULATOR CO.

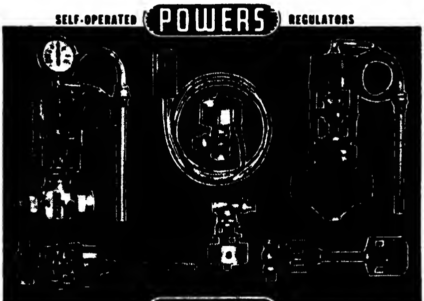
Over 55 Years of Temperature and Humidity Control—Offices in 47 Cities

NEW YORK 17, N. Y., 231 East 46th St.—Chicago 14, 2719 Greenview Ave.—LOS ANGELES 5, 1808 W. 8th St.—BOSTON 15, 125 St. Botolph St.—DETROIT 1, 320 McKerchey Bldg.—PHILADELPHIA 32, 2240 N. Broad St. —CLEVELAND 13, 2012 W. 25th St.—ST. LOUIS 3, 2118 Pine St.—PITTSBURGH 22, Oliver Bldg. —BUFFALO 2, 250 Delaware Ave.—NEW ORLEANS 12, 208-209 Vincent Bldg.—CINCINNATI 2, 702 American Bldg.—KANSAS CITY 8, 1510 Main St.—SEATTLE 1, 3150 Elliot Ave.—HOUSTON 6, 809 Stuart Ave.—ATLANTA 3, 142 Spring St. N.W.—DALLAS 1, 602 N. Akard St. —GREENSBORO, 733 Jefferson Standard Bldg.

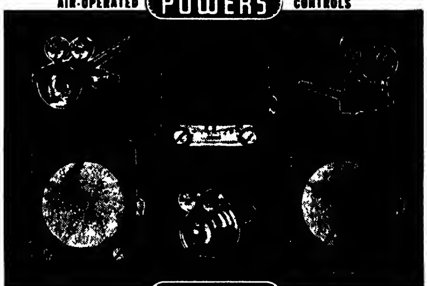
A very complete line of Temperature, Humidity, Pressure — indicating, controlling, and recording regulators —for heating and air conditioning systems, industrial processes and all types of hot water heaters.



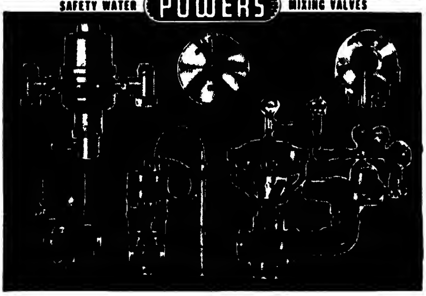
SELF-OPERATED POWERS REGULATORS



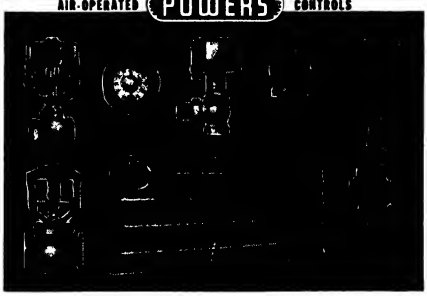
AIR-OPERATED POWERS CONTROLS



SAFETY WATER POWERS MIXING VALVES



AIR-OPERATED POWERS CONTROLS



Over 55 years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings have given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose.

Catalogs and Bulletins describing any or all of our products furnished upon request. Phone or write our nearest office. See your phone directory.

Spence Engineering Company, Inc.

28 Grant Street, Walden, N. Y.

SPENCE METAL DIAPHRAGM "DEAD END" REGULATORS

Advantages of Spence Regulators

Dead-end Shutoff—Spence Regulators are guaranteed to hold a dead-end.

Single Seat—Spence design makes possible a balanced single seat even in large sizes.

Metal Diaphragms—Under normal conditions never require replacement.

Accurate Regulation—Regardless of fluctuations in either load or initial pressure.

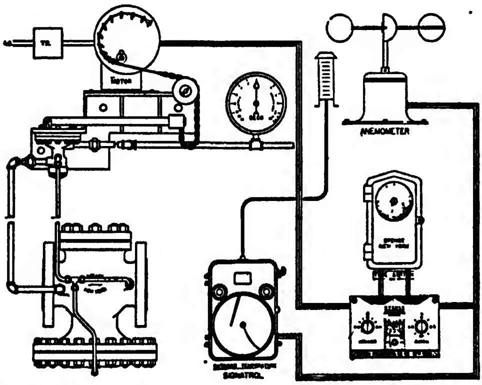
SECO Metal—Guaranteed to resist the wiredrawing action of steam.

Interchangeable Pilots—Any type of pilot will fit any size main valve.

Accessibility—Pilot is connected to main valve with unions.

No Stuffing Boxes—All main valves and most pilots are packless.

Spence Weather Compensator and Orifice Zone Control System



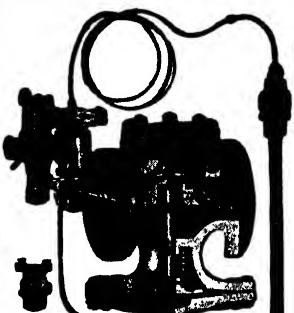
This simple, dependable Control, when installed on a properly designed orificed heating system, will show a substantial degree-day steam saving, at a low maintenance cost.

The delivery pressure of the Regulator is automatically adjusted in direct proportion to the building heat losses. In other words, as the losses become greater, steam pressure on the system is automatically increased.

Any number of zones can be controlled by one automatic Signatrol, automatic Wind Loss Compensator (Anemometer), Time Switch and Master Control Panel equipped with Manual and Automatic Dials for each zone. In this way each zone can be set individually and at the same time be under the Master Control.

Pressure Regulator—Type ED

Designed to regulate a steady or varying initial pressure so as to maintain a constant, adjustable, delivery pressure. Applicable to heating systems, power plant operations, or manufacturing processes.



Combined Temperature and Pressure Regulator —Type ETD

Self-contained, pilot operated, dead-end. Designed to control flow of fluid to a heating or cooling element, so as to maintain a constant, adjustable temperature, and protect the element against excessive pressure.

Electrically Operated Valve—Type EM

Can be opened or closed independently by an electrical switch.

Type ET—Same as ETD except pressure control is omitted.

Order a SPENCE Regulator for 40 days' free trial.

Fall-O-Matic Universal Pipe Intersection Cutter.

Sterling, Inc.

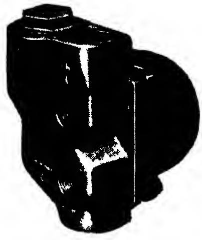
3738 North Holton Street • Milwaukee 12, Wisconsin

Sterlco

Heating and Temperature Control Products

Distributed through leading Heating and Plumbing Wholesalers
Sales Representatives in Principal Cities

HEATING SPECIALTIES



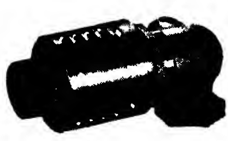
Thermostatic Traps: sizes ½ in., ¾ in., 1 in.; max. pressure 15, 65, 100, 125 psi; Angle, Straightway, Corner, Vertical Body Styles. Float and Thermostatic Traps: sizes ¾ in. to 2 in.; max. pressures 15, 100 psi. Strainers: max. pressure 125 psi; Cast Iron Sizes ½ in. to 2 in.; Brass Sizes ⅜ in. to 1 in. Also Vents, Radiator Valves, Boiler Return Traps.

Illustrated at right is the compact, sturdy ¾ in. 69B F & T Trap, which like all other Sterlco traps has a bellows type thermostat and replaceable seats and valve members.

CONDENSATION and VACUUM PUMPS

4100 Series (illustrated): neat, convenient unit with 15 gal tank, dual voltage capacitor type motor, and carbon type rotary seal—for all jobs up to 6000 sq ft EDR, 20 psi. 3500 Series: heavy duty units with pedestal mounted steel tank for a wide range of jobs—2000 to 65,000 sq ft, up to 150 psi. 3700 Series: with heavy cast iron tank for installation underground or in wet locations. 2000 to 20,000 sq ft, 20 or 30 psi. 3800 Series: up to 10,000 sq ft, 20 or 30 psi. Vacuum Pumps: Type V, 2500 and 5000 sq ft, 20 psi. Type S, 10,000 to 40,000 sq ft, 20 to 40 psi.

TEMPERATURE CONTROL VALVES



No. 120 Thermotrol (illustrated): self-contained individual radiator temperature control for steam or hot water. Easily installed in place of manual radiator valve without wiring or other external connections. Tank Temperature Controls: self-powered modulating control valves for fluid heaters and process equipment. Sizes: ⅜ in., ½ in., ¾ in., 1 in., 1¼ in., 1½ in., 2 in.

Taylor Instrument Companies

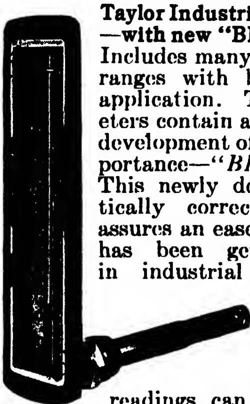
Rochester 1, N. Y., U. S. A.

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD., TORONTO

NEW YORK BUFFALO CLEVELAND HOUSTON BALTIMORE
CHICAGO LOS ANGELES ST. LOUIS CINCINNATI ATLANTA
BOSTON PITTSBURGH SAN FRANCISCO TULSA MINNEAPOLIS
PHILADELPHIA WILMINGTON SCHENECTADY

Manufacturing Distributors in Great Britain, Short & Mason, Ltd. London

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level

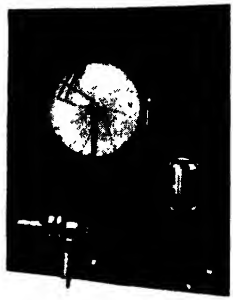


Taylor Industrial Thermometers —with new "BINOC" Tubing— Includes many styles and scale ranges with bulbs for every application. These thermometers contain a new and radical development of tremendous importance—*"BINOC" Tubing.* This newly designed and optically correct glass tubing assures an ease of reading that has been generally lacking in industrial thermometers. *BINOC Tubing* more than doubles the angle of vision within which readings can be made. Because of the patented triple-lens construction its broad mercury column can be read easily and accurately with both eyes. Bore reflection is absent.

Taylor Biram's Anemometer—For measuring air velocities with fan revolutions indicated on dial. Various models for a wide range of air speeds and registration limits.

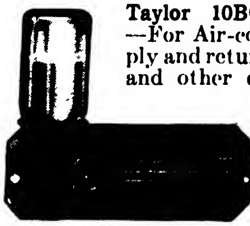
The Taylor "Fulscope" Recording Controller—An air-operated controller that gives practically any character of process control regardless of *time lag* in apparatus.

Available for controlling temperature, pressure, humidity, rate of flow, liquid level. Where extreme load changes or badly balanced operating conditions exist, precision control can be maintained by the automatic reset feature. For applications where a record is not needed, Taylor supplies an Indicating "Fulscope" Controller.



Taylor Recording Hygrometer—Records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

Type shown has motor-driven fan for conditioned rooms or passages where circulation is poor. Furnished without fan for installations where circulation across bulb is good.



Taylor 10BG Hygrometers—For Air-conditioning supply and return ducts, dryers, and other closed compartments where temperature and humidity readings are desired. Combines the accuracy of an etched-stem thermometer and ruggedness of an industrial thermometer. Easily installed. Available with bottle or constant automatic water supply.

Taylor Sling Psychrometer—Two accurate etched-stem thermometers mounted on die-cast frame, with the bulb of one covered with a wick to be moistened. Whirling bulbs subject this hygrometer to complete air contact to produce extreme accuracy of temperature and humidity measurement.

Taylor also offers a complete line of the famous Taylor Recording and Dial Thermometers; Ratio, Pneumatic Set, Self-Acting and Type "P" Controllers; Indicating Hygrometers and many types of Humidiguides.

United States Gauge

Sellersville 44, Penna.

Division of American Machine and Metals, Inc.

MANUFACTURERS OF PRESSURE, TEMPERATURE, FLOW AND ELECTRICAL MEASURING INSTRUMENTS

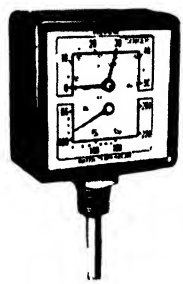
OFFICES AND DISTRIBUTORS  **IN PRINCIPAL CITIES**

MOST COMPLETE LINE OF ITS KIND

You will probably find in the U. S. Gauge line, the instrument you need for indicating or recording pressure, temperature, flow or electrical values.

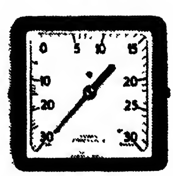
USG Instruments have a reputation for performing accurately, uniformly over a long period—as evidenced by the choice of experts · the manufacturers who install U. S. Gauges on over 6 out o every 10 pieces of original equipment The **USG** system of quality control an(40 years of instrument manufacturin experience is your guarantee of depend able service from instruments that al ways "TELL THE TRUTH."

COMBINATION THERMOMETER AND ALTITUDE GAUGE



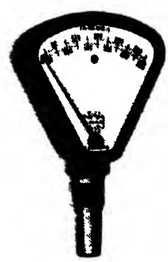
For use on hot water heating systems. Indicates on one dial: water temperature, head of water above gauge and pressure in system. Rugged construction with easy-to-read dial. Available in round or square case.

COMPOUND PRESSURE AND VACUUM GAUGE



Designed for installation on boilers, steam pressure, refrigeration, air conditioning and house heating systems, where it is desired to read pressure and vacuum on one dial. Available with or without an internal syphon, it has a sturdy corrosion resistant movement and a bronze bourdon tube. Furnished in dial sizes from 2 in. to 4½ in. Ranges from 15 to 300 lbs per sq in.

BI-METAL THERMOMETER



Attractive fan shaped dial thermometer, for installation on hot water boilers in house heating systems or other apparatus where sensitive bi-metallic element mounted in bulb can be used. Can be supplied in black. satin, chromium or nickel-finished case. Temperature range 60-260 F. Also made with back connection.

White-Rodgers Electric Company

1209 Cass Avenue, St. Louis 6, Mo.

AUTOMATIC CONTROLS

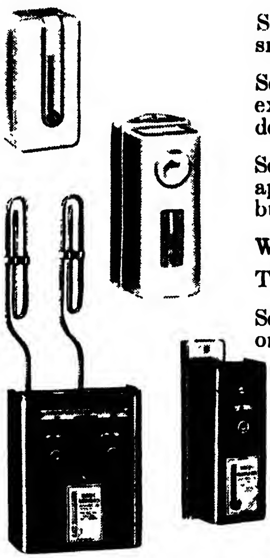
for

HEATING • AIR-CONDITIONING • REFRIGERATION

HYDRAULIC ACTION CONTROL

White-Rodgers controls with Hydraulic-Action have many features that assure complete accuracy and dependability in all applications—

- Maximum Sensitivity.
- Positive Settings.
- Full Range Accuracy.
- Consistent Differential.
- No Temperature Drift.
- Easily Adjusted.
- Extra Sturdy Switch.
- High Electrical Ratings.
- Versatility.
- Protected Switch.
- Not affected by Ambient Temperature or Atmospheric pressure.



THERMOSTATS

Series 120 and 130—Light-duty, low voltage thermostats smartly designed in Ivory and Chrome.

Series 150—Heavy-duty, line voltage. Hydraulic-Action, exceptionally rugged switch mechanism provides long, dependable service.

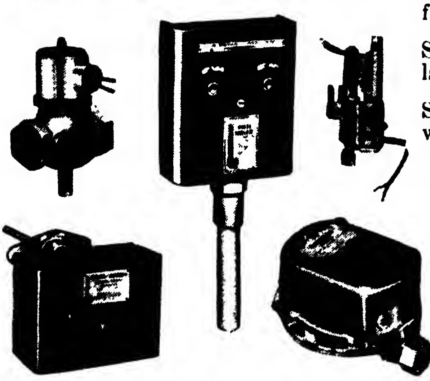
Series 200—Space Thermostats for heavy-duty, line voltage applications. Available in both self-contained and remote-bulb types.

WARM AIR, STEAM AND HOT WATER CONTROLS

Types for fan or circulator control, and for limit service.

Series 1100—Hydraulic-Action Hot-water controls in strap-on, vertical or angle immersion wells; single or dual types.

Series 400-500—Hydraulic-Action warm air fan and limit controls, single or dual types.
Series 1200—Steam and vapor pressure controls.



PRIMARY CONTROLS

Series 600—Oil Burner Primary controls featuring a new, extremely sensitive element. Available in one and two-piece types for continuous or intermittent ignition.

Series 2500—Solenoid Gas Valves for installations requiring small overall dimensions.

Series 2600—Diaphragm Gas Valves. Light weight, smooth operation, high capacity.

Series 2530—Combination Solenoid Gas Valve and Liquid Filled Pilot provides simplicity of installation.

Series 3000—Gas Safety Pilots. Provide exceptionally dependable performance. Also available with push button igniter.

Series 700—Stoker Timers, available for low voltage or line voltage.

AIR-CONDITIONING AND REFRIGERATION CONTROLS

White-Rodgers offers a complete line of Thermostats, Pressure and Temperature controls available in all needed ranges and types.

WRITE FOR CATALOGS GIVING COMPLETE INFORMATION

Aldrich Company

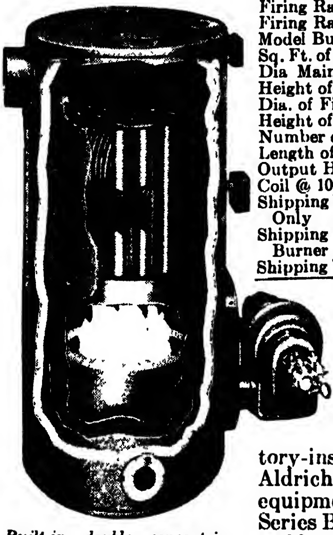
121 E. Williams St., Wyoming, Illinois

Boiler-Burner Units, Domestic and Commercial Oil Burners

Universal tapping on all models provides openings for water gage, side-arm heater (except on boilers ordered with coils), high limit controls, pressure gage and pop valve, two risers, and two returns.

•

Cold drawn seamless steel fire tubes, which are submerged their full length below water line, are readily accessible for inspection and cleaning.



Built-in double concentric spiral hot water coils (in Models SC and WC) provide adequate supply of hot water without external storage tank.

•

Rock wool insulation 2 in. thick provides high insulating value and minimizes heat loss through radiation.

•

Welded steel fire box is lined with cast refractory having high resistance to deterioration. Large burner-mounting plate provides access to fire box for replacement of combustion chamber.

•

Matched Aldrich Oil Burner provides efficient heat source. Standard automatic control system furnished as ordered to suit installation.

BOILER-BURNER UNITS

Sizes—Specifications—Dimensions

Size of Boiler	118	160	225	315	514	808
Rating Sq. Ft. Hot Water EDR	750	1000	1500	2100	3300	5000
Rating Sq. Ft. Hot Water Standing Radiation	500	650	1000	1400	2200	3333
Rating Sq. Ft. Steam EDR	500	630	935	1275	2025	3000
Rating Sq. Ft. Steam Standing Radiation	333	420	620	850	1350	2000
Rating BTU Per Hr. (Max.)	118,000	160,000	225,000	315,000	514,000	808,000
Water Heater Delivery GPH @ 100° Rise	125	190	280	450	610	850
Storage Capacity Gallons	28	38.5	72	99	120	170
Firing Rate—GPH Maximum	1.25	1.65	2.75	3.65	5.7	9.00
Firing Rate—GPH Minimum	.75	1.25	1.75	2.4	3.7	6.00
Firing Rate—GPH Best	1.00	1.35	2.00	2.5	4.5	7.5
Model Burner Furnished	SAX1	SAX2	SAX3	SAX3	BX	BX
Sq. Ft. of Heating Surface	21	27	44	67	93	136
Dia Main Shell (Inside)	19"	21"	24¼"	28"	32"	38"
Height of Main Shell	46½"	50½"	58½"	66½"	66½"	70½"
Dia. of Fire Box (Inside)	14¼"	16"	19"	22½"	27"	32½"
Height of Fire Box	24"	24"	26"	26"	26"	26"
Number of Tubes	16	20	30	42	60	94
Length of Tubes	15⅝"	19¼"	23⅜"	30⅝"	29⅝"	31⅝
Output Hot Water Coil @ 100° Rise WC—GPM	1.75	2.00	2.66	5.25	6.50	7.25
Output Hot Water Coil @ 100° Rise SC—GPM	2.00	2.75	3.67	6.75	8.00	10.00
Shipping Weight Lbs.—Boiler Only	775	940	1330	1720	1975	2350
Shipping Weight Lbs.—Burner Only	85	85	85	85	140	140
Shipping Weight Lbs.—Total	860	1025	1415	1805	2115	2490

Aldrich Series B Boiler-Burner Units—available in a range of 6 sizes for Hot Water Heating, Steam Heating, and Hot Water Supply—are designed for home, apartment, garage, club, restaurant, hotel, and factory installations. Gages, covers, and trim are furnished to suit model and type of unit ordered. Heavy-duty, double-spiral hot water coils are factory-installed and tested in SC and WC models. Matched Aldrich Oil Burner furnished with each unit. Standard equipment includes full set of basic automatic controls. Series BG Boilers are gas-fired with *AGA*-approved burners and have ratings identical to corresponding oil-fired boilers.

MODEL DESIGNATIONS—SERIES B AND BG

	Steam Heating	*Hot Water Heating*	*Hot Water Supply*
Without Coil	S	W	HSG*
With Coil	SC	WC	—

* Galvanized, with dress jacket. All models, except HSG, have universal tapping.

ALDRICH OIL BURNERS

Capacities from 0.75 gph to 19 gph

Models SAX-BX-JU

Aldrich Oil Burners are made in 3 sizes and 5 models for domestic and commercial warm air, steam, or hot water systems, and are UL-approved. All-steel model SAX (illustrated) has interchangeable draft tube components to permit 3 firing ranges—0.75 to 1.35 gph, 1.35 to 2 gph, and 2 to 4.5 gph. Aldrich-built motors and transformers are used. Model BX is rated from 4 to 8.5 gph. Capacity of Model JU is 7 to 19 gph.

AMERICAN RADIATOR & Standard Sanitary CORPORATION

Pittsburgh 30, Pa.

SEVERN BOILER FOR Coal (hand fired or stoker), or Oil

An efficient Boiler with many new features for convenience and economy. *Ratings:* Steam—350 to 780 sq ft, Water—560 to 1250 sq ft, installed radiation.

OAKMONT OIL BOILER

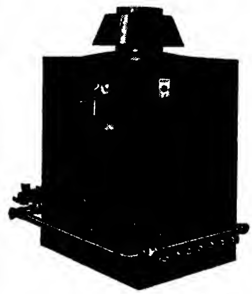
Exclusively for oil firing. Also supplied as complete boiler-burner unit with Arcoflame Burner. *Ratings:* Steam 390 to 810 sq ft, Water—625 to 1295 sq ft, installed radiation.

EXBROOK BOILER—For Oil or Stoker

A boiler in sizes adapted for larger than average homes and buildings. *Ratings:* Steam 775 to 1650 sq ft, Water 1240 to 2640 sq ft installed radiation.

ARCOLINER OIL BOILER

A boiler of advanced design for smaller homes. Wet base construction. *Ratings:* Steam—260 to 520 sq ft, Water—420 to 835 sq ft, installed radiation.



STANDARD GAS BOILER

Designed by experts to burn gas efficiently, and economically. Approved by *American Gas Association. Ratings:* Steam—400 to 12,420 sq ft, Water—135 to 19,860 sq ft, installed radiation.

REDFLASH BOILER For All Fuels

Economical heat for any size or kind of building. Ingot red jacket, fully insulated. *Ratings:* Steam—770 to 9900 sq ft, Water—1230 to 15,840 sq ft, installed radiation.

WATER TUBE BOILERS Stoker or Oil

For medium to large buildings. Efficient and economical. *Ratings:* Steam—930 to 4600 sq ft, Water—1490 to 7360 sq ft, installed radiation.

MOHAWK Winter Air Conditioner Gas-Fired

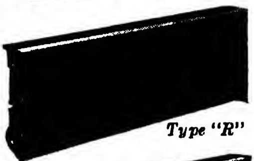
Provides automatic gas-fired Winter Air Conditioning in homes of any size. Capacities range from 43,200 to 216,000 Btu input per hour.

YOU CAN FILL EVERY HEATING NEED FROM THIS COMPLETE LINE

From **American-Standard** comes Boilers, Warm Air Furnaces and Winter Air Conditioners of all types and sizes for Coal—stoker or hand-fired—Oil, or Gas, and a full line of Radiators, Convectors, Oil Burners, Domestic Water Heaters, and Accessories —all backed by the undivided responsibility of one of the world's largest manufacturers of Heating and Plumbing Equipment.

WYANDOTTE
Winter Air Conditioner Gas-Fired

For small homes with or without basements. Factory assembled for units of 55,000, 70,000 and 85,000 Btu imput per hour. 105,000 Btu input per hour, non-assembled.



Type "R"

Type "RC"

BASE BOARD RADIANT PANELS

Made to replace ordinary baseboards. Available in two styles: Radiant type 1.25 sq ft per lineal foot. Convector type 2.08 sq ft.

Arco Radiator

ARCO RADIATOR

The modern, slim type radiator that occupies less space and gives more heat. It comes in four narrow widths and in four heights - 19, 22, 25 and 32 inches.

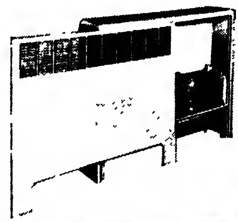
Sunrad Radiator

SUNRAD RADIATOR

Installed recessed or free-standing. Requires no enclosure. Two sizes: 20 in. high x 5 in. wide = 2 sq ft, and 23 in. x 7½ in. = 3 sq ft.

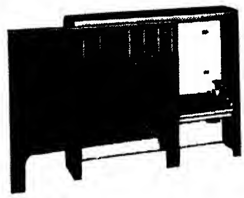
ARCO CAST IRON CONVECTOR

For convection heating. Available in three widths and in virtually any desired length.



ARCO MULTIFIN CONVECTOR

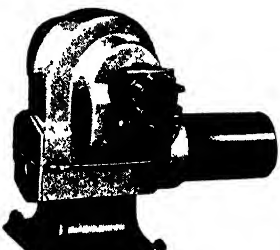
For all systems except one-pipe steam. Made in four widths. Non-ferrous.



ARCOFLAME OIL BURNERS

Approved by Underwriter's Laboratories

The Model "C" Arcoflame has a capacity of one to 3 gallons per hour. The Model "L" (not shown) from 3 to 7 gallons per hour. Both are highly efficient.



BUDGET GAS FIRED WATER HEATER

Heats water automatically, stores it for instant use. Jacket finished in white enamel, black trim. Thrifty and dependable. Three sizes —20, 30, and 40 gallons.

Budget Gas Fired

No. 861 Hurivent Valve (for mains)

No. 300 Multiport Adjustable Valve (for adiators)

No. 999 Arco Packless Steam Radiator Valve

No. 500 Airid for Rapid Venting (for radiators)

No. 5000 Variport Airid Adjustable Valve (for radiators)

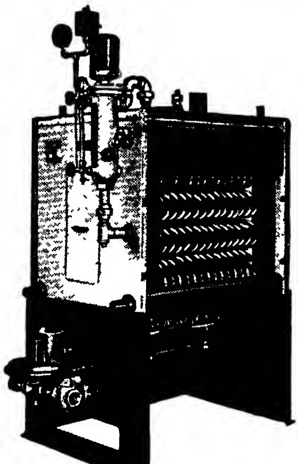
Bryan Steam Corporation

Chile Pike, Peru, Indiana

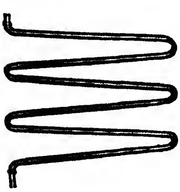
Manufacturers of

HOT WATER AND STEAM BOILERS DESIGNED EXCLUSIVELY FOR OIL OR GAS FIRING

PRODUCTS—Domestic Hot Water or Vapor Steam Boilers.
Commercial Low Pressure Heating Boilers for Hot Water or Steam
Non-explosive High Pressure Boilers up to 50 H. P.



Bryan Copper Tube Boilers are engineered expressly for oil or gas firing, where the flame is either on at full intensity or entirely off. Such combustion characteristics call for a boiler that is able to capture heat units with utmost rapidity—and able to withstand sudden expansion and contraction. Average stack temperature of the Bryan is 412 degrees.

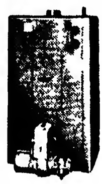


BRYAN COPPER TUBES

The Bryan starts with copper tubes. Copper has a coefficient of heat transfer approximately 6 times that of iron or steel. Heat applied to the outside of a Bryan tube is therefore transferred to the water inside 6 times as fast.

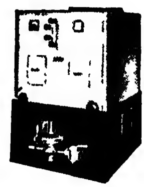
The design of the Bryan tube is such that water circulates rapidly through all parts of the boiler. There is a veritable nest of tubes directly over the flame where heat is most intense. These tubes break up the path of heat travel, reducing "surface film" to a minimum.

A Bryan Copper Tube Boiler never explodes. The worst that can happen, is a split tube which may be replaced in a few minutes time by an inexperienced person.



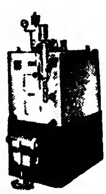
DOMESTIC BOILERS

The Bryan Domestic Boiler is made in 5 sizes that will supply steam radiation of from 375 to 1500 sq ft. They have firing rates of from 1.25 gal oil (or 175 cu ft natural gas) to 5 gal oil (or 700 cu ft of gas). All five models come complete with built-in combustion chamber and flange mounted oil burners. In gas fired models the burner is built in.



COMMERCIAL HEATING BOILERS

Any of the domestic Bryans may be used for small offices or buildings; but they are supplemented with three additional larger capacity boilers. These boilers are rated 2250 to 4400 sq ft of steam radiation or 3600 to 7050 sq ft of hot water radiation. They are used also for supplying hot water or low pressure steam in industrial applications.



HIGH PRESSURE BOILERS

Bryan High Pressure Boilers are made in 5-10-20 and 50 hp ratings. Every one carries the *A.S.M.E.* stamp. They are made for high efficiency on long, hard pulls but are exceptionally useful in operations requiring fast, safe steam on short notice. Hospitals, dry cleaning plants, laundries, milk plants, tire repair shops and many others find them ideal for their operations.

Burnham Corporation

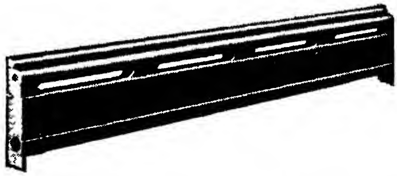
BOILERS and RADIATORS

Irvington-on-Hudson, N. Y.

***There's a Burnham for Every Purpose*—Catalog No. 81 Sent on Request**

BASE-RAY Radiant Heating with Radiant Baseboards

STANDARD Model BASE-RAY—Ratings—1.25 sq ft per lineal foot. Tappings—¾ in. at bottom only of both end sections. Sections are 7 in. high, 1¾ in. thick and in 12 and 24 in. length.



HY-POWER Model BASE-RAY—Ratings—2.08 sq ft per lineal foot. Tappings—¾ in. at bottom only of both end sections. Sections are 7 in. high, 2 in. thick and in 12 and 24 in. length.

No. 1, 2, 3, and 36 in. Series—All Fuel. 230 to 4920 sq ft Steam and 370 to 7880 for Water.

Burnham Radiant Radiator—Two heights. 20 and 23 in.

Burnham Slenderized Radiators (not shown) made in three to six tubes in all heights from 19 in. to 33 in.

YELLO-JACKET Boiler (with extended Jacket)—All Fuel Convertible. 305 to 935 sq ft for Steam and 490 to 1495 sq ft for Water.

Welded Steel Boiler—Capacities from 2500 to 35,000 sq ft for Steam and 4800 to 56,000 sq ft for Water. Furnished for coal, oil or stoker firing.

50 in. Twin-Section—4500 to 14,600 sq ft for Steam and 7200 to 23,360 sq ft for Water.

Crane Co.

BOILERS, RADIATORS, FURNACES, VALVES, FITTINGS, PIPE, WATER AND STEAM SPECIALTIES, CONTROLS AND PLUMBING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago 5, Illinois**

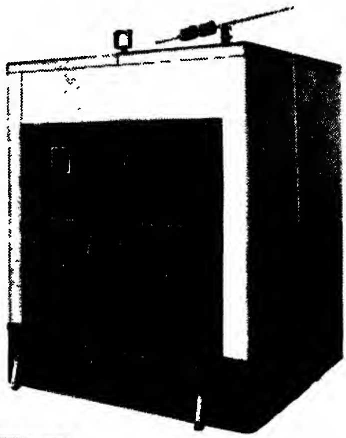
Nation-Wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors

CRANE OFFERS EVERYTHING IN HOME HEATING

There's a Crane Boiler to fit any home—regardless of its size—whether coal, coke, oil or gas is the preferred fuel or whether the owner wants steam or hot water heating. These boilers are styled for beauty, dependability and simplicity, and they incorporate many new engineering features that assure greater fuel economy and operating efficiency.

Crane also builds boilers for many types of non-residential buildings. In addition, the Crane line includes everything else needed for heating systems—furnaces, stokers, oil burners, radiant baseboard panels, radiators, convectors, controls, water and steam specialties, pipe, valves and fittings.

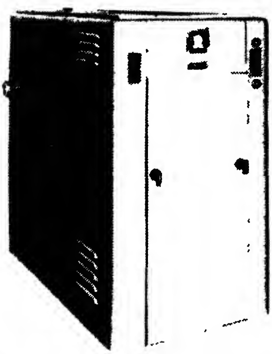
For full information on Crane Boilers and other heating equipment, consult your Crane Branch or Crane Wholesaler.



CRANE FORTY BOILER

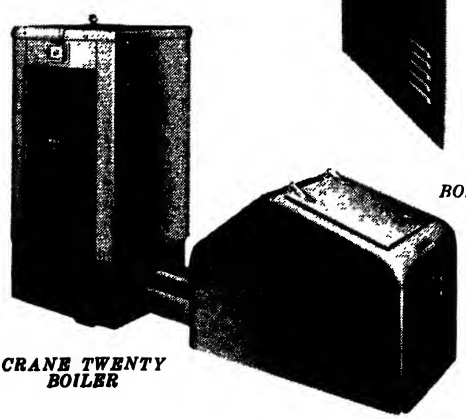
CONTROLS

The complete line of controls includes thermostats, stack switches, limit switches, aquastats, damper controls—everything necessary for a heating system.



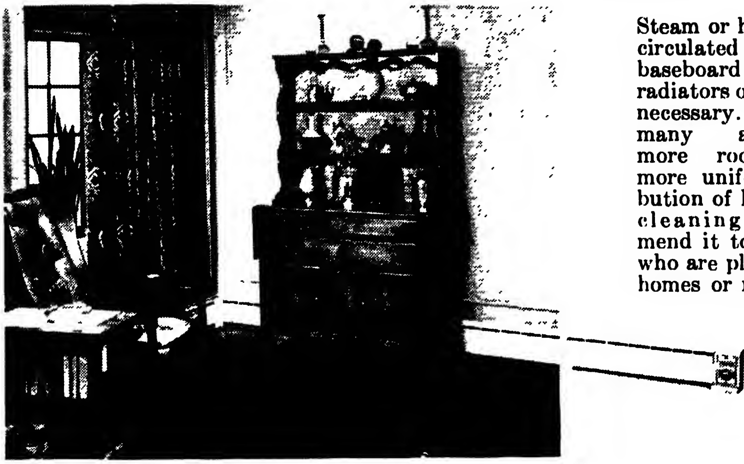
CRANE SIXTEEN BOILER—BURNER UNIT

CRANE FOURTEEN BOILER



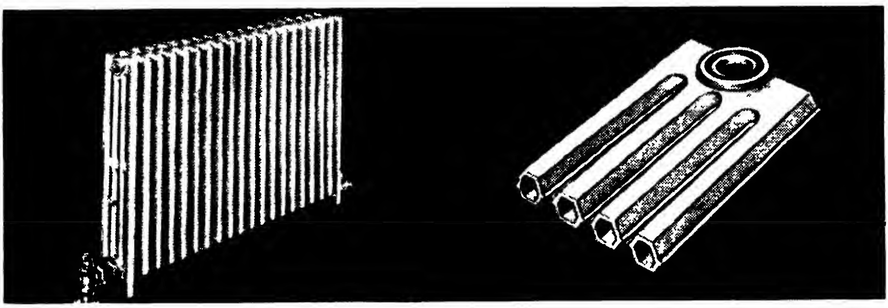
CRANE TWENTY BOILER

CRANE RADIANT BASEBOARD HEATING

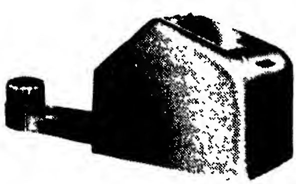


Steam or hot water is circulated through the baseboard panel—no radiators or grilles are necessary. Will give many advantages: more room space; more uniform distribution of heat; easier cleaning. Recommend it to customers who are planning new homes or remodeling.

CRANE RADIATORS

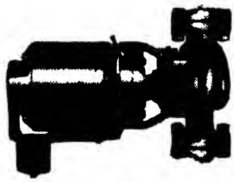


COMPAC RADIATORS: Slender in line—modern in design—give more heat—use less space. For free-standing or recessed installation, steam or hot water. Built in sizes to suit every need—in 3, 4, 5 and 6-tube types of 6 to 56 sections. Replace ordinary radiators with little or no change in piping.



STOKERS

Newly designed and engineered. Assure uniform heating with minimum fuel consumption and minimum attention. Wide range of sizes.



WATER SPECIALTIES

Crane hot water specialties are made to suit every installation. They include water heaters, circulators and flow control valves—all top-quality equipment.

OIL BURNERS

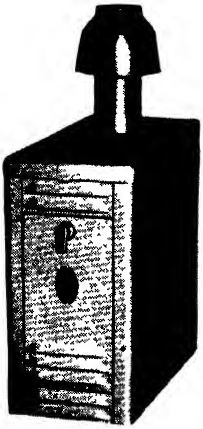
Newly engineered. Crane oil burners provide heating comfort at low fuel costs. Simplified design and sturdy construction reduce maintenance.

Hook & Ackerman, Inc.

PITTSBURGH 4, PA. **NEW YORK 17, N. Y.**

HYDROTHERM

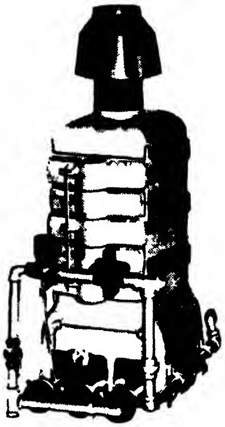
The Midget Automatic Gas Heating Plant
For residential, commercial and industrial hot water systems.



MODEL 2HW3
288 sq ft Installed Radiation
250 lbs, 13 in. x 26 in. x 26 in.

HYDROTHERM is engineered for maximum life and maximum fuel economy and fully *AGA* approved for use on Manufactured, Natural and LP gases. Outstanding for:

Radiant heating systems
Convector heating systems
Gravity hot water systems
Volume hot water heating
Booster service for 180 deg sterilizing water



MODEL 2HW5 w/o Jacket
400 sq ft Installed Radiation
375 lbs, 18 in. x 27 in. x 30 in.

CONSTRUCTION:—All cast iron, the HYDROTHERM absorption unit has horizontal sections, deep ribbed, overlapped and connected in zig-zag. This fully patented design is to give maximum heat transfer surface for a minimum water volume resulting in great efficiency and quick pick up. Absorption unit is furnished completely assembled and factory tested at 250 lb hydrostatic pressure. All controls including automatic gas control valve, Baso safety pilot, pressure regulator, aquastat and tridicator are enclosed in De Luxe jacket of Hammeroid finish.

CAPACITY RANGE OF HYDROTHERMS

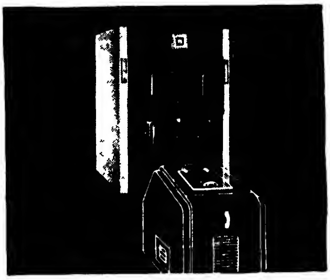
	HYDRO-THERM Model No.	A.G.A. RATINGS		Net Available Btu Per Hour	Will Supply Sq. Ft. Direct Radiation Water 170° F.	CAPACITY FOR HOT WATER STORAGE TANKS Gallons per Hour for Temperature Rise Shown				
		Btu Input	Water Rating Sq. Ft.			60°	80°	100°	120°	140°
SINGLE UNIT	2HW3	72 000	384	57 600	288	113	85	68	56	48
	2HW5	100 000	530	80 000	400	158	119	95	79	68
	2½HW3	150 000	800	120 000	600	237	178	142	118	100
	2½HW5	250 000	1333	200 000	1000	396	297	238	198	170
DUAL UNIT	2HW6	144 000	768	115 200	576	228	171	137	114	96
	2HW8	172 000	914	137 600	688	274	205	164	137	116
	2HW10	200 000	1060	160 000	800	317	237	190	158	136
	2½HW6	300 000	1600	240 000	1200	474	356	284	236	200
	2½HW8	400 000	2133	320 000	1600	633	475	380	316	270
	2½HW10	500 000	2666	400 000	2000	792	594	476	396	340

The H. B. Smith Company, Inc.

Westfield, Mass.

Branch offices and Sales Representatives in Principal Cities

A complete line of modern cast iron sectional boilers for residential, commercial and industrial heating and for domestic hot water supply



20 Mills

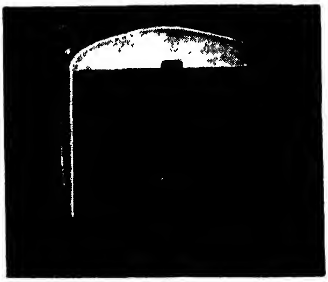
15-20-25 SMITH-MILLS BOILERS

Capacities 200 sq ft to 2275 sq ft steam radiation. This complete line of modern push nipple boilers is available in models for oil, gas, stoker and hand firing. Provisions have been made for built-in domestic hot water heaters and controls.

MILLS WATER TUBE BOILERS

Series 24-34-44

Capacities 900 sq ft to 13,380 sq ft of steam radiation. Independent header type construction--tens of thousands of these famous Mills Boilers are installed in schools, hospitals, apartment houses, stores, and other commercial and public buildings. Models for hand and all types of automatic firing.



44 Mills

60 Smith

42 AND 60 SMITH BOILERS

May be used in batteries for heating loads up to and over 100,000 sq ft steam radiation. Many of these large units installed in industrial plants furnish steam for process requirements as well as for heating and domestic hot water.

SMITH HY-TEST BOILERS

Smith Hy-Test Boilers for hot water supply, are available in several models and many sizes for tank capacities to 20,000 gal. Constructed of the finest quality grey iron castings, these Hy-Test units are carefully tested at high pressures before shipment. The No. 17 series, for example, is tested at 350 lbs hydrostatic pressure—the highest test pressure of any cast iron boiler made.

17 HY-Test

Complete catalog information describing Smith boilers is filed in current issues of Sweet's "Architectural" and Domestic Engineering Catalog Directory

THE NATIONAL RADIATOR COMPANY

MODERN DESIGN HEATING EQUIPMENT

Johnstown Pennsylvania

Branch Offices

BALTIMORE	2100 St. Paul Street	PHILADELPHIA	401 North Broad Street
BOSTON	620 Newbury Street	PITTSBURGH	125 First Avenue
CHICAGO	400 West Madison Street	RICHMOND	12 South Third Street
NEW YORK	60 East 42nd Street	WASHINGTON	4034 Georgia Avenue, N.W.

"100" Series

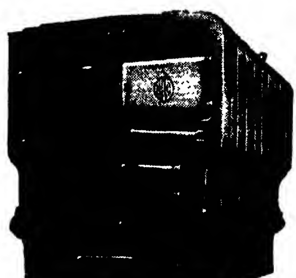
NATIONAL HEAT EXTRACTOR CAST IRON BOILERS. The distinguished family of National Heat Extractors was designed with full appreciation of the requirements of automatic heating. National Heat Extractors are provided with many features to insure high operating efficiency and maximum fuel economy. These include extended heating surface, multiple-flue section construction, effectively insulated jacket and doors, and heat conserver baffles. Convertible from hand to automatic firing after installation and readily adaptable to any desired fuel or method of firing. Wide range of both storage and tankless domestic water heaters available.

NATIONAL OIL HEATING UNITS, cast iron or steel, provide complete "one package" equipment designed for maximum efficiency and top performance with this fuel. Complete automatic controls, prefabricated combustion chamber of proper proportions, quiet burner for rear firing and attractive all-enclosing jacket.

NATIONAL GAS BOILERS are modern, compact and designed exclusively for gas firing. Cast iron sections for long life and dependability. Tapered flues, long zigzag fire travel and heavy insulation insure efficiency and economy.

"200" Series

"300" and "400" Series



"500" Series

Oil Heating Unit

HEAT EXTRACTOR BOILERS

Boiler Series	Net I-B-R Ratings, Sq Ft Steam	Water
100	170 to 410	270 to 660
200	350 to 880	560 to 1410
300	700 to 2300	1120 to 3680
400	2500 to 6000	4000 to 9600
500	4000 to 10300	6400 to 16480

Gas Boiler

OIL HEATING UNITS

Type of Unit	Net Ratings, Sq Ft Steam	Water
100 Series Cast Iron	230 to 410	370 to 660
200 Series Cast Iron	400 to 880	640 to 1410
Residential Steel	275 to 700	440 to 1120

GAS BOILERS

Boiler Series	Net Ratings, Sq Ft A.G.A. Approved Steam	Water
20 and 22	109 to 390	174 to 624
30	353 to 983	565 to 1570
40	445 to 1920	711 to 3070

18"-23" Steel Boiler

NATIONAL STEEL BOILERS meet all the requirements of the *SBI* Testing and Rating Code and the *ASME* Boiler Construction Code. All are inspected and approved by a representative of the *Hartford Steam Boiler Inspection and Insurance Company.* The 18 in. and 23 in. Series RESIDENTIAL STEEL BOILERS are designed for smaller homes. The 26 in., 29 in., and 39 in. Series RESIDENTIAL STEEL BOILERS are designed specifically for large homes, apartments, and commercial installations. Can be furnished unjacketed, or with modernly styled heavily insulated jacket.

RESIDENTIAL STEEL BOILERS

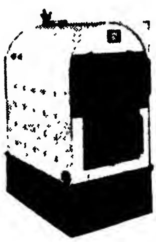
Boiler Series	SBI Net Ratings, Sq Ft	
	Steam	Water
18" and 23"	275 to 700	440 to 1120
26", 29", and 39"..	700 to 3000	1120 to 4800

26"-29"-39" Steel Boiler

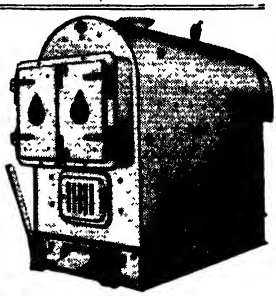
NATIONAL COMMERCIAL STEEL BOILERS are designed to supply the heating needs of larger buildings. Their firebox proportions conform with the recommendations of the *Stoker Manufacturers' Association.* Hand fired boilers may be converted to stoker, oil, or gas firing at any time after installation.

COMMERCIAL STEEL BOILERS

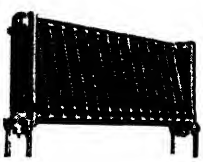
Boiler Type	SBI Net Ratings, Sq Ft	
	Steam	Water
Automatically fired . .	3000 to 35000	4800 to 56000
Hand fired	2500 to 29170	4000 to 46670



26"-29"-39" Steel Boiler Unjacketed



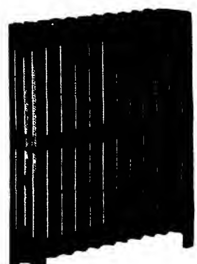
Commercial Steel Boiler



Aero Convector

NATIONAL AERO CONVECTORS are of compact cast iron design with fins cast integral with tubes. Adaptable to any type of heating system and can be used with direct radiation.

NATIONAL ART RADIATORS blend inconspicuously with most decorative schemes. Compact proportions require minimum floor space. A wide variety of sizes and ratings can be furnished.



Art Radiator

NATIONAL CONVECTOR ENCLOSURES are made in several different types applicable to modern structural or decorative requirements. They were designed to develop maximum output of the convector.

Convector Enclosure

NATIONAL UNIT HEATERS are designed for quiet, economical operation with steam or hot water. Available in both Horizontal and Vertical delivery types in a number of sizes.

Unit Heater Horizontal Delivery

Spencer Heater

Division—AVCO Manufacturing Corporation

Williamsport, Pennsylvania

Sales Representatives

ALLENTOWN, PA.	CHICAGO, ILL.	GRAND RAPIDS, MICH.	LOS ANGELES, CALIF.	PORTLAND, ORE.
ATLANTA, GA.	CINCINNATI, OHIO	GREENSBORO, N. C.	MEMPHIS, TENN.	RICHMOND, VA.
BALTIMORE, MD.	CLEVELAND, OHIO	HARRISBURG, PA.	NASHVILLE, TENN.	SALT LAKE CITY, UTAH
BINGHAMTON, N.Y.	COLUMBUS, OHIO	HOUSTON, TEXAS	NEW YORK, N. Y.	SAN ANTONIO, TEXAS
BIRMINGTON, ALA.	DALLAS, TEXAS	INDIANAPOLIS, IND.	PHILADELPHIA, PA.	ST. PAUL, MINN.
BOSTON, MASS	DETROIT, MICH.	KANSAS CITY, MO.	PITTSBURGH, PA.	WASHINGTON, D. C.

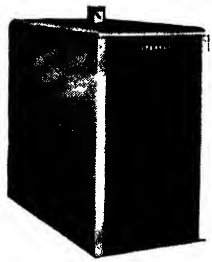
There is a modern SPENCER in size and type for every fuel and heating need. All products are manufactured in strict accordance with the *ASME* Code and carry the Code Seal. A SPENCER assures the engineer and architect of specifying with confidence, the contractor ease of installation and the owner efficient economical operation.

SPENCER Residential

Steel Heating Boilers. 320 to 3000 Net Load Steam

The SPENCER "C" (700–3000 Net Steam) contains many features found only in large commercial boilers. Design includes precision ground nonwarping water cooled steel door frames. Easily and quickly converted from automatic to hand firing.

The SPENCER "K" (320 to 550 Net Steam) is a star performer for the small home where the conversion firing feature is desired.



The SPENCER "R" (320 to 700 Net Steam) is an all steel boiler designed especially for automatic firing.

All Series Fully Approved

"C" Series Cutaway With Extended Jacket.

All SPENCER Steel Boilers available with attractive standard or extended model jackets and year round domestic hot water heating coils.

THE NEW "21 SERIES" ALL PURPOSE BOILER

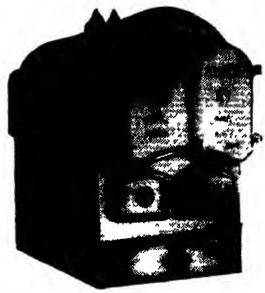
340 to 1000 Net Load Steam

All Purpose Cutaway Baffle For Oil Burner Installation.

An entirely new cast iron sectional boiler incorporating features that make it an outstanding boiler in its field. Peaked combustion chamber providing extra height so important to mechanical firing installations. Available with year round domestic hot water coils entirely submerged in active waterway of boiler. No protruding equipment to mar appearance of boiler jacket. Hand fired grate assembly located in base, easily removed for mechanical conversion.

SPENCER SERIES "A" ALL FUEL COMMERCIAL STEEL HEATING BOILER

1800 to 42,500 EDR (Steam)



America's standout boiler for commercial buildings, apartments or industrial plants. The many features of the Spencer Series "A" are essentially the same for hand firing, oil, gas or stoker types. It is especially noted as a quick-steaming boiler.

The Series "A" is of proven design– in use in thousands of installations. Spencer's efficient fire box design obtains all the heat from the fuel, giving real economy. All its internal areas are easily accessible—easy to clean, keeping boiler at peak efficiency—resulting in economical performance. The "A" Boiler has either rear or top smoke box outlet and you can readily install the Spencer year-round hot water system.

All sizes available with either mechanical (oil, gas or stoker) or hand-fired bases. Smokeless arch supplied on special order.

SPENCER L-2 Showing Sloping Gravity Feed Grate Design.

SPENCER SERIES "A" CASTIRON MAGAZINE FEED BOILERS

Capacities 290 to 4510 EDR (Steam)

Designed for quick, economical heat with minimum service requirements for homes, apartments, churches and commercial buildings. Fuel feeds by gravity, automatically, constantly,—needs little attention in 24-hour period in coldest weather. No mechanical parts—burns small sized anthracite or coke at substantial savings.

There is the "F" series (290 to 740 EDR Steam) for the average home—available with attractive beauty jackets. The L-2 series (690 to 1570 EDR Steam) for the larger homes and small buildings and the L-3 series (1430 to 4510 EDR Steam) for larger homes, commercial and public buildings. The L-3 has the advantage of operating only half of the boiler for year round domestic service water or mild weather operation.

Pacific Steel Boiler DIVISION

United States Radiator Corporation

Detroit 31, Michigan

Sales Offices in Principal Cities

Member SBI Reg U.S Pat. Off.

A Complete Line of Low-Pressure Steel Heating Boilers

All Pacific Boilers are built using the *A.S.M.E.* Boiler Code Standards as minimums, and rated in accordance with *Steel Boiler Institute* code.

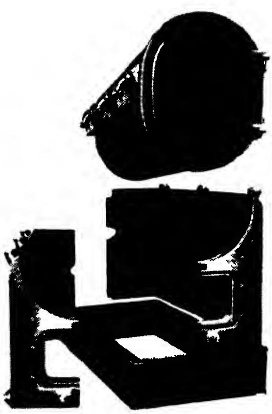
Pacific High Firebox Boiler

PACIFIC HIGH FIREBOX BOILERS

Pacific High Firebox Boilers for mechanical firing, stoker, oil or gas, are built in capacities of 2680 to 56,830 sq ft for steam and in corresponding capacities for water.

Pacific Direct Draft and Smokeless Boilers for coal firing are built in capacities of 2200 to 35,000 sq ft for steam and in corresponding capacities for water.

All Pacific Boilers are made of flange-quality steel, electrically welded—built to last. They are built, inspected and tested under the supervision of the *Hartford Steam Boiler Inspection and Insurance Company.*



Pacific Split Firebox Boiler

PACIFIC THREE-PIECE CONSTRUCTION

All commercial sizes of Pacific Boilers are built in three sections—shell, firebox and base—and require minimum building opening. Where necessary, Pacific fireboxes can be split (as illustrated) to allow the boiler to be taken into the building in four sections. No cutting, no welding is required in assembling any Pacific Boiler.

PACIFIC RESIDENTIAL BOILERS FOR COAL, STOKER, OIL OR GAS

Built in the following capacities for steam: 400 to 3000 sq ft and in corresponding capacities for water.

Pacific "Plate Flue" Boiler

PACIFIC *"Plate Flue"* BOILERS

For oil or gas—built in 4 sizes ranging from 400 to 900 sq ft for steam and from 640 to 1440 sq ft for water.

Descriptive Bulletins on Pacific Boilers will be mailed on request

UNITED STATES RADIATOR CORPORATION
FOR 58 YEARS A GREAT NAME IN HEATING

I-B-R Member
Reg. U. S. Pat. Off.

General Office, Detroit 31, Mich:
Branches and Sales Offices in Principal Cities

The United States Radiator Corporation carries a complete line of hand, stoker, gas and oil-fired boilers for steam or hot water heating, warm air furnaces, gas burners and oil burners, radiators, baseboard convectors, controls and heating accessories for residential, commercial and industrial heating.

U. S. "Comfort Cub" Oil Boiler-Burner Unit

U.S. Oil-Fired Boilers are completely automatic in operation and contain all necessary controls. They are available with standard or extended jackets for small, residential or commercial heating installations.

Ratings and Dimensions—U.S. "COMFORT CUB" OIL-FIRED BOILER

Net I-B-R Rating—400 sq ft Water
Net I-B-R Rating—60,000 Btu/hr.
Gross I-B-R Output—93,000 Btu/hr.
I-B-R Burner Capacity—0.90 gals/hr.
Chimney Size—8 in. x 8 in. x 20 ft

Dimensions of Complete Assembly
Jacketed Boiler, Burner and Controls require 22⅝ in. width x 32¾ in. depth Floor Space

U.S. Gas-fired Boilers

U.S. Gas Boilers include a complete range of sizes and capacities from 800 to 12,480 sq ft for steam heating, and from 240 to 20,000 sq ft for hot water heating. U.S. Gas Boilers are furnished with necessary control equipment and are completely automatic in every operation. All units are *A.G.A.* approved.

U.S. Copper Convectors

U.S. Copper Convectors are designed for use with forced circulating hot water, two-pipe steam, vapor or vacuum systems. They are constructed of copper tubing with non-ferrous fins. Cabinets are made of reinforced heavy gage steel. Simple chain control damper facilitates heat regulation. For complete data, write for catalog.

U.S. Fin-Ray Baseboard Convectors

U. S. Fin-Ray Baseboard Convectors—for homes, offices and commercial buildings. They can be furnished in 1 in. and 1¼ in. pipe sizes. Write for Catalog AR-299 for complete data and specifications.

THINTUBE RADIATORS

Heights In.	Per Section Heating Surface
3-Tube	
25	1.6 Sq Ft
4-Tube	
22	1.8 Sq Ft
25	2.0 Sq Ft
5-Tube	
22	2.1 Sq Ft
25	2.4 Sq Ft
6-Tube	
19	2.3 Sq Ft
25	3.0 Sq Ft
32	3.7 Sq Ft

*** 1¾ in. Centers.**

U.S. SUNRAY RADIATORS

19 in. height is standard, 21 in. and 22½ in. heights can be furnished by use of higher legs.

No. 5—Depth 5 5/16 in.

Heights In.	Per Section Heating Surface
19	1.8 Sq Ft

No. 6—Depth 6 1/8 in.

Heights In.	Per Section Heating Surface
19	2.3 Sq Ft

UNITED STATES RADIATOR CORPORATION
FOR 58 YEARS A GREAT NAME IN HEATING

Member I.B.R. Reg. U. S. Pat. Off.

General Office, Detroit 31, Mich.
Branches and Sales Offices in Principal Cities

U.S. 2 Oil-Fired Boiler

Boiler No.	Net Sq Ft Direct Cast Iron Radiation		Net Btu Per Hr.	Gross Output Btu Per Hr.
	Steam	Water		
2-03	350	560	84,000	129,000
2-04	500	800	120,000	182,000
2-05	650	1040	156,000	234,000
2-06	800	1280	192,000	285,000

U.S. 25 Oil-Fired Boiler

Boiler No.	Net Sq Ft Direct Cast Iron Radiation		Net Btu Per Hr.	Gross Output Per Hr.
	Steam	Water		
US-25-3	545	875	131,000	197,000
US-25-4	905	1450	217,000	320,000
US-25-5	1265	2025	304,000	439,000
US-25-6	1625	2600	390,000	555,000
US-25-7	1985	3175	476,000	669,000

U.S. 3 "Thrift Model" Boiler

Boiler Number	Sq Ft Direct Cast Iron Radiation					
	Hand-Fired		Stoker-Fired		Oil-Fired	
	Steam	Water	Steam	Water	Steam	Water
R43-S or W	300	480	400	640	400	640
R53-S or W	450	720	550	880	550	880
R63-S or W	600	960	700	1120	700	1120

U.S. Red Top Boilers

Boiler No.	Sq Ft Direct Cast Iron Radiation Hand-Fired	
	Steam	Water
"A" Series—Handfired (ratings shown) Stoker-Oil		
A-7	340	545
A-8	440	705
A-9	540	865
A-10	640	1025
A 11	740	1185
"B" Series—Handfired-Stoker		
B-7	740	1180
B-8	920	1470
B-9	1110	1770
B-10	1275	2040
B-11	1440	2300
B-12	1580	2530
B-13	1730	2770
B-14	1880	3000
"C" Series—Handfired (ratings shown) Stoker-Oil		
C-14	2715	4345
C-16	3180	5090
C-18	3645	5830
C-20	4110	6575
C-22	4575	7320
C-24	5040	8065
C-26	5450	8720
C-28	5800	9280
C-30	6110	9775

U.S. Water Heaters

U.S. Hot Water Supply Boilers, for an ample supply of domestic hot water, are available with either hand, oil, gas or electrical firing. Write for catalog.

U.S. Vertical or Horizontal Unit Heaters

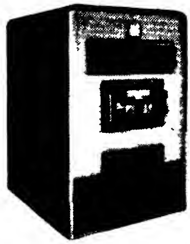
U.S. Vertical or Horizontal Unit Heaters are available with either 60 cycle AC or 25 cycle AC or DC motors in a wide range of sizes and capacities. Optional control equipment for special applications may be obtained at extra cost. Write for catalog.

Weil-McLain Company

Manufacturing Division: Michigan City, Ind. and Erie, Pa.
General Offices: 641 W. Lake Street, Chicago 6, Ill.
NEW YORK OFFICE: 501 Fifth Avenue
Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers.

Nos. 57, 67, 77 All-Fuel Boilers

Conversion type boilers for hand or automatic firing. Connected Load Ratings: Steam 210 to 1130 sq ft, Water 340 to 1810 sq ft.



No. 87 All-Fuel Boiler

Conversion type boiler for hand or automatic firing. Insulated and jacketed. Connected Load Ratings: Steam 900 to 2,100 sq ft, Water 1,440 to 3,360 sq ft.

Square-Type Boilers

Sectional boilers for larger installations 28, 40 and 44 Series. Connected Load Ratings: Steam 1,580 to 11,300 sq ft, Water 2,530 to 17,900 sq ft.

Nos. 68 & 78 Boilers for Automatic Firing

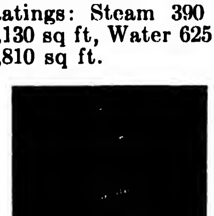
Boilers have insulated, enameled, extended jackets. Net I-B-R Ratings: Steam 390 to 1,130 sq ft, Water 625 to 1,810 sq ft.

Round-Type Boiler

Unjacketed Round Boiler with corrugated heating surfaces for economical home heating. Connected Load Ratings: Steam 310 to 900 sq ft Water 490 to 1440 sq ft.

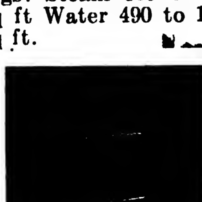
Type G Gas Boiler

Jacketed gas water boiler for natural, mixed or manufactured gas. *AGA* approved. Connected load ratings: Water 305–625 sq ft.



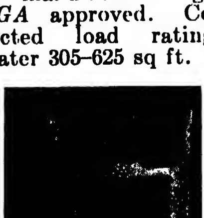
Raydiant "Concealed"

A Radiant convector type all cast-iron Radiator. Made in "Concealed," also Partially Recessed types.



Solray Radiator

Free standing, all cast-iron Cabinet-type Radiator with metal cover top. Three depths: 21, 24, and 27 in. high; and one depth-18 in. high.



Junior Radiator

Smaller Tubular type Radiation which conserves space. Available in 1¾ in. centers in 3, 4, 5 and 6 tube widths and 19 to 32 in. heights.

THE BABCOCK & WILCOX COMPANY

85 Liberty Street | **AG-137** | **New York 6, N. Y.**

Manufacturers of

Water-Tube Boilers | **Chain-Grate Stokers**

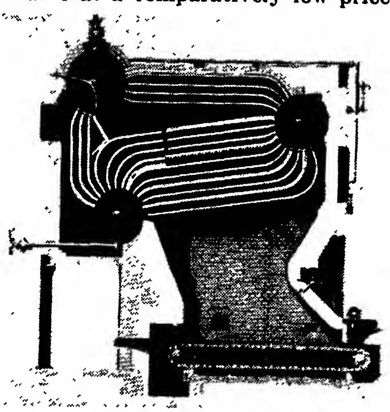
Oil Burners | **Seamless Steel Tubing and Pipe**

Branch Offices and Representatives in all Principal Cities

Type H Stirling Boiler

This three-drum, bent-tube unit is a relatively small water-tube boiler designed primarily for installation where head-room and floor-space is limited. It is particularly suitable for industrial power plants, for process steam requirements, and for heating purposes, and is available in four classes and 36 sizes in steam outputs up to 50,000 lb per hour and pressures up to 450 lb. Draining type superheaters can be furnished, and the unit is adaptable to all kinds of stokers, and to oil and gas firing.

Its conservative simple design and its production by standardized methods make the B&W Type H Stirling Boiler available at a comparatively low price.



B&W Type H Stirling Boiler fired by a B&W Chain Grate Stoker

Principal Advantages

High efficiency and unusually quick steaming capacity for the limited floor space and head-room required.

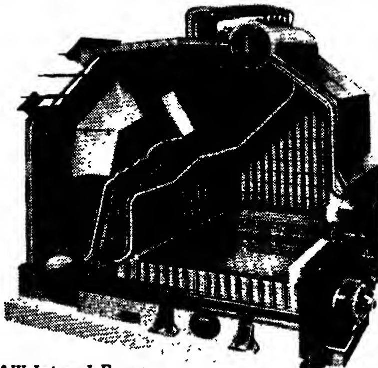
Choice of three locations for gas exit, reducing cost of flues and breeching.

High rates of heat transfer, with corresponding high efficiencies.

Easy accessibility for thorough cleaning and inspection.

Elimination of many difficulties usually encountered in the repair and upkeep of fire-tube boilers.

Ask for Bulletin G-8-E.



B&W Integral-Furnace Boiler, Type FF, equipped for oil firing with B&W Combination Oil and Gas Burners.

B&W Integral-Furnace Boiler, Type FF

This self-contained steam-generating unit for power, process, or heating service is setting new high standards of performance in the moderate-capacity field. It is designed to a degree of standardization that makes it available at very reasonable cost, and incorporates many of the latest advances in steam generation employed by power stations, and large industrial plants. It provides steam capacities ranging between 8,000 and 50,000 lb per hour, in pressures from 160 to 600 psi, and with superheated steam temperatures as high as 750F in the standard designs.

Outstanding Features

A water-cooled furnace, adaptable to stoker, oil, or gas firing, provides in small space a combustion chamber that efficiently performs the function of an all-refractory furnace of larger size. It virtually eliminates furnace maintenance, and permits higher steam inputs, higher furnace temperatures, and higher steaming rates.

A secondary furnace, or Open-Pass, which insures thorough mixing of the gases at high temperatures, thus improving combustion.

Ask for Bulletin G-64.

The Bigelow Company

105 River Street, New Haven 3, Connecticut

MANUFACTURERS OF

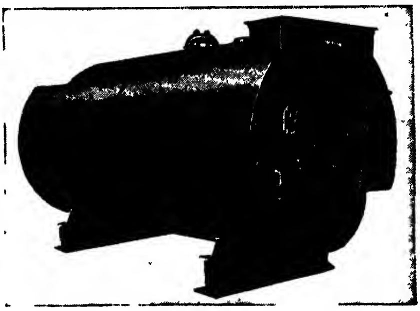
FIRE TUBE BOILERS **WATER TUBE BOILERS**

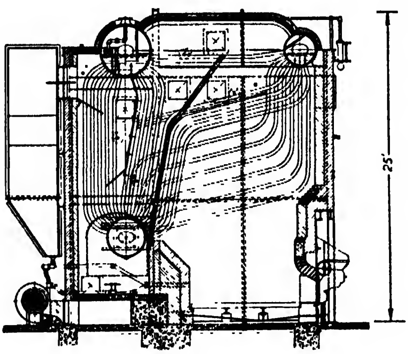
SALES REPRESENTATIVES IN PRINCIPAL CITIES

Catalog Furnished on Request

BIGELOW SCOTCH TYPE HEATING BOILER

A compact, self contained, highly efficient oil fired boiler that requires low head room and has low water line. All parts internally and externally are readily accessible. The boiler is of welded construction and contains no water leg or stay-bolted surfaces. Built in units with steam rating from 1,820 to 42,500 sq ft.



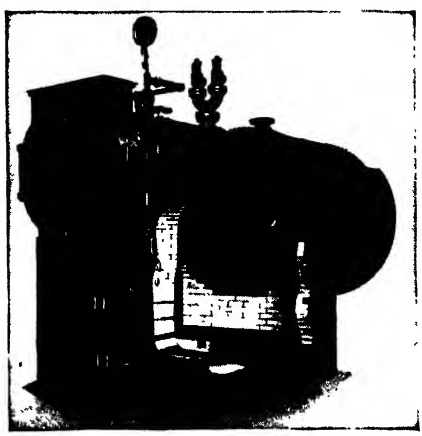


BIGELOW TYPE F WATER TUBE BOILER

Cut shows boiler equipped with spreader type stoker. The flexibility of design permits oil and other type stoker firing. Partial water wall surface is provided without extra headers, downcomers and circulating tubes. The Type F Steam Generating Unit is built in sizes ranging from 10,000 to 60,000 lb of steam per hour.

BIGELOW TWO-PASS BOILER

This boiler is designed to meet heating and power requirements, especially where space limitations prevail. The unit may be oil, stoker or hand fired. The elimination of special brick shapes and stay-bolts reduces the cost of maintenance to a minimum. The Two-Pass boiler is built in standard sizes from 25 hp to 250 hp.



The Brownell Company

Established 1855

452 N. Findlay St.

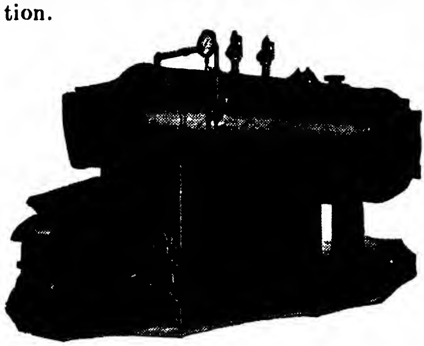
Dayton 1, Ohio

Manufacturers of

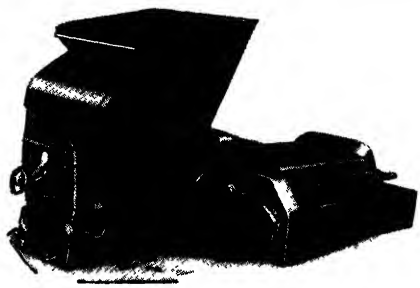
BROWNELL BOILERS, STOKERS, AND HEAT EXCHANGERS

Representatives in All Principal Cities

Welded Triple Pass Heating Boilers built in either high leg or low water line types. Hand fired ratings 500 to 35,000 sq ft steam, 800 to 56,800 sq ft water radiation. Stoker, Oil or Gas fired up to 43,100 sq ft steam or 69,000 sq ft water radiation. *A.S.M.E.* Code construction.



High or Low Pressure Double Pass Boiler with Type LR Stoker. Designed and manufactured as a matched unit steam generating plant. Furnished in working pressures from 15 to 150 lb and sizes up to 300 hp. For power, heating, and process steam. Steam ratings 3,600 to 42,500 sq ft. Water rating, 5,800 to 68,000 sq ft when used with stoker, oil, or gas. *A.S.M.E.* Code construction.



Type LR (Low Set) Underfeed Ram Type Stoker. Ideal where height of setting is limited. Sizes to 300 hp; "R" models up to 700 hp. Automatic air volume control. Can be furnished with Brownell exclusive, fully automatic intermittent coal feed control.

Type C Screw Feed Stoker, proved by years of service to be sturdy, reliable and efficient. Illustration shows dead plates; can also be furnished with dump plates in the larger sizes. 30-300 hp.

Heat Exchangers, Generators, Converters. Hourly capacities from 60 to 4,400 gal; storage capacities 25 to 1,904 gal.

The illustrations show only a part of the complete Brownell line. We shall gladly send literature describing Brownell equipment. Our field organization is ready to assist in problems of steam generation and heating.

Combustion Engineering-Superheater, Inc.

All Types of Fire Tube and Water Tube Boilers
Mechanical Stokers

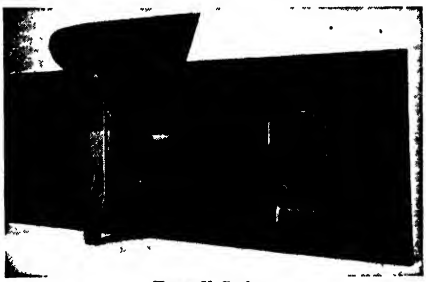
Complete Steam Generating Units
Pulverized Fuel Systems

200 Madison Avenue, **New York 16, N. Y.**

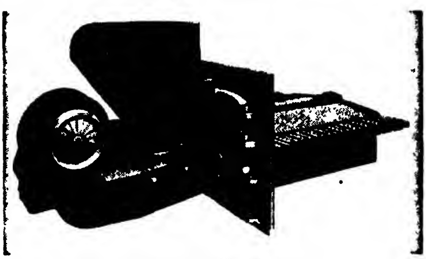
Offices in all principal cities of the United States and Canada

More than 20,000 C-E Stokers purchased to date

C-E Skelly Stoker



Type E Stoker



C-E Low Ram Stoker

C-E Spreader Stoker

C-E Skelly Stoker—A compact, self-contained unit with integral forced-draft fan, adapted to burn either anthracite or bituminous coal. Alternate fixed and moving grate bars assure lateral distribution of fuel. Automatic control is standard equipment. Approximate application range—20 to 200 rated boiler hp.

Type E Stoker—A single-retort, underfeed stoker with an established reputation of many years' standing for dependable service. Designed to burn a variety of bituminous coals under boilers up to about 600 rated hp. Available with steam, mechanical or electro-hydraulic drive.

C-E Low Ram Stoker—A single-retort, stationary-grate underfeed stoker for burning bituminous coals under boilers in the upper size range of the C-E Skelly Stoker.

C-E Spreader Stoker—A simple, rugged overfeed stoker designed to burn a wide variety of coals. Fines are burned in suspension and the coarser coal on a grate which may be of either stationary, dumping or continuous discharge type. Rate of coal feed and air supply may be regulated over a wide range and are readily adaptable to automatic control. Applicable to boilers from about 100 boiler hp up.

C-E Multiple Retort Stoker—For burning bituminous and semi-bituminous coals under boilers up to the largest sizes.

C-E Traveling Grate and Chain Grate Stokers—Including both Coxe and Green types. Available with grate surfaces suitable for anthracite, coke breeze, lignite or bituminous coal, as required. Traveling grates are all forced-draft types; chain grates are either forced draft or natural draft types.

C-E Boilers—All fire tube and water tube types in sizes ranging from 25 hp up to the largest. Standard and special designs to suit all conditions of fuel, load and space. Included are all types formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges," "Ladd" and "Nuway."

Separate Catalogs describing each of these products are available.

A-531-E

Fitzgibbons Boiler Company, Inc.

General Offices:
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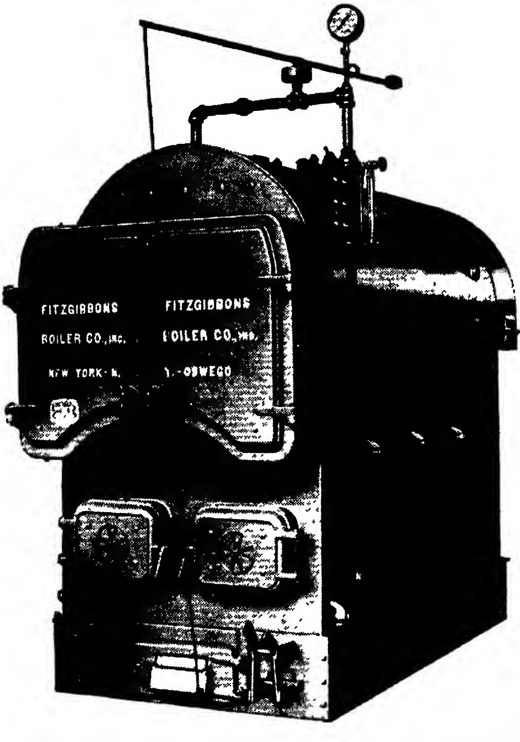
Manufactured at: OSWEGO, N. Y.
Sales Branches in Principal Cities

Established 1886

Member SBI
Reg. U.S. Pat. Off.

PRODUCTS—STEEL BOILER HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *A.S.M.E.* and *S.B.I.* Codes.—AIR CONDITIONERS for Direct-Fired installations in residences of all sizes.



FITZGIBBONS "D" TYPE

Welded Steel Firebox Boilers
FOR 15 lb W.S.P.

•
***A.S.M.E.* BUILT**
•
***S.B.I.* RATED**
•
***HARTFORD* INSPECTED**

The boiler for apartments, office buildings, theatres, schools, hospitals—capacities up to 42,500 sq ft steam EDR.

Full and complete combustion is a feature of the Fitzgibbons "D" Type steel boiler. In the oil or gas fired type, the generous combustion area insures the liberation of every Btu.

In the coal burning type (stoker or hand fired) grate aperatures and secondary air intake are correct for admitting the right amount of air. Flue gases make several passes over the length of the boiler.

Positive water circulation—a rapid continuous surging sweep induced by the concentration of heat at the crown sheet. This powerful circulation is further helped by the absence of convolutions or distorted forms which might impede it.

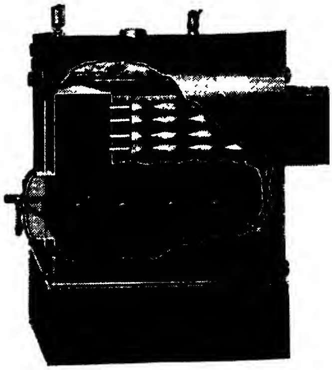
These two factors—efficient combustion and positive water circulation—insure the quick heating so valuable in automatic firing, where rapid response to thermostat demands cuts down firing periods, and saves fuel.

Ample domestic hot water is provided with no storage tank by the Fitzgibbons "Tanksaver."* Where a storage tank is used, the "Tankheater" may be furnished.

* **Reg. U. S. Pat. Off.**

FITZGIBBONS "80" SERIES

The Steel Boiler for Smaller Buildings



This is the latest boiler in the Fitzgibbons line, carrying to a new high point those principles for which Fitzgibbons steel boilers have long been known. Particularly high in efficiency, fuel economy, quick heating, due to large rectangular firebox and rapid water circulation which promote heat transfer. A willing teammate for any oil burner, gas burner, anthracite stoker. Also available for hand-fired anthracite coal.

Extra large domestic hot water capacity from the TANKSAVER* all year 'round, with no storage tank needed.

Sizes from 1100 to 3000 sq ft steam *net* load and 1760 to 4800 sq ft water *net* load.

FITZGIBBONS "400" SERIES

The Steel Boiler for Homes

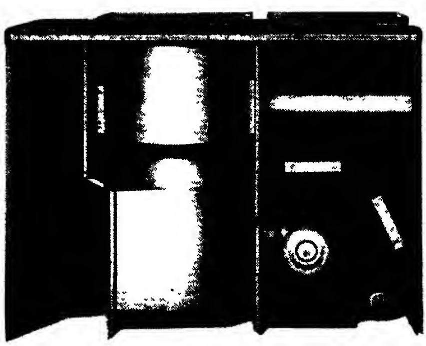
A boiler that brings to the small home every advantage of Fitzgibbons steel boiler construction. Electrically welded in a one-piece unit, it combines strength and durability with leak-proof, crack-proof construction.

The 400 Series is a particularly quick heating boiler, giving immediate response to thermostat control. This reduces the periods of burner operation and is the reason for outstanding fuel savings and comfortable temperature uniformity enjoyed by the thousands of 400 series heated homes. Abundant domestic hot water is provided by the Fitzgibbons Tanksaver* summer and winter, with no storage tank required.

This boiler is available in types for all mechanical firing methods, and for hand firing with coal. Sizes from 320 to 900 sq ft *net* steam, 512 to 1440 sq ft *net* water.

THE FITZGIBBONS DIRECTAIRE

The Winter Air Conditioner for Residential Comfort



The Directaire tempers the air in response to thermostatic action, humidifies the air automatically to the degree correct for comfort, cleans the air of dirt, dust and foreign matter, and circulates the air quietly, and without drafts. The Directaire is notable for fuel economy, due to the Fitzgibbons "Contra Flo" principle, in which the passage of air through the unit is opposed to the flow of combustion gases.

Fitzgibbons "Weld-Seal" crack-proof construction, prevents all chance of combustion gas leakage into the air stream. Installation is quick and easy. Six sizes, 65,000 to 200,000 Btu/hr, mechanically fired.

* Reg. U. S. Pat. Off.

Farrar & Trefts

Incorporated

ESTABLISHED 1863

20 Milburn Street, Buffalo 12, N. Y.

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The Bison Compact Boiler Series 100 and 900

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 2,200 to 35,000 sq ft of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.

Firebox Return Tubular Boiler Series 500 and 600

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions. Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc.

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and riveted, or, Class 1 fusion welded x-rayed and stress-relieved for power purposes at 100, 125 and 150 lb working pressure in accordance with the *A.S.M.E.* Code. Sizes from 4,500 to 35,000 sq ft of steam radiation, as rated by the *Steel Heating Boiler Institute*, are designed for hand firing with coal or for mechanical firing with oil, gas or stoker.

Scotch Wet Top Back Boilers are designed so that no refractory tile are required at the top of the rear combustion chamber. The steam space extends the entire length of the boiler and the furnace is entirely surrounded by water which permits immediate maximum heat transfer and absorption. This special design results in a boiler that is extremely efficient to operate and is maintained at a minimum cost.

The Scotch Wet Top Back Boiler is a self-contained unit. It can be moved easily and can be installed on two saddles. No expensive foundation or pit is required. No external brickwork is needed. Because of its short length, low height and low water line, this compact boiler unit can be installed in small spaces where there is lack of headroom and where no other type will fit.

Designed for oil, gas or mechanical firing, in accordance with the requirements of the *A.S.M.E.* Code for 15 lb working pressure, ratings of these boilers conform to *S.B.I.* Sizes range from 3160 to 42500 sq ft of steam radiation.

Write for Complete Catalog

The International Boiler Works Company

350 Birch St., East Stroudsburg, Pa.

SALES OFFICES IN PRINCIPAL CITIES

years "Fuel-Saver" Boilers Type C have met the requirements for low cost heating in office and apartment buildings, hotels, schools, theatres, industrial plants, etc.

Their design and construction makes them especially suitable for modern heating requirements:

QUICK STEAMING

Due to rapid and positive internal water circulation.

MAXIMUM HEAT ABSORPTION

Due to effective distribution of heated gases.

EASE OF CLEANING

Due to accessibility of heating surfaces.

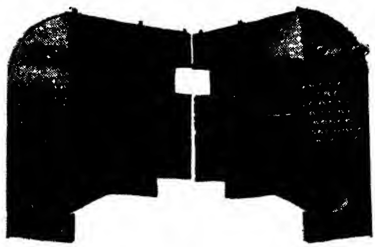
"FUEL-SAVER" Boilers have cut fuel costs in thousands of heating installations.

Complete range of standard sizes rated in accordance with *S.B.I.*-15 lb *A.S.M.E.* standard—for hand, stoker, oil or gas firing.

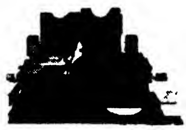
Type C twin section—a heating boiler in halves. For installation where Type C one piece cannot be carried through existing passages.

Type KD—knocked down—a heating boiler designed for shipment so that sections can be carried through a door or window. Eliminates expensive cutting or patching of building. Reduces time out when in need of steam.

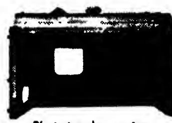
Typical Type C Installation



Halves of twin section ready for bolting



Type KD knocked down for shipment.



First step in erection.

Second step in erection.

Erected in boiler room.

Type CR water-tube power boiler for processing and power. 100-125-150 s.w.p. *A.S.M.E.* standard.

Complete range of sizes—10 to 300 hp for hand, stoker, oil or gas firing.

BOILER BUILDERS FOR 63 YEARS

Write for bulletins

KEWANEE BOILER CORPORATION

Division of American Radiator & Standard Sanitary corporation

Kewanee, Illinois

BRANCHES IN 64 PRINCIPAL CITIES

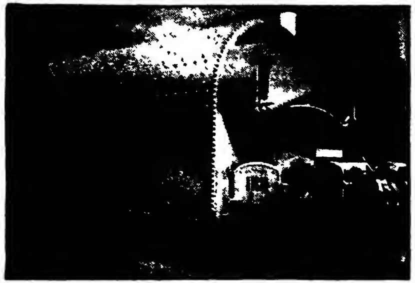
Steel Heating and Power Boilers, Water Heating Garbage Burners, and Tabasco Heaters

SBI Reg U S Pat Off MEMBER

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building with high efficiency burning any kind of fuel. Standard sizes and types of Kewanee Boilers are kept in stock ordinarily for immediate delivery.

Over eighty years of intensive study and effort are back of Kewanee Boiler designs. They are all constructed in our extensively equipped factory at Kewanee, Illinois, in conformity with these Codes: American Society of Mechanical Engineers for construction, and for rating with the Steel Boiler Institute Simplified Practice. The Kewanee series include:



Firebox Boiler Portable Up-draft Type "400" and "500" Series, Hi-Pressure

Heavy Duty Riveted Firebox Types: 1,380 ft to 42,500 ft. 10 to 304 Horsepower. Steam Working Pressures 100, 125 and 150 lbs. Single-pass tubes for rear smoke outlet; Two-pass tubes for front smoke outlet.

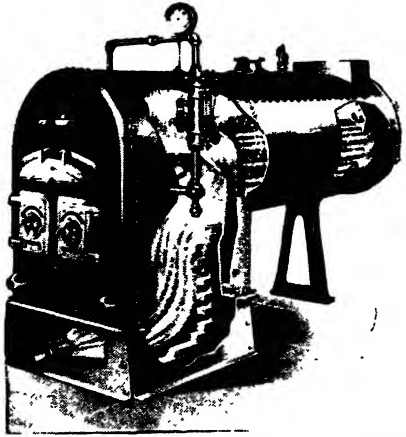
Welded Boilers: 2,200 ft to 42,500 ft. Direct Draft Type "C" with Corrugated Crown Sheet, rear Smoke outlet, and welded firebox two-pass for front Smoke outlet.

Residence Steel Boilers: 275 ft to 3,000 ft. Square and Round Type "R" with and without Jackets and Hot Water Heating Coils for Storage Tank or Instantaneous flow.

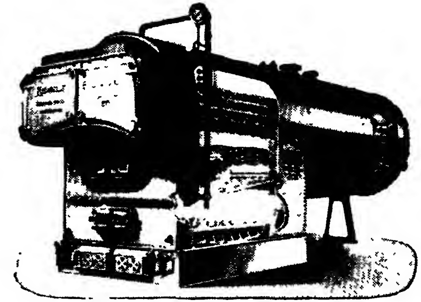
Firebox Boiler for Oil, Gas or Stoker "500" Series

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

Boiler No.		576	577	578	579	580 480	581 481	582 482	583 483	584 484	585 485	586 486	587 487	588 488	589 489	590 490
SBI Rating Steam, Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Horsepower	Hp	25	29	32	36	43	50	61	72	89	107	125	143	179	214	250
SBI Rating Oil, Gas, Stoker	Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Horsepower	Hp	30	35	39	44	52	61	74	87	109	130	152	174	217	261	304
Width and Length	In. x Ft In.	42x8-7	42x9-6½	48x8-11½	48x9-8	54x9-2½	54x10-6	60x11-1	60x12-9	66x12-10	66x14-9	72x14-9	78x14-9	78x17-7½	84x17-9½	84x20-0½
Overall Height Shell	In	80½	80½	86½	86½	94½	94½	102	102	108	108	114	116	116	126	126
Height of Water Line	In.	73	73	76	76	83	83	88	88	93	93	97½	100	100	108	108
Shipping Weight Coal	125 Lb	8500	9000	10500	11000	12500	14000	16000	18000	20500	23500	26000	29000	33500	37500	41500
Oil, Gas or Stoker	125 Lb	7980	8510	9590	10100	11700	12760	14930	16530	19230	21940	24100	26840	31210	35000	38820



Type "K" Up-draft Boiler for Hi-Pressure



Smokeless Boiler Portable Down-draft Type "300" Series for Hi-Pressure

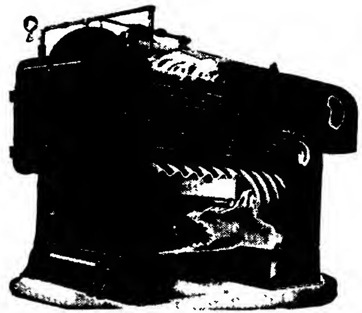
SPECIFICATIONS—TYPE "K" PORTABLE UP-DRAFT BOILER

Boiler No.		4K	5K	6K	8K	9K	10K	11K	12K
SBI Rating Steam, Coal	Sq Ft	1380	1800	2200	3000	3500	4000	4500	5000
Horsepower	Hp	10	13	16	22	25	29	32	36
SBI Rating Oil, Gas, Stoker	Sq Ft	2020	2190	2680	3650	4250	4860	5470	6080
Horsepower	Hp	14	16	19	26	30	35	39	44
Width and Length	In. x Ft In.	36x8-10	36x10-4	36x11-10	42x11-1¾	42x12-7¼	48x11-10½	48x13-4½	48x14-10½
Overall Height Shell	In.	65½	65½	65½	71½	71½	77½	77½	77½
Height of Water Line	In.	57	57	57	60½	60½	63	63	63
Shipping Weight Coal	125 Lb	4800	5000	6500	7500	8500	9000	10500	11000
Oil, Gas or Stoker	125 Lb	4440	4710	5820	6880	7980	8510	9590	10100

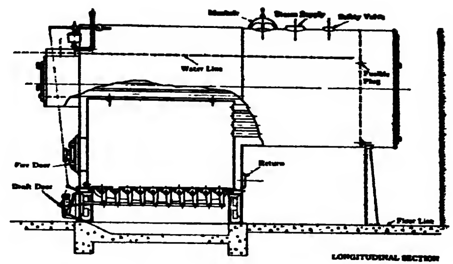
SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No.		376	377	378	379	380	381	382	383	384	385	386	387	388	389	390
SBI Rating Steam, Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Horsepower	Hp	25	29	32	36	43	50	61	72	89	107	125	143	179	214	250
SBI Rating Oil, Gas, Stoker	Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Horsepower	Hp	30	35	39	44	52	61	74	87	109	130	152	174	217	261	304
Width and Length	In. x Ft In.	42x8-3	42x9-3	48x8-9	48x9-6½	54x9-4	54x12-11	60x12-8½	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0½	78x20-7½	84x19-11	84x22-8
Overall Height Shell	In.	80½	80½	86½	86½	94½	94½	102	102	108	108	114	116	116	126	126
Height of Water Line	In.	73	73	76	76	83	81	88	88	92½	92½	98	98½	98½	110	110
Shipping Weight Coal	125 Lb	8000	9000	10000	10500	12000	15000	17500	19500	23000	26500	29500	32000	38500	44500	49500
Oil, Gas or Stoker	125 Lb	7400	8510	9010	9520	11130	13910	16080	18260	21530	24820	27550	30290	36380	41900	46870

Type "C" Boiler Hi-Firebox Series 7L70 for Oil, Gas or Stoker



Type "C" (cut-away View) 27L70 for Hand-fired Coal



"5000" Series Diagram
Welded Firebox Boilers Updraft 5176–5190 Mechanical rating same dimension as 5076–5090 Hand-fired. Smokeless Downdraft 6177–6190 Mechanical rating same dimensions as 6077–6090 Hand-fired.

SPECIFICATIONS—TYPE "C" HI-FIREBOX WELDED BOILERS — 7L70 Series & 27L70 Series

Boiler No. Oil, Gas, Stoker	7L73	7L74	7L75	7L76	7L77	7L78	7L79	7L80	7L81	7L82	7L83	7L84	7L85	7L86	7L87	7L88	7L89	7L90
SBI Rating Steam . . Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
SBI Net Rating . . Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Width x Lgth., In. x Ft In.	36x5-10½	36x6-4½	36x6-10½	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height Shell . In.	84	84	84	84	88½	88½	88½	92½	92½	105½	105½	115½	120½	130½	130½	138	162½	162½
Height of Water Line In.	75	75	75	75	76½	76½	76½	79½	79½	90	90	100½	103	113	113	119	141½	141½
Shipping Weight . . Lb	3100	3500	3900	4400	4900	5400	5800	6700	7600	8800	10000	11800	13600	15300	17000	20100	23200	26100
Boiler No. Hand-fired Coal	27L73	27L74	27L75	27L76	27L77	27L78	27L79	27L80	27L81	27L82	27L83	27L84	27L85	27L86	27L87	27L88	27L89	27L90
SBI Rating Steam Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
SBI Net Rating . . Sq Ft	1830	2170	2500	2920	3330	3750	4170	5000	5830	7080	8330	10420	12500	14580	16670	20830	25000	29170
Shipping Weight . . Lb	3600	4000	4500	5100	5600	6200	6700	7800	8800	10200	11500	13600	15700	17600	19500	23200	26700	30200

SPECIFICATIONS—WELDED FIREBOX BOILER — "5000" Series & "6000" Series

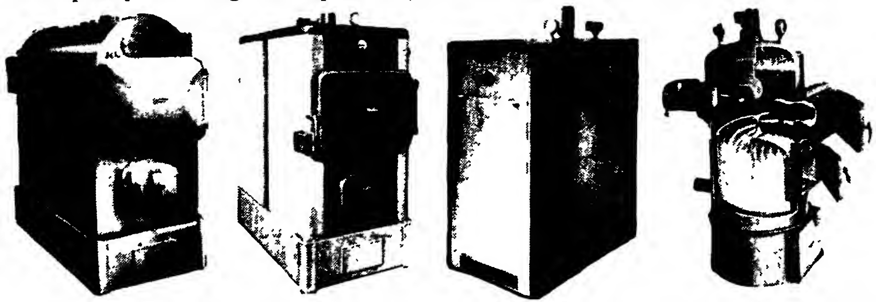
Boiler No. Updraft	5076	5077	5078	5079	5080	5081	5082	5083	5084	5085	5086	5087	5088	5089	5090
SBI Rating Steam, Coal . . Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
SBI Net Rating . . . . Sq Ft	2920	3330	3750	4170	5000	5830	7080	8330	10420	12500	14580	16670	20830	25000	29170
SBI Rating Steam: Oil, Gas, Stoker . . . . Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
SBI Net Rating . Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Width x Length . . In. x Ft In.	42x7-3	42x8-0	42x8-11	42x9-10	48x9-9	48x11-1½	54x11-1	54x12-9	60x13-2½	60x15-3	66x14-5½	66x16-0½	72x16-8½	78x17-5½	78x19-8½
Overall Height. In.	81	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line. In.	70	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Shipping Weight, Coal . Lb	5100	5600	6100	6600	7500	8400	9600	10800	12600	14400	16100	17900	21200	24400	27500
Boiler No. Downdraft		6077	6078	6079	6080	6081	6082	6083	6084	6085	6086	6087	6088	6089	6090
Shipping Weight, Coal . . Lb		5800	6300	6800	7700	8600	9900	11200	13100	14900	16600	18300	21500	24500	27300

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler

Table for two series of Boilers lists maximum dimensions only.

KEWANEE TYPE "R" RESIDENCE BOILERS

Kewanee Type "R" Boilers are especially designed and constructed to meet all heating and hot water requirements for homes and small buildings. Every kind of solid fuel, coke, all grades of hard or soft coals and their briquette or treated forms are burned with excellent results. Also, any liquid fuel, oil, and natural or commercial gas can be used with high efficiency.
Standard jackets will be available for Round "R" and Square "R" Boilers. Hot Water Copper Coil may be ordered for Round and Square "R" Boilers, 55 to 720 gallons capacity for storage tank operation, also 135 to 700 gallons instantaneous flow.



SPECIFICATIONS—RESIDENCE BOILERS SQUARE "R" ROUND TYPE "R"

Boiler No.		3R1	3R2	3R3	3R4	3R5	3R6	3R7	3R8	1733	734	735	736	1737
Net Load Steam, Handfired	Sq Ft	740	910	1080	1230	1480	1810	2140	2480		330	450	570	
Water	Sq Ft	1180	1450	1720	1970	2370	2890	3430	3970		530	720	910	
Btu per Hour	1000's	178	218	261	295	355	434	514	595		79	108	137	
Net Load Steam, Mechanical	Sq Ft	900	1100	1300	1500	1800	2200	2600	3000	275	400	550	700	900
Water	Sq Ft	1440	1760	2080	2400	2880	3520	4160	4800	440	640	880	1120	1440
Btu per Hour	1000's	216	264	312	360	432	528	624	720	66	96	132	168	216
Heating Surface	Sq Ft	53	65	77	88	106	129	153	177	16	24	32	41	53
Grate Area	Sq Ft	4.2	4.9	5.7	6.3	7.2	8.4	9.4	10.5		2.2	2.9	3.7	
Furnace Volume	Cu Ft	8.2	10.0	11.8	13.6	16.4	20.0	23.6	27.3	3.1	4.4	6.4	8.9	8.9
Width and Length	In x In.	30x30½	30x36½	30x42½	30x48½	34x42½	34x50½	34x58	34x67					
Diameter Firebox Inside	In.									17	20	23	26	26
Overall Height from Floor	In.	65	65	65	65	72½	72½	72½	72½	54	54½	58	60½	60½
Height of Water Line	In.	55½	55½	55½	55½	62½	62½	62½	62½	46	44	47	49	49
Shipping Weight, Hand UnJ.	Lb	1775	1975	2175	2350	2600	2900	3200	3500		880	1040	1280	
Mech UnJ.	Lb	1500	1650	1800	1950	2250	2475	2725	2000	600	800	920	1130	1400
Outside Surface to Cover	Sq Ft	38	43	48	53	58	65	73	82	17	24	35	41	47

KEWANEE STORAGE WATER HEATERS

Kewanee Storage Heaters—use exhaust or live steam. 15 standard Coil Elements in 29 standard size storage tanks. Capacities 95 to 2240 gals per hr.

KEWANEE SCOTCH MARINE BOILERS

Kewanee Scottie Junior for High Pressure Steam in small industrial usage. 6 sizes 6 to 30 hp at 100 lbs steam pressure.

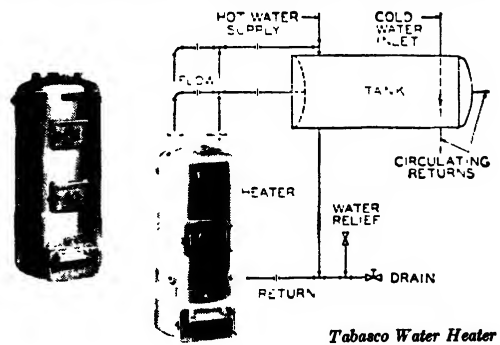
Kewanee Welded Scotch Marine, Low Pressure Steam. 13 sizes. 5470 to 42,500 sq ft, mechanically fired.

KEWANEE HI-TEST BOILER

Kewanee Hi-Test Fusion Welded Series for High Pressure Steam in power or industrial process. 6 stock sizes, 50 to 150 hp, 125 and 150 lbs steam working pressure. All *A.S.M.E.* Code.

KEWANEE WATER HEATERS

All-Weld Water Heaters, Tabasco or Garbage Burning, 5 sizes to heat 165–700 gal 50 deg per hour. Extra Heavy W. P. 100 lb per sq in. 150 lb Test.



Tabasco Water Heater

Johnston Brothers, Inc.

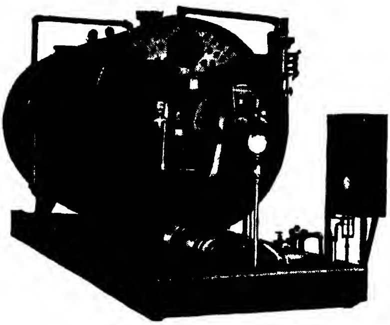
ESTABLISHED 1864

Ferrysburg, Michigan

Member SBI Reg. U. S. Pat. Off.

"PACKAGED"
STEAM BOILER UNIT
FULLY AUTOMATIC
HEAVY OIL

WATER-BACK TYPE,
FORCED DRAFT, THREE-PASS,
TROUBLE-FREE, EFFICIENT.



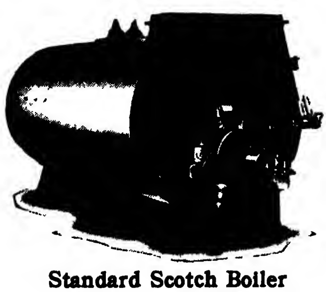
Rating guaranteed and overloads readily obtained.

HIGH PRESSURE TYPE from 50 to 500 hp and pressures of 125, 150 & 200 lb psi.

LOW PRESSURE TYPE for heating; 15 lb pressure, and EDR rating from 2190 sq ft to 42,500 sq ft steam.

A.S.M.E. Code construction. Ask for Bulletin 505.

Firebox Heating Boiler, Compact Type. Oil, Gas, Stoker or Hand Firing. Three-Pass 15 lb pressure. *A.S.M.E.* Code. Capacities 2190 to 42,500 sq ft, SBI rating. Ask for Bulletin 1500.



Standard Scotch Boiler Water-back Type, Natural Draft.

POWER, PROCESS, HEATING.

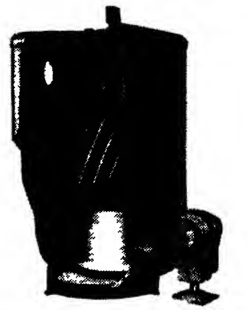
25 to 300 hp and pressures 15 lb to 200 lb psi. For mechanical firing with Coal, Gas or Oil. Overloads up to 200 per cent of rating readily developed. Oil or Gas firing equipment and controls can be installed for completely automatic operation and in emergency or fuel scarcity, can be readily converted to Coal firing.

Ask for Bulletin 4000

Mt. Hawley Mfg. Co.

Mt. Hawley Airport, Peoria, Illinois

Manufacturers of *ASME* Oil and Gas-fired Boiler-Burner Units, Water Heaters, Conversion Oil Burners



New Spiral Tube Design (Pat. applied for) extracts more heat; lowers stack temperature approximately 100 deg.

All-Steel. Sides and top heavily insulated. Wet base. Pre-cast combustion chamber or tankless heater easily replaced.

Underwriters Approved oil burners; *AGA* approved gas units to burn natural, manufactured, or LP gas.

Simplified Installation. Completely assembled at factory.

Water Heaters. All models available galvanized as water heaters for commercial and institutional use.

SIZES AND SPECIFICATIONS—MT. HAWLEY BOILER-BURNER UNITS

	DOMESTIC			COMMERCIAL			
Boiler Model Number	No. 10	No. 16	No 25	No. 40	No 55	No 75	No. 100
Rating Sq. Ft. HW Standing Radiation	540	920	1400	2240	3072	4160	5600
Rating Sq Ft. Steam Standing Radiation	340	575	875	1400	1920	2600	3500
B t.u. Output per hour (max)	104000	172000	261000	417000	574000	782000	1043000
H P.	2 3	4-6	6-8	10-12	12-16	18-22	25-30
Sq. Ft Heating Surface	20	32	43	80	131	174	232
Capacity of blr gph dely-100° rise when used as Water Heater	120	181.5	275	440	605	825	1100
Water Capacity of Boiler, gal.	20	27	37	62	106	155	194
*G P. H. Capacity Water Heating Coil (tankless type)	120	120	280	320	450	600	600
Mt Hawley burner model recom.	G-2	G-2	G-4	G-4	G-6A	G-6B	G-6B
Firing Rate Recommended, G P H	.75-1 00	1.00-1.65	1 65 2.50	2.50-4.00	4.00-5.50	5.50-7 50	7.50-10.00
Boiler Main Shell: Diameter.	17½"	20½"	24½"	29"	34"	39"	44½"
Boiler Main Shell: Gauge of Steel	¼"	¼"	¼"	¼"	¼"	¼"	¼"
Boiler Main Shell: Height	48¼"	48¼"	48¼"	57¼"	67"	79"	79"
Tubes: Diameter O. D.	2"	2"	2"	2"	2"	2"	2"
Tubes: Gauge of Steel	12 ga.	12 ga.	12 ga.	12 ga.	12 ga.	12 ga.	12 ga.
Tubes: No. in each size blr	10	19	28	47	65	88	124
Flue Heads ff Gauge of Steel.	5/16"	5/16"	5/16"	5/16"	5/16"	5/16"	5/16"
Jacket: Shape	Square	Square	Square	Round	Round	Round	Round
Jacket: Diameter or width	20½"	23½"	27½"	31"	36"	41"	46½"
Jacket: Length	22½"	25½"	29½"	—	—	—	—
Jacket: Height	52½"	52½"	52½"	59¾"	69"	81½"	81½"
Height Water Line	32½"	32½"	32½"	42"	49¼"	53"	53"
Steam Outlet: Diameter	2"	2"	2"	2½"	3"	4"	4"
Steam Outlet: Height above floor	39¾"	39¾"	39¾"	49"	57¼"	67"	67"
Diameter of return	2"	2"	2"	2½"	3"	4"	4"
Smoke Outlet: Diameter	8"	8"	9"	10"	12"	12"	12"
Smoke Outlet: Height above floor	43½"	43½"	43½"	51¼"	60¾"	71¾"	71¾"
Diameter Burner Opening	4⅝"	4⅝"	4⅝"	4⅝"	5 7/16"	5 7/16"	5 7/16"
Chimney Size Recommended.	8x8"	8x8"	9x9"	12x12"	12x16"	16x16"	16x16"
Shipping Weight—boiler	612	763	1009	1530	2280	2700	3350
Shipping Weight—burner	60	60	72	72	82	82	82
Shipping Weight—total	672	823	1081	1602	2362	2782	3432

* Estimated—Based on 180° boiler water, 100° temperature rise.

Boiler shipping weights are estimated since almost every shipment will vary in weight due to lumber and building of the crate.

Write for complete specifications on Mt. Hawley Boiler-Burner Units and gun-type oil burners—0.65 to 20.0 gph. New Mt. Hawley high-efficiency combustion head especially designed for catalytic oils available.

THE TITUSVILLE IRON WORKS CO.

TITUSVILLE, PENNSYLVANIA, U.S.A.

DIVISION OF STRUTHERS WELLS CORPORATION

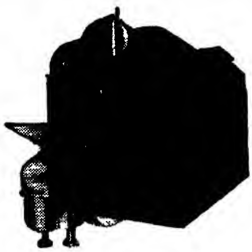
American Headquarters for Dependable A.S.M.E. CODE BOILERS...since 1860

Type Designation CM

Titusville Compact Steel Heating Boilers built in 19 sizes ranging from 129 square feet to 2500 square feet heating surface, and maximum steam working pressure 15 psig.

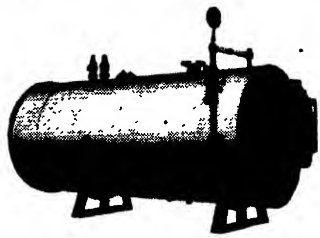
Type Designation RP

Titusville Portable High Pressure Firebox Boilers built in 13 sizes ranging from 250 square feet to 2500 square feet heating surface, steam pressure 100 psig and 125 psig.



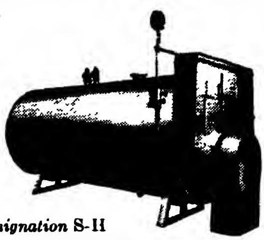
Type Designation Ticotherm

Titusville Ticotherm Steam Generators built in 13 sizes ranging from 1000 square feet to 5000 square feet heating surface. Pressures 160 psig, 200 psig, 250 psig and higher.



Type Designation Wee Scot to 60 HP, SP *above 60 HP*

Titusville Scotch Marine Power Boilers built in 13 sizes ranging from 97 square feet to 3000 square feet heating surface. Pressures 125 psig and 150 psig.



Type Designation S-H

Titusville Scotch Marine Heating Boilers built in 19 sizes ranging from 129 square feet to 2500 square feet heating surface, and maximum steam working pressure 15 psig.

Type Designation TDL

Titusville Three Drum Low Head Water Tube Boilers built in numerous sizes ranging from 729 square feet to 6109 square feet heating surface. Pressures 160 psig, 200 psig, 250 psig and higher.

Descriptive technical literature is available on request. To insure you the highest quality and fastest possible delivery—The Titusville Iron Works Company combines the production facilities of three great plants with 86 years experience in building dependable boilers, complying in every detail with *ASME* Code Rules.

The Webster Engineering Company

419 West 2nd St., **Tulsa, Oklahoma**

Division of

SURFACE COMBUSTION CORPORATION, TOLEDO, OHIO

FOR STEEL FIREBOX, SECTIONAL, WATER TUBE and HRT BOILERS

SERIES F600

Low Pressure Gas

This is an outstanding vertical atmospheric venturi gas burner, with the radiant refractory top that utilizes straight natural gas and mixed gas to 800 Btu at pressures from 2″w.c. to 8 ounces. Series F600 Burner Assemblies are available with ratings from 50,000 to 10,000,000 Btu/hr for firing all types of steel firebox and sectional boilers.

SERIES 200

Intermediate Pressure Gas with Combination Oil.

The Series 200 Webster Radiant Combination Gas and Oil Burner is a straight atmospheric horizontal firing assembly that makes it possible to utilize natural, mixed or manufactured gas in combination with all types of fuel oil.

Recommended for application to refractory furnaces, it is designed for firing Water Tube and HRT boilers when oil fuel is used over extended periods and a gas pressure of at least 1 lb is available.

SERIES 1

Low Pressure Gas and Standby Oil.

It is a 100 per cent secondary air-horizontal firing, multi-jet burner that operates equally well with natural, mixed or manufactured gas. It is suitable for firing Water Tube and HRT boilers and, should design conditions prevent the use of a vertical burner such as Series F600 or Series 650 or the KINETIC burner, the Series 1 will prove very acceptable for firing steel firebox and sectional boilers when combustion space is not limited.

KINETIC

Low Pressure Gas and Standby Oil.

Consisting of a multiplicity of full venturi mixers with flame retention nozzles assembled in a metal casing complete with pilot and louvre, the KINETIC burner is presented as the latest addition to the Webster line of firing equipment. It may be used with natural, mixed and liquefied petroleum gases.

Being a multiple head assembly, the Kinetic burner can be supplied in any size or shape. The firing of Scotch Marine boilers with extremely low pressure gas without noise, vibration or electrical power and with minimum furnace draft was the sole objective when the Kinetic burner was developed. When design conditions preclude the use of a vertical burner in a steel firebox or sectional boiler, the Kinetic burner is highly recommended as the next best gas application.

Ace Engineering Company

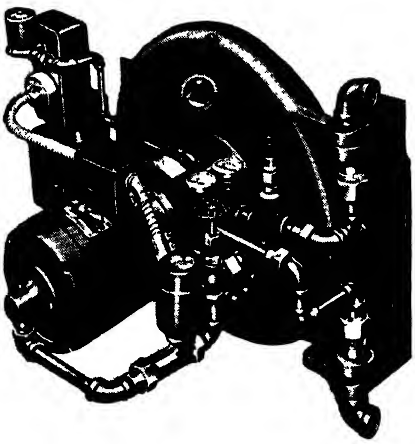
1435 West 15th St., **Chicago 8, Ill.**

"Custom Engineered Oil Burning Systems Since 1931"

The Ace Uniflow oil burner is built in ten sizes to burn all grades of fuel oil. The oil pump is available either as an integral part of the burner or as a separate unit, depending upon the job requirements. Standard models include both belt and direct drive and for operation with all electrical current characteristics.

The Ace oil burner features as standard equipment the patented Ace Uniflow valve, which permits constant uniform flow and flame regardless of oil temperature or viscosity.

Approved by Underwriters' Laboratories, Inc., New York Board of Standards & Appeals and the States of Massachusetts and Connecticut, these burners are designed to oil fire commercial and industrial boilers in the capacity ranges as shown below.



TECHNICAL DATA AND SPECIFICATIONS

Burner Size	Maximum G. P. H.	Boiler Horse Power	Sq. Ft. Steam Radiation	Motor Size H. P.
12	8	24	3,300	1/3
14	15	45	6,000	1/2
15	22	66	8,800	1/2
16	30	90	12,000	3/4
17	45	135	18,000	1
18	70	210	28,000	1.5
19	85	255	34,000	2.0
20	100	300	40,000	3.0
22	130	400	52,000	5.0
24	165	500	64,000	5.0

ACE AIR NOZZLE

(Patent Pending)

The Ace variable-vaned air nozzle permits accurate, efficient shaping of the flame to conform with the shape of the combustion chamber. The angle of the vanes can easily be changed in the field to any desired position to suit the boiler requirements and thus insure the maximum flame without oil impingement.

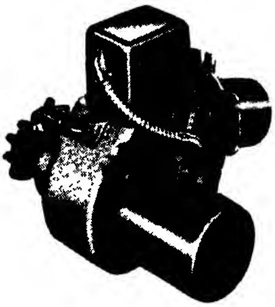
SEND FOR ACE MANUAL...
PREPARED SPECIFICALLY FOR ARCHITECTS AND ENGINEERS.

ABC Oil Burner

Automatic Burner Corp.

1823 Carroll Ave.
Chicago 12, Ill.

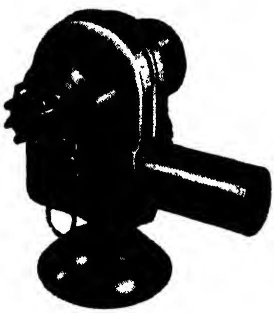
THE TRADE MARK OF QUALITY OIL BURNERS FOR 29 YEARS



PRESSURE TYPE OIL BURNER

Model 51

Here is the burner which meets every demand of the complete range of domestic heating needs. It combines beauty of design, excellence of engineering and precision production . . . to deliver guaranteed performance. The Model 51 mounts directly to the boiler by means of a flange, accurately machined for precise fit. Its $\frac{1}{8}$th hp motor insures trouble-free, amazingly quiet operation. It is attractively finished in the nonrusting, wear-resistant beauty of Hammerloid. Available in standard and odd cycles and voltages.



PRESSURE TYPE OIL BURNER

Model 52-A

This gun type oil burner has the famous ABC Oilairator mechanism that atomizes the fuel, mixes it with air, and delivers it with correct twist and velocity to insure complete combustion. The ABC nozzle, together with the proper choke and turbulator guarantees precise control and flexibility . . . guarantees a thoroughly satisfactory economical home heating unit. Capacity 0.6 to 6.0 gph.

RANGE BURNER

This ABC oil range burner offers the advantages of all-steel construction, complete interchangeability of parts, and precision manufacture. Tight joints insure constant efficiency of the flame. Easy to clean . . . easy to operate . . . ideal for stoves, ranges, water heaters. Uses kerosene as a fuel.

ABC . . . MAKERS OF

Domestic Boiler and Furnace Units . . . Range Burners . . . Water Heaters . . . Special Products.

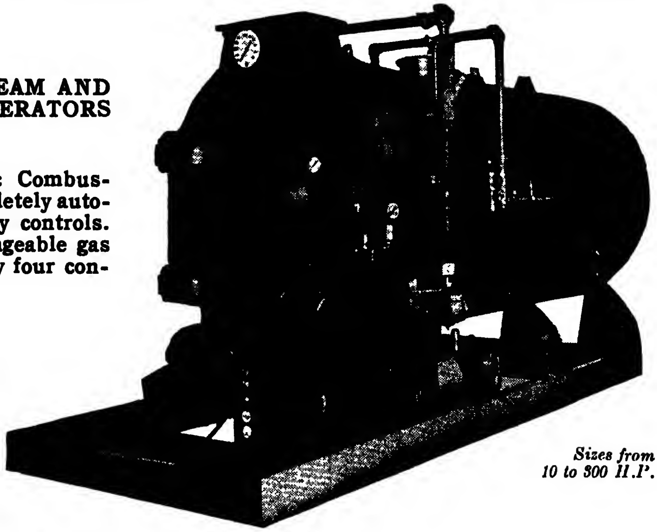
Cyclotherm Corporation

90 Broad Street, New York 4, N. Y.

Factory Distributors throughout the United States
Sole Export Distributor: Drake America Corp., 20 E. 50th St., New York 22, N. Y.

CYCLOTHERM STEAM AND HOT WATER GENERATORS

Operate on Cyclonic Combustion principle. Completely automatic with full safety controls. Oil, gas or interchangeable gas and oil firing. Only four connections required.



Sizes from 10 to 300 H.P.

The Cyclonic Combustion operating principle causes the fuel to be burned while spiraling at high velocity around the inside wall of the combustion chamber, forming virtually a tube of flame. This provides uniform and highly efficient heat transfer over the entire furnace wall without hot spots. Complete utilization of heat transfer areas provides quick heat as well as greater than 80 per cent efficiency with the reduced maintenance of a two-pass design.

10 to 50 hp models are equipped with "on-off" control; 80 to 300 hp models have modulated control. (60 hp models available with either control form.) Automatic combustion safeguard, low water and other safety controls are included.

Pressure Range

Available in standard low pressure units or high pressures up to 200 psi.

Fuel Burner Arrangements

Fuel	Cyclotherm Size
Light Oil, 1 to 3	10 to 300 HP
Heavy Oil, 5, 6 and Bunker C	80 to 300 HP
Gas, mfd., mixed and natural	10 to 300 HP
Comb. light oil and gas	10 to 300 HP
Comb. heavy oil and gas	80 to 300 HP

STANDARD RATINGS AND DIMENSIONS

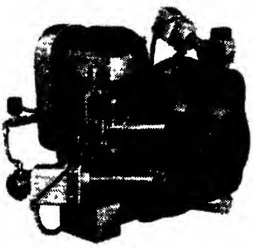
Boiler H.P.	Output BTU per Hr.	Steam Units: Steam per Hr. (Pounds)	Steam Units: Equivalent Direct Radiation (Sq. Ft.)	Hot Water Units: Gallons per Hr. (100° Rise)	Hot Water Units: Equivalent Direct Radiation (Sq. Ft.)	Overall Dimensions (Inches): Length	Height	Width
10	335,000	345	1,395	410	2,230	66	41	40
15	502,500	520	2,090	620	3,350	84 3/4	48	44
20	670,000	690	2,790	825	4,465	93 3/8	48	44
30	1,005,000	1035	4,185	1,240	6,700	110 1/4	58 1/2	55
40	1,340,000	1380	5,580	1,655	8,930	131 1/4	58 1/2	55
50	1,675,000	1725	6,975	2,065	11,165	133	69 3/4	62
60	2,010,000	2070	8,375	2,480	13,400	152	69 3/8	62
80	2,680,000	2760	11,165	3,305	17,865	174 1/2	78 3/8	66 1/2
100	3,350,000	3450	13,955	4,135	22,330	173 5/8	84 3/8	75
125	4,187,500	4312.5	17,445	5,170	27,915	173 5/8	91 1/2	80
150	5,025,000	5175	20,935	6,205	33,500	200 7/8	91 1/2	80
200	6,700,000	6900	27,915	8,270	44,665	227 7/8	98 1/2	87
300	10,050,000	10,350	41,850	12,405	66,900	237	110 5/8	92 1/2

Enterprise Engine & Foundry Co.

BURNER DIVISION

18th & Florida Sts., **San Francisco 10, Calif.**

Distributors in Principal Cities



ENTERPRISE V-BELT HORIZONTAL ROTARY TYPE OIL BURNERS

Enterprise burners are completely self-contained. The fan and atomizing cup are mounted on a rigid tubular shaft, of ample inside diameter to allow even flow of oil. The burner shaft is mounted in ball bearings and is V-belt driven. Shaft rotation speed is maintained regardless of motor RPM. The flexible drive arrangement permits interchange of modern type motors.

Oil metered into burner passes through main shaft to atomizing cup, then into primary airstream, where it is converted to a combustible fog. Air surrounds each drop of oil, giving the right density for complete, efficient burning.

OUTSTANDING METERING PUMP FEATURES:

1. Fully field tested under practical operating conditions.
2. Constant oil delivery regardless of viscosity changes.
3. Two-stage unit, includes primary gear pump and secondary metering pump.
4. Adaptable for automatic modulating control or manual control.
5. Oil reservoir provides self-priming, and bleeds air through line in case of leaky suction line.
6. Eliminates relief valves and oil pressure adjustments.
7. Eliminates metering valves.
8. Pump completely immersed in oil.
9. Reduces pressure on return line.
10. Secondary metering pump adaptable to gravity type heavy oil burners operating from single pump set. Allows reduced pressure.

BURNER CAPACITIES

Burner Size	Steam 240 BTU	Hot Water 150 BTU	Boiler HP	Gals. Per Hr.
AA	1,650	2,650	12	4
A	2,800	4,500	20	7
C	5,600	6,900	40	15
E	9,000	14,400	65	22
G2	16,000	25,500	115	35
H2	23,000	36,750	165	50
J2	31,500	50,400	225	70
K2	46,100	73,750	330	100
L2	62,900	100,650	450	135
M2	90,000	145,300	650	200

COMBINATION OIL AND GAS BURNERS

Enterprise Oil-Gas Burners can be used with whichever fuel is more readily obtainable. The gas unit is a ring type, multijet head mounted on standard burner, with primary air for either fuel supplied by the same fan. Gas jet orifices are factory drilled for individual requirements depending upon pressure, specific gravity and Btu content of gas. Burner operation and firebox construction are the same with either fuel. Burners are available with manual, semi or full automatic control. A constant burning gas pilot is advisable when gas fuel is used.

In all combination gas-oil installations and in large oil installations, electronic controls are recommended. Automatic burners are supplied with standard gas-electric ignition and can be arranged for constant gas pilot with gas fuel, and for intermittent pilot with oil fuel. Pilot is automatically lighted for each start and extinguished when burner shuts down on safety. Enterprise factory engineers should be consulted for complete wiring diagrams.

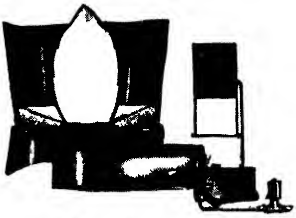
Enterprise Burners are Approved by *Underwriters' Laboratories, Inc.*

Trade Mark

H.C.Little Burner Co.

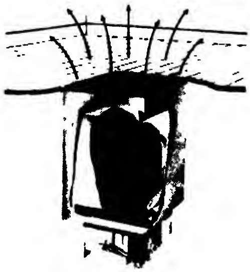
Head Office: **San Rafael, Calif.**

Factory Representatives in 18 Principal Cities



Natural Draft, Automatic, Vaporizing Burner

Natural draft vaporizing burner, specifically designed for low cost catalytic furnace oil and listed by the *Underwriters' Laboratories*. Cutaway model shown in typical conversion installation (with controls attached) is fully automatic, with electric ignition (no pilot light) and thermostatic control. Manually operated models, which do not require electricity, are also listed for low cost catalytic oil. Seven sizes: 1.45 qts to 2½ gal per hour.



Fully Automatic Oil Burning Floor Furnace

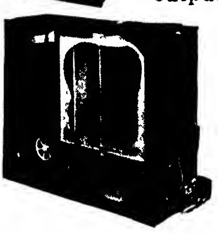
These low cost units are factory assembled, including all controls, ready to hang in floor. Easy installation, no basement needed. Natural draft vaporizing burner, automatic operation, with electric ignition and thermostatic control. Inexpensive to operate. Listed by Underwriters' Laboratories for low cost catalytic furnace oil. Two sizes: No. **70-47**—47,250 Btu output (22 in. wide x 28 in. long x 44 in. high); No. **100-47**—75,000 Btu output (22 in. wide x 40 in. long x 45½ in. high).

Utility Room Furnace

Ideal for modern homes without basement. Compact, fits neatly in small utility room. Overall size: 20 in. wide x 32 in. deep x 73½ in. Very high efficiency. Two filters, 16 in. x 20 in. x 2 in.; 9 in. blower, 1000 cfm capacity at ¼ in. SP. Cold air return bottom or sides. 84,000 Btu output.

Coil Type Water Heater

For radiant heating in small homes or for hot water supply in industrial and commercial use. Employs separate hot water storage tank. Two sizes: No. 150—106,960 Btu output, 150 gph (80° rise); No. 250—160,440 Btu output, 250 gph (80° rise).

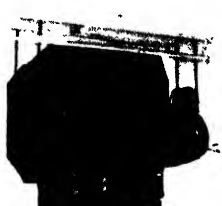


Winter Air Conditioner

Size A, 85,000 Btu output. Size B, 125,000 Btu output. Multivane blower, filters, vaporizing burner.

Ceiling Furnace

Industrial overhead heating saves floor space. Units are factory assembled, including vaporizing oil burner (no refractory brick saves much weight), thermostatic control, high capacity heat distributing blower with ¼ hp motor. Conforms to requirement of the Underwriters' Laboratories for use in garages. 140,000 Btu output.



A complete line of oil-fired small home heating units —plus special industrial units of moderate capacity.

S. T. Johnson Co.

Builders of Domestic and Commercial Oil Burners

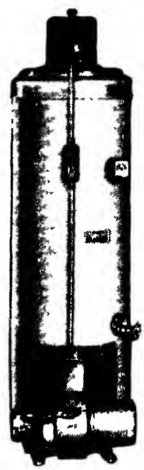
940 Arlington Ave., **Oakland 8, Calif.**
401 No. Broad St., **Philadelphia 8, Pa.**

Self-storage water heaters, separate burner units, burner-boiler units, conditioned air units, range burners and various specialized items comprise the line-up of Johnson light-oil Burners.

There is a wide range of sizes and capacities in each classification with which heating engineers and contractors can successfully meet every type of problem.

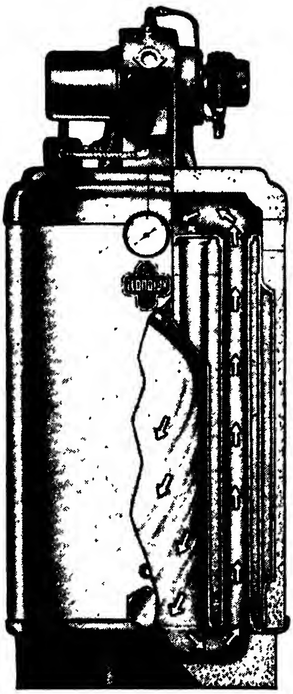
Every Johnson Burner is backed by an unbroken 45-year record of fine engineering and excellent craftsmanship.

←
BANK-HEAT BURNERS
Fully automatic, pressure atomizing type. Sizes up to 20 gph.

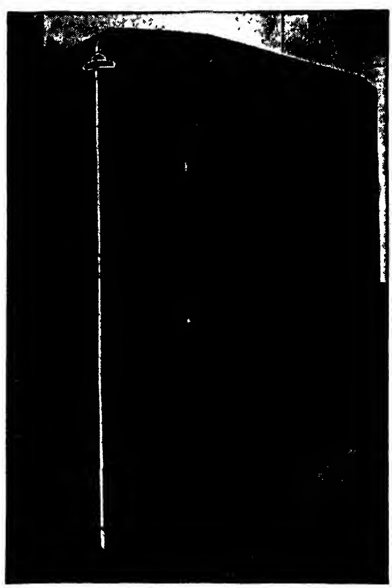


HEATLUX HOUSE HEATERS
Fully automatic Bankheat Burners. Hot water heat. 2 sizes.

AQULUX WATER HEATERS
Fully automatic, self-storage. Capacity: 100 to 540 gph.



←
ECONOLUX HEATERS
Heat and hot water. Fully automatic Bankheat Burners.



→
SELECTAIR HEATERS
Hot air, hot water or steam heat. 3 sizes. Bankheat Burners.

S. T. Johnson Co.

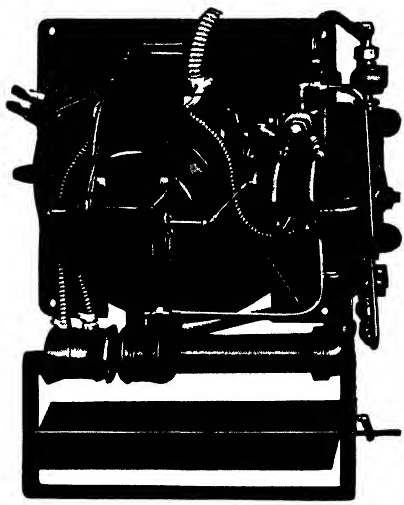
Builders of Heavy-Duty Industrial Oil Burners

940 Arlington Ave., Oakland 8, Calif.
401 No. Broad St., Philadelphia 8, Pa.

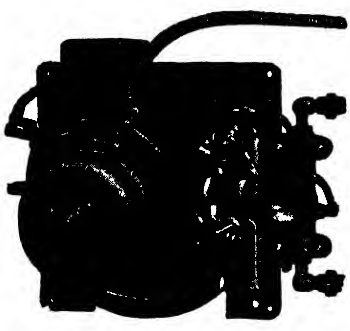
Johnson Industrial Burners are designed to operate on Heavy Oils which produce extra heat at low cost. They increase the capacity of equipment formerly fired with coal and produce desired steam pressures more quickly. Automatic regulation permits the boiler to operate with maximum efficiency at any specified steam pressure, without watching, care or attention, thus reducing labor costs.

They have been installed with marked success in hotels, hospitals, factories, office buildings and other large structures all over America because they provide heating engineers with a wide range of capacities and with every desired feature of economy, performance and automatic control.

In their design and in their construction, Johnson Burners represent the new and advanced engineering and building techniques backed by 45 years of practical experience.

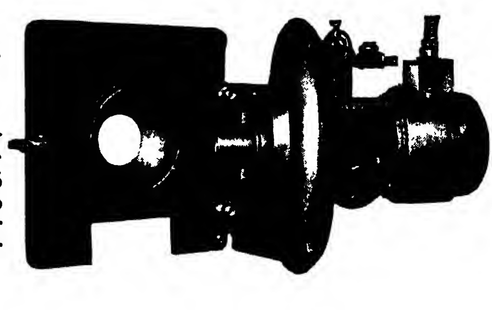


TYPE 30 AVH
Fully automatic. Pre-heater type. Burns No. 6 Oil. Six sizes, 20 to 300 horse power output.



TYPE 30 AV
Fully automatic. Burns No. 5 Oil. Six sizes, 2 to 100 gph.

TYPE 28
Manual and semi-automatic. With or without built-in pumps. Burns No. 5 and No. 6 Oils. Seven sizes, 2 to 135 gph. Illustration shows burner swung away from fire-hole-plate for easy inspection.



Petroleum Heat & Power Company

Main Office and Factory: **Stamford, Conn.**

"Since 1903" . . . Good Oil Burning Equipment . . . Fuel Oils

PETRO REG. U.S. PAT. OFF.

DOMESTIC OIL HEATING EQUIPMENT

PRESSURE ATOMIZING DOMESTIC BURNERS

Models P 20-A, P 21, P 22

Applicable to steam, hot water, or warm air systems. Burn No. 3 fuel oil (or lighter), heaviest and lowest priced fuel oil approved by Underwriters for domestic use. Constant electric ignition for reliable, safe operation. Order or inquiry should specify the type, size and rating of boiler or furnace to be fired, together with the total load.

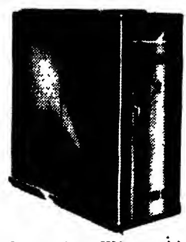
Models P 12, P 13-A, P 13

Applicable to heating large residences, stores, garages and other commercial buildings. Approved for No. 3 (or lighter) fuel oil.

Burner Model Number	Nozzle Size Gal per Hr.	Total Capacity Steam Sq Ft	Total Capacity Hot Water Sq Ft	App. Sh'p'g. Wgt.
P-20-A	1.00	350	560	120
P-20	1.25 To 2.50	435 To 875	695 To 1400	120
P-21	2.00 To 4.50	700 To 1575	1120 To 2520	120
P-22	3.00 To 6.00	1050 To 2100	1680 To 3360	135
P-12	6.00 To 10.00	2100 To 3500	3360 To 5600	190
P-13-A	9.00 To 12.00	3150 To 4200	5040 To 6720	250
P-13	12.00 To 18.00	4200 To 6300	6720 To 10080	250

UNITS AND HEATERS

Front view of Petro Standard Automatic Boiler for Steam



Petro A-75 Winter Air Conditioner

Petro Storage Type Hot Water Heater

Standard Model Petro Automatic Boilers, in two sizes, for steam and water, are steel boilers designed for small home field. Winter Air Conditioner rated 75,000 Btu at register. Storage Type Water Heaters operate on No. 3 oil and heat 120 gph 100 temperature rise.

Send for Catalog of Petro Domestic Oil Heating Equipment.

Petroleum Heat & Power Company

Main Office and Factory: **Stamford, Conn.**

Oil Burning Equipment . . . "Since 1903" . . . Fuel Oils

PETRO REG. U.S. PAT. OFF.

INDUSTRIAL AND COMMERCIAL OIL BURNING SYSTEMS

For Unheated Commercial Oils:

Model W-A—Automatic ignition and operation with synchronized control of oil and air.

Model W-SA—Semi-automatic, *i.e.*: automatic variation of firing rate with manual ignition; also available for manual variation and manual ignition.

For Heated Oils: Heavy No. 5, No. 6 (Bunker "C") oil.

Model W-AH—Automatic ignition and operation with synchronized control of oil and air, and of oil heaters.

Model W-SAH—Semi-automatic with oil heaters, *i.e.*: automatic variation of firing rate with manual ignition; also available for manual variation and manual ignition.

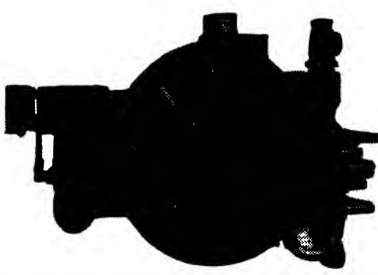
CAPACITIES

Model	Motor H.P.	Max. Gals. per Hour	Rated Cap. B.H.P.	Sq. Ft. C.I. Steam Rad.
W-2½	1/3	11	37	5,150
W-3	1/2	15	50	7,030
W-4	1/2	25	84	11,720
W-5	1	33	117	16,400
W-6	2	50	168	23,440
W-7	2	70	235	32,800
W-8	3	100	336	46,600
W-8½	3	120	403	56,200
W-9	3	145	487	68,000

These burners are available for all types of electrical current supply.

PETRO'S THERMAL VISCOSITY CONTROL

A dependable and accurate control of viscosity—and hence delivered combustion efficiency—is through the heat applied to the oil. Petro's Thermal Viscosity System controls this heat-application at its source.



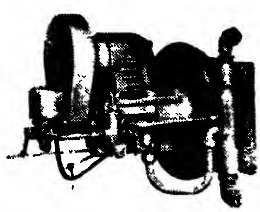
Model WD—Direct Driven, Rotary Cup Type Burner. Illustration Shows Modutrol Mounted on Burner.

PETRO MODEL W BURNER is available in two types: (1) Direct-Driven, which includes electric motor, fan, pump, rotary cup atomizer in one self-contained assembly, together with air and oil controls. (2) Belt-Driven, containing the same assembly as above except that the integral motor is replaced by motor mounted outside burner housing and arranged for belt drive. Removable rotary cup and nozzle permit changing shape of flame to suit any boiler furnace requirements.

Interlocking air and oil control mechanism permits any minimum or maximum operation required within the burner's range of operation. Counter-flow Angular Air Vanes at nozzle increase air and oil turbulence and aid efficient combustion of heavy fuel oils.

Special oil adjustment valve meters oil to rotary cup, yet permits manual operation without disturbing permanent burner adjustment.

Model WO-A Belt Driven, Rotary Cup Type Oil Burner.



Send for catalog of Petro Commercial and Industrial Oil Burners

Ray Oil Burner Co.

Since 1872

401–499 Bernal Avenue
San Francisco 12, Cal.

Distributors in
All Principal Cities of the World

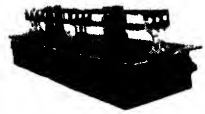
Atlantic Seaboard Division
629 Grove Street
Jersey City 2, N. J.
Consult your local
Telephone Directory

Products: **A complete line of Horizontal Rotary and Pressure Atomizing Oil Burners; Combination Oil-or-Gas Burners; Industrial Gas Burners; Oil Burning Water Heaters; Winter Air Conditioning. Units, Commercial Ranges.**

1 to 1000 Boiler hp Type AG. Manual, Semi-Automatic.

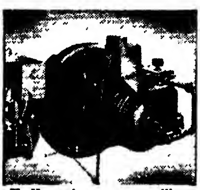
Quad-Naught for #3 oil, manual operation.

Steam Turbine Drive, Type TG, all grades oil. Tested and approved for U. S. Navy Service.



Two, 6 oven, fully automatic oil Ranges.

Type ARJP, Fully Automatic for heavy oil where gas for ignition not available.



Fully Automatic Type AR-134, for No. 5 oil

Combination Oil and High Pressure Gas, Type AR-HN.

RAY HORIZONTAL ROTARY OIL BURNERS

Built in fully automatic, semi-automatic and manual types; in sizes from 1 to 1000 Boiler hp; to burn all grades of fuel oil.

Standard models include both direct and belt drives—the latter being recommended for use where other than 50 or 60 Cycles AC, or only DC is available. Types for straight electric or straight gas ignition; pump or gravity feeds. Direct drive types include a steam turbine driven model.

All fully automatic types for heavy oil incorporate the Ray Dual Pump and Reservoir, with the Ray VISCOSITY Valve, a patented, exclusive feature which automatically meters the correct amount of fuel at all times, regardless of changes in viscosity of the oil due to temperature variations. All larger sizes employ dual ignition, for maximum reliability.

RAY PRESSURE ATOMIZING OIL BURNERS

Fully automatic, for No. 3 oil or lighter. AC or DC; capacities to 18 gal/hr.

RAY INDUSTRIAL GAS BURNERS

For gas pressures above 1 lb/sq in. may be used alone or in combination with a Ray Oil Burner. Built in eleven sizes; in capacities to 43,000,000 Btu/hr.

RAY WINTER AIR CONDITIONING UNITS

Built in four sizes, with input capacities of 105,000, 140,000, 230,000, 350,000, 450,000 Btu.

RAY OIL WATER HEATERS

Two sizes. Capacities: 45 and 75 gal. Maximum recover rates: 154 and 240 gph.

RAY OIL FIRED RANGES

Available in seven sizes for manual or fully automatic operation.

Type BR-141, Belt Drive Automatic. Available in 9 sizes from 2½ to 100 gph.

Ray Oil Furnace, Winter Air Conditioning Unit.

Small capacity (½ to 2½ gph) rotary burner.

Fully Automatic, Type JP, #3 or lighter oil.

Ray Oil Burner Co.

Since 1872

401–499 Bernal Avenue
San Francisco 12, Cal.

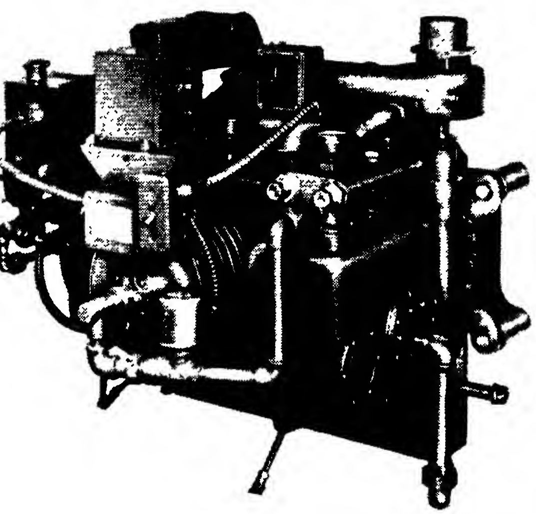
REG. U. S. PAT. OFF.

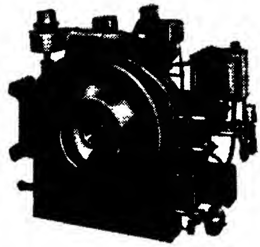
Atlantic Seaboard Division
629 Grove Street
Jersey City 2, N. J.

There is a RAY Burner for every Heating Purpose.

RAY COMBINATION OIL-GAS BURNER

Type ARC-131





Burns any grade of gas or heavy fuel oil and the operation is fully automatic for either fuel. Prevention against ignition or flame failure one hundred per cent electronically controlled. Develops from 7 to 335 Boiler hp. Quickly interchangeable in two to three minutes. Other types with choice of direct or belt drive, also sizes rated from 120 to 1000 Boiler hp.

HOURLY CAPACITY RATINGS of RAY OIL BURNERS

Burner Size	Motor H. P.	Oil Capacity U. S. Gallons		Equivalent Boiler HP		Equivalent Lbs. Steam Generated		Heat Capacity Input Thou. Btu		Equivalent Sq. Ft. Steam Radiation	
		Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
JP	[illegible]	1	3	3	9	110	325	140	420	438	1310
XPJ	[illegible]	1	4	3	12	110	430	140	560	438	1750
XPJ-1	[illegible]	3	8	9	25	325	865	420	1120	1310	3500
XP-2	[illegible]	8	18	25	56	865	1950	1120	2500	3500	7900
0000	1/10	0.5	2.5	2	8	60	290	75	375	234	1170
000	[illegible]	0.5	2.5	2	8	60	290	75	375	234	1170
00	[illegible]	1	2.5	3	8	110	290	150	375	469	1170
0	[illegible]	2	5	7	16	230	580	300	750	938	2350
1	[illegible]	4	11	13	35	460	1250	600	1650	1870	5170
2	[illegible]	5	15	17	50	580	1720	750	2250	2350	7030
3	[illegible]	8	20	27	65	930	2300	1200	3000	3750	9380
5	1	10	33	35	110	1150	3800	1500	4950	4690	15500
6	1½	12	50	40	165	1400	5800	1800	7500	5620	23400
7	2	15	67	50	225	1720	7750	2250	10000	7030	31400
8	3	25	100	85	335	2900	11500	3750	15000	11700	46900
9	5	35	150	120	500	4000	17400	5250	22500	—	—
10	7½	50	210	165	700	5800	24400	7500	31500	—	—
12	7½	75	320	250	1000	8700	37000	11250	48000	—	—

NOTE: These ratings are predicated upon specific conditions of draft and furnace volume. It may be permissible, under desirable conditions, to operate at higher rates, or advisable under restricted conditions, to operate at reduced rates.
Heating capacities are based upon 150,000 Btu per gal of oil for rotary burners and 140,000 Btu per gal of oil for pressure atomizing burners and upon an overall boiler efficiency of 75 per cent.

Williams Oil-O-Matic Division

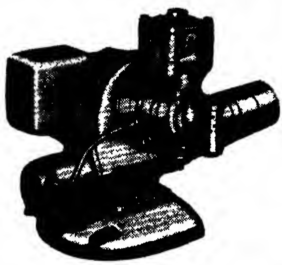
WILLIAMS OIL-O-MATIC HEATING COOLING PRODUCTS

EUREKA WILLIAMS CORPORATION, BLOOMINGTON, ILLINOIS

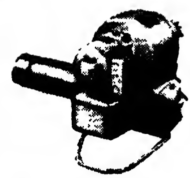
Manufacturers of Automatic and Manually Controlled Fuel Oil Burners

OIL BURNERS

Williams Oil-O-Matic Lo-Pressure burners are offered in 4 sizes ranging in capacity from ½ to 7 gal of fuel oil per operating hour. Easily installed in any type heating plant. Patented Thrift Meter pre-meters oil to exactly meet heating plant needs. Wide, non-clogging orifice handles even the heaviest oils.



Williams Hi-Pressure precision-built burners are available for those who want high quality at minimum cost. 3 sizes with capacities from 1.00 to 7 gph.



How to Decide Size of Burner

For low pressure domestic boilers, 1 gal of fuel oil per hour (140,000 Btu) is required for approximately 300 sq ft of steam radiation or its equivalent, or for 480 sq ft of hot water radiation or its equivalent. 70,000 Btu when using warm air furnace ratings. 24 sq ft steam boiler heating surface (or 2.2 hp). For exact data, see Oil-O-Matic Installation and Service Manual.

**Oil-O-Matic Lo-Pressure Burners
Leg or Flange Mt.**

Model	Oil Capacity		Motor		Atom.
	Min.	Max.	Hp	Rpm	Pressure
K-150	.50	1.50	1/10	1800	2 lbs
K-3	1.00	3.00	1/10	1800	2 lbs
K-4.5	1.35	4.5	1/8	1800	2½ lbs
K-7	4.00	7.00	1/8	1800	3 lbs

**Williams Hi-Pressure Burners
Leg or Flange Mt.**

HP-1	1.00	1.50	1/12	1725	100 lbs
HP-3A	1.35	3.00	1/12	1725	100 lbs
HP-7	3.50	7.00	1/8	1800	100 lbs

New WINTER AIR CONDITIONERS

Eye-catching beauty, superlative styling by George Walker in complete units—engineered from the inside out to take full advantage of the famous Low Pressure Principle. Capacities of units 70,000; 100,000; and 150,000 Btu's. Smallest size (Model 70) completely packaged unit only 58 in. l. 22 in. w. 45 in. h. (Fits most utility rooms).

Oil-O-Matic Winter Air Conditioners

Model	Btu at Bonnet	C.F.M. (a) ⅜" S.P.	Oil Input G.P.H	Filters Sq. In.	Blower Motor
10A	100,000	1300	1.00	1000	¼ hp
15A	150,000	1800	1.50	1920	¼ hp
70	70,000	900	.65	600	⅓ hp

OIL-O-MATIC
LOW PRESSURE PRINCIPLE

• Oil-O-Matic has a *two-source* air supply instead of only one as in the ordinary or high pressure burner. A marked aid in achieving clean, complete combustion.

• The Low Pressure Principle permits use of a large, non-clog nozzle opening (instead of the pin-point nozzle opening in ordinary or high pressure burners).

• Combination of the Low Pressure Principle and a unique Thrift Meter which permits accurate oil measurement give Oil-O-Matic top efficiency even at low firing rates.

• In independent laboratory test Oil-O-Matic has burned a *100% catalytically cracked oil* with the same ease and efficiency as a *No. 1 oil.*

New Oil-O-Matic Boilers

***New* Oil-O-Matic Products will be announced in APRIL 1949—Write in now for special information.**

The Kent Company, Inc.

433 Canal St., **Rome, N. Y.**

Representatives in Principal Cities

KENT DOUBLE SUCTION FURNACE AND BOILER CLEANER

OUTSTANDING FEATURES

Kent provides powerful double suction with two fans powered by a $\frac{3}{4}$ hp motor. The vertical design separates dust, soot and scale from air by gravity. Permits use of double dust bag. Fans fully protected—no damp soot can cake Kent fans. Dust bag fully enclosed in metal tank. May be quickly emptied by removing cover from metal tank and taking double dust bag out.

SPECIFICATIONS

Weight of Suction Unit 39 lbs.

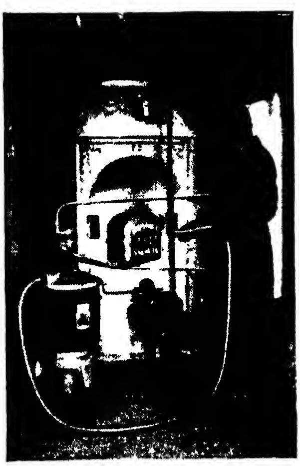
Weight of Dirt Can 24 lbs.

Total weight 63 lbs.

Diameter of Dirt Can 14 inches.

Capacity of Dust Bag 1200 cubic inches.

Air volume 200 cu ft per minute.



Built to last

Water vacuum 43 inches.

Shipping weight 116 lbs.

$\frac{3}{4}$ hp Westinghouse Universal Motor.

EASY TO CARRY

Kent is strong—yet light—easy to handle.

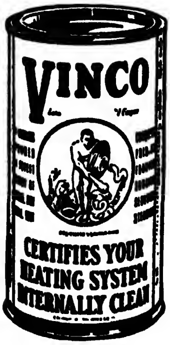
GUARANTEE

Kent products are always guaranteed against defective materials and workmanship for a period of one year.

The Vinco Company, Inc.

305 East 45th Street **New York 17, N. Y.**

Only a clean boiler can be an efficient boiler. A clean boiler means saving fuel, as well as safeguarding boiler metal.



Boiler Cleaner
3 and 5 lb. cans

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Boiler Cleaner Does

VINCO removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor, expense, and uncertain results of blowing boilers over the top or of wasting returns.*

By this thorough cleaning Vinco prevents or cures foaming, priming, surging, and slow steaming.

How Vinco Boiler Cleaner Works

Each minute grain of VINCO powder adsorbs several times its own weight of oil, grease, rust and dirt. These larger grains of adsorbed impurities then settle and are drained through the bottom according to directions on each can.

Vinco Guarantees

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.

Rust Preventer
1 qt. cans only

VINCO RUST PREVENTER

When used after VINCO Boiler Cleaner has removed oil, grease, rust, scale and dirt, it will add and keep the rust inhibiting factors at the optimal constant for a year or more. (Test kit below has complete instructions and chart.)

Vinco Field Test Kit No. 10

VINCO FIELD TEST KIT No. 10

for Testing and Treating Heating Boiler Waters

The kit enables the layman to make simple, rapid tests to diagnose and prescribe correct treatment of boiler waters right on the job.

A new time saving method that permits valid conclusions heretofore requiring complicated and often lengthy laboratory analysis and technique.

Each kit has sufficient material for complete tests on 100 jobs.

Refills cost about 2 cents for testing each job.

SPECIFICATIONS FOR COMPLETE VINCO TREATMENT OF NEW OR REMODELED STEAM, VAPOR, OR HOT WATER SYSTEMS

Do not use as a cleaning agent soda or any alkali, vinegar or any acid. Use Vinco.

1. AFTER THE SYSTEM IS TESTED AND TIGHT, USE THE PROPER QUANTITY OF VINCO LISTED.

After this first clean-out of any new or remodeled heating system, Vinco Boiler Cleaner need be used only if more piping, radiation, or another boiler is added to the original installation, or if the system is fouled by unwise cleaning or leak-sealing experiments.

2. After using Vinco Boiler Cleaner, Vinco Field Test Kit should be used to determine and apply the proper quantity of Vinco Rust Preventer. Vinco Rust Preventer should be applied annually or whenever the boiler water is drained for necessary repairs to the system.

SPECIFICATION FOR OLD HEATING SYSTEMS THAT DO NOT PERFORM PROPERLY

Diagnose and treat according to Vinco Field Test Kit. If a test kit is not available, consult table of quantities on this page and follow directions on Vinco cans.

SPECIFICATION FOR HOT WATER SYSTEMS

Use half quantities listed for treatment of steam systems to remove impurities. Then use test kit to determine proper quantity of Vinco Rust Preventer.

CONSULT THIS TABLE FOR NEW AND REMODELED HEATING SYSTEMS AND When a Vinco Field Test Kit No. 10 is not available if cleaning old heating systems.

QUANTITIES OF VINCO (IN POUNDS) REQUIRED FOR HEATING SYSTEMS

(Note that quantities are based on actual installed radiation, not on boiler capacity.)

Sq Ft of Radiation	For Steam or Vapor Systems, to prevent or cure priming or foaming. Also for Hot Water Heating Systems Maintained at approx. 200 F or above.	Annually, to remove rust scale, dirt and for Hot Water Systems below 200 F.
up to 350	3	1½
351 " 600	5	2½
601 " 1100	8	4
1101 " 1400	10	5
1401 " 1800	13	6½
1801 " 2100	15	7½
2101 " 2700	18	9
2701 " 3100	20	10
3101 " 3700	23	11½
3701 " 4200	26	13
4201 " 4600	28	14
4601 " 5000	30	15
5001 " 5300	31	15½
5301 " 5600	32	16
5601 " 5900	33	16½
5901 " 6200	34	17
6201 " 6500	35	17½
6501 " 6800	36	18
6801 " 7100	37	18½
7101 " 7400	38	19
7401 " 7700	39	19½
7701 " 8000	40	20
8001 " 8300	41	20½
8301 " 8600	42	21
8601 " 8900	43	21½
8901 " 9200	44	22
9201 " 9500	45	22½
9501 " 9800	46	23
9801 " 10100*	47	23½

*Above 10100 sq ft. use an additional pound Vinco for each additional 300 sq ft of actual installed radiation.

Soot-Off—1 lb. cans 50 and 100 lb. drums

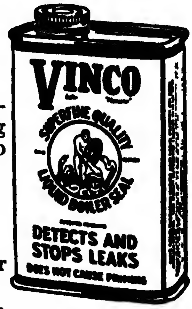
VINCO SOOT-OFF

Safely and thoroughly removes the insulating blanket of soot on fire pot, flues and chimney. It also insures against external corrosion (caused by dampness and soot forming sulfuric acid during summer layoff.) No dangerous chemicals.

REMOVE SOOT WITH VINCO SOOT-OFF SEVERAL TIMES A YEAR

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. Makes lasting repairs of boiler and heating system leaks. Fine to tighten up new jobs. Directions simple.



Liquid Boiler Seal 1 qt. cans only

Quantities

Steam and Vapor Systems—Use 1 quart VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

Hot Water Systems—Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

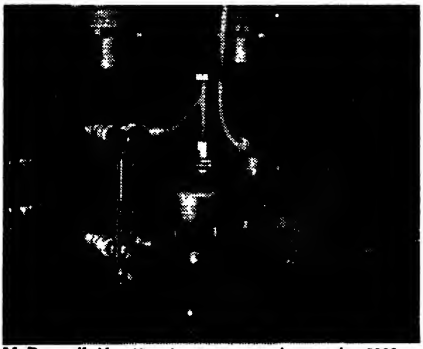
McDonnell & Miller, Inc.

Safety Devices for Steam and Hot Water Boilers and Liquid Level Controls

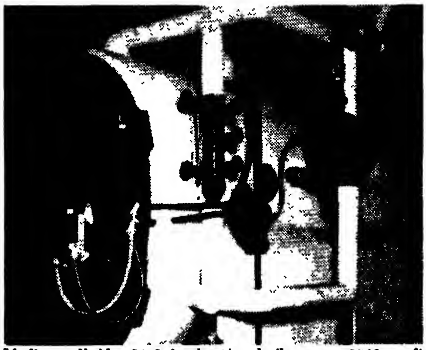
General Offices: Wrigley Building, **Chicago 11, Illinois**

"Doing one thing well"

McDonnell Combined Boiler Water Feeder and Low Water Cut-off



McDonnell No. 47-2 for heating boilers under 5000 sq ft capacity. Maximum steam pressure, 25 lbs.



McDonnell No. 51-2 for heating boilers over 5000 sq ft capacity. Maximum steam pressure, 35 lbs.

McDonnell No. 47-2, shown installed above, maintains a safe water level in boiler by feeding water whenever necessary. If emergencies such as priming or foaming permit water to fall to ¼ in. in gauge glass, cut-off switch cuts current to burner until emergency has passed. Has "Quick-Hook-Up" for fast, accurate installation right in gauge glass tappings; "cool" feed valve; extra-deep sediment chamber; *ASME*-approved blow-off valve. Also available for hand fired boilers without No. 2 switch, as No. 47 Feeder.

McDonnell No. 51-2, shown installed above, is same as No. 47-2 described at left, except it has greater feeding capacity for larger boilers, and is installed with 1 in. equalizing pipes instead of "Quick-Hook-Up." Also available for hand fired boilers without No. 2 switch, as No. 51 Feeder.

For boilers operating at higher pressures, from 35 to 75 lbs, use McDonnell No. 53-2 (or No. 53, without switch.)

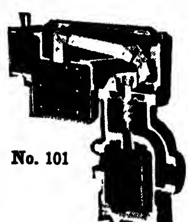
McDonnell No. 67 Low Water Cut-off

For automatically fired steam boilers of any size

Maximum steam pressure, 20 lbs.

No. 67

Has the McDonnell "Quick-Hook-Up" for quick, easy and trouble-proof installation in gauge glass tappings; deep sediment chamber with large quick-opening blow-off; packless, non-binding construction; adjustable terminal box to make wiring neat and easy; dependable snap-action twin switches. One switch can be used to sound low water alarm, or control McDonnell No. 101 Electric Water Feeder described below. Second switch cuts current to burner if water level drops to ½ in. in gauge glass.



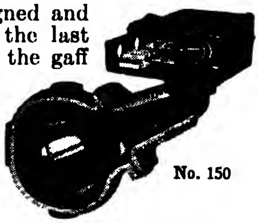
No. 101

McDonnell No. 101 Electric Boiler Water Feeder

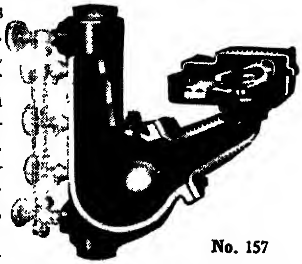
For boilers up to 5000 sq ft capacity

For use with No. 67 Low Water Cut-off or with McDonnell "built-in" Low Water Cut-offs which are standard equipment on many modern heating boilers. It converts the cut-off into a feeder cut-off combination as described at top of this page.

McDonnell Pump Control, Low Water Cut-off, and Alarm Switch
For boilers of any size; maximum steam pressure, 150 lbs.

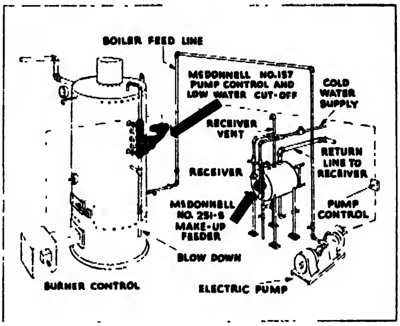

No. 150

No. 150 is designed and built, down to the last detail, to stand the gaff of high pressure and temperature. Equipped with two switches. One closes on small float drop to control electric boiler feed pump, or electric valve in line to steam pump. Second operates on greater drop to stop burner and complete low water alarm circuit. *Underwriters' Laboratories* approved. No. 150 has automatic reset; for manual reset order No. 150-M.


No. 157

No. 157 is same as No. 150, but in a new form for boilers with separate water columns. Greatly simplifies installation, assures ideal reproduction of boiler water level in float chamber, provides effective direct blow-down. No. 157 has automatic reset; for manual reset order No. 157-M.

A typical hook-up of the McDonnell No. 157, controlling an electric pump and providing low water cut-off, is shown below. When water level drops ¾ in. below normal, No. 157 starts pump and then stops it when normal water line has been restored. If emergency occurs and water level falls to arrow mark on body of control, cut-off switch stops burner. If steam pump is used, No. 157 is wired to control electric valve in steam line to pump. Note that McDonnell No. 251-S Feeder is used to supply make-up water to the pump receiver.



Drawings available covering the use of these controls on two or more boilers supplied by a common feed pump.

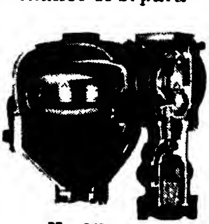
McDonnell No. 29 Safety Relief Valve and No. 247 Feeder
For Hot Water Heating Systems with Boiler and Radiation on same level.

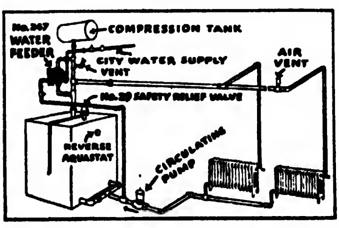
The drawing below illustrates how the McDonnell No. 29 and No. 247 team up to provide good operation as well as protection for heating systems of this sort. The No. 247 float-operated feeder maintains the proper level above the high point in the supply main, eliminating the chance of separation of the water and loss of connected load. The No. 29 Btu rated safety relief valve (see table) protects the boiler against excessive pressure when matched to the gross Btu output of the boiler.

For Hot Water Heating Boilers

Valve No.	Opening Pressure	Inlet Pipe Size	Outlet Pipe Size	Btu. Output
29	29 lbs.	1 in.	¾ in.	177,000
129	29 lbs.	1¼ in.	1 in.	289,000


No. 29


No. 247



McDonnell Safety Relief Valves for Domestic Hot Water Tanks and Heaters

Revolutionary "snap-action" provides such rapid heat dissipation the McDonnell No. 29 Series are rated, and *ASME*-certified in Btu capacity. Matched to gross Btu of heater or tank, these safety relief valves prevent excess pressure under all conditions.

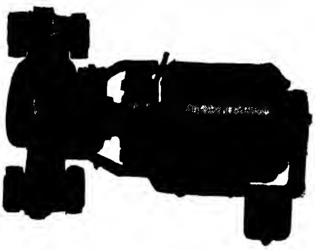
Valve No.	Opening Pressure	For use on supply pressure up to	Inlet Pipe Size	Output Pipe Size	Btu Output
229	75 lbs	50 lbs	1 in.	¾ in.	186,000
329	100 lbs	75 lbs	1 in.	¾ in.	189,000
429	150 lbs	100 lbs	1 in.	¾ in.	229,500

Bell and Gossett Company

Morton Grove, Illinois

HYDRO-FLO HOT WATER SYSTEMS AND SPECIALTIES

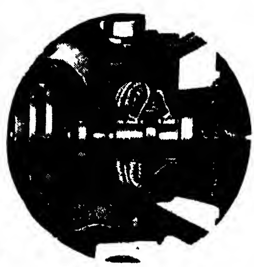
B & G Booster Pumps



The B & G Booster is the basic unit of a B & G *Hydro-Flo* Forced Hot Water Heating System. It is built as a horizontally driven unit for sound engineering reasons which have demonstrated their practical value in the thousands of installations now in operation. This construction makes possible many desirable and exclusive features.

For example, the patented water-tight Seal eliminates the need for a stuffing box. This Seal positively prevents entry of water into the bearings. The shaft is of highly polished, hardened machine steel and the close-fitted Impeller makes every revolution count by holding water slippage to a minimum.

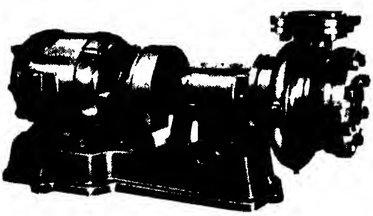
B & G Boosters have a genuine oil-circulating lubrication system—one of the greatest reasons for the quiet, dependable and economical operation of this pump. Oil is drawn up from the oil well by wool fibre wicking and dropped on the horizontal bearing surfaces. Medium grade motor oil is used and only a few drops at infrequent intervals required.

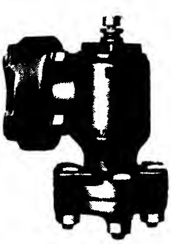


Oil Lubricated Bearings

B & G Universal Pumps →

The B & G Universal Pump is designed for large forced hot water heating systems in apartment buildings, office buildings, factories, schools, etc. The installation can be operated as a large single zone or divided into several zones in which circulation of the pumped water in each circuit is controlled by a B & G Motorized Valve, operated by a zone thermostat.





B & G Angle Flo-Control ← Valves

This valve, installed in the main, shuts off circulation to radiators when heat is not needed, permitting summer operation of a B & G Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.

B & G Motorized Valves →

Thermostatically operated valves used for controlling boiler water flow through the individual circuits of zoned heating systems.

← B & G Monoflo Fittings

B & G Monoflo Fittings permit the use of a *single* pipe main instead of the conventional flow and return lines. They are installed at the junction of the radiator risers to the single main and assure the diversion of the proper amount of heated water into each radiator. Savings in space, labor and materials are obviously effected. Available in cast-iron and copper.

SEE THE B & G HANDBOOK FOR COMPLETE DESIGNING DATA

HOT WATER SYSTEMS AND SPECIALTIES

B & G Relief Valves

For relieving excess boiler pressures in hot water heating systems, and in the lines of service water systems. B & G Relief Valves have the design features which assure dependable service.

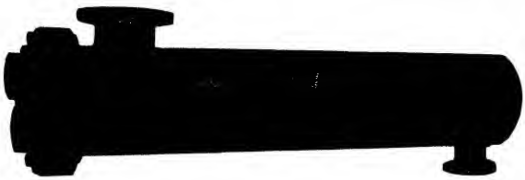
B & G Reducing Valves

Fast operating valves for keeping hot water heating systems properly filled. Easily adjusted to meet varying building heights. Also high pressure reducing valves for protection of plumbing fixtures.

B & G Type "CWU" Radiation Heater

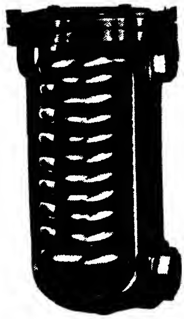
The B & G Radiation Heater is a "shell and tube" heat exchanger and is installed below the water line of a steam boiler. Hot boiler water is pumped by a B & G Booster through the shell, thereby heating the water for the heating system, which is pumped through the tubes of the Heater.

Pumping the water through both the heater and the heating system not only affords excellent temperature control, but also permits the use of much smaller pipe and fittings.



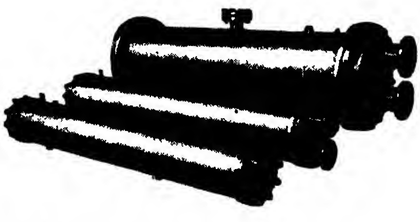
B & G Type "SU" Instantaneous Water Heaters

For heating water with steam. Ideal for industrial plants or wherever large volumes of hot water are required continuously for service water supply or process work. No storage tank required—the large heat transfer surface in these units heats water instantly as needed. Available in a wide range of capacities.



B & G Indirect Water Heaters

Any steam, vapor or hot water heating boiler can be equipped with a B & G Indirect Water Heater. With the proper electrical controls, the Heater will furnish an ample supply of hot water, *winter and summer*, at very low operating cost. Heater must be used with a storage tank of suitable capacity.

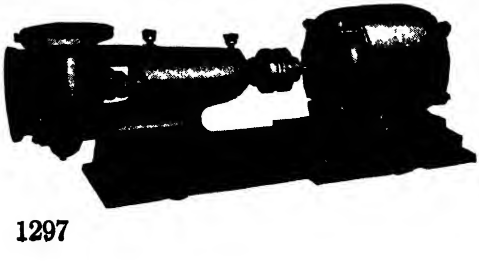


←— B & G Refrigeration Components

A very flexible line of direct expansion evaporators, condensers, liquid receivers, combination liquid receivers and subcoolers for refrigeration purposes are now available. Special alloys may be incorporated in the units for those critical heat transfer applications.

B & G Centrifugal Pumps —→

Design and construction based on years of experience in the industrial field. Rugged, compact units—built to stand up under the strain of continuous operation. Available with semi-open or enclosed impellers—motors flexible coupled or integral with pump. Send for Catalog.

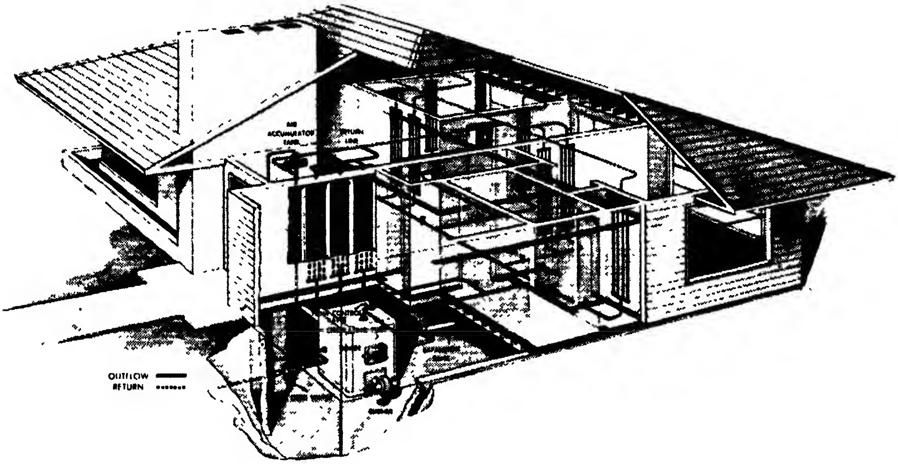


Pan-L Heat Corporation

Manufacturers of PAN-L-GRID for Radiant Heating

2838 N.E. Columbia Blvd., Portland 11, Oregon

Available through Plumbing and Heating Supply Houses throughout the 11 Western States



Pan-L Grids are units manufactured for radiant panel heating in sizes to fit building construction. They are installed within the walls, ceilings or floors during construction. There are no grilles, radiators or registers. Truly, Pan-L Grids provide "magic" heat.

Unlike other panel heating equipment which must be bent while installing, each Pan-L Grid is precisely designed and delivered to the job complete. This enables the installer to determine accurately the flow of water and the size of pipe required to insure a completely satisfactory heating system.

Pan-L Grids can and are being used in conjunction with hot water heating plants using cast iron radiators, convectors, unit heaters and ventilating systems.

Any engineer operating in the 11 western states will be sent, upon request, a copy of Pan-L Grid Design Manual. This valuable book contains all the information required for the design and application of Pan-L Grid installations.

Grid Size	Nominal Length	Overall Length	Overall Width	Overall Thickness	Shipping Weights
4 x 7′	7′-0″	7′-4″	1′-1¼″	0′-1⁵⁄₁₆″	28 lb 10 oz
3 x 7′	7′-0″	7′-4″	0′-9¼″	0′-1⁵⁄₁₆″	20 lb 9 oz
4 x 6′	6′-0″	6′-4″	1′-1¼″	0′-1⁵⁄₁₆″	24 lb 11 oz
4 x 10′ x ½	10′-0″	10′-4″	2′-2″	0′-1⁵⁄₁₆″	43 lb 10 oz
4 x 10′ x ¾	10′-0″	10′-4″	3′-2½″	0′-1⅝″	59 lb 13 oz

Taco Heaters, Incorporated

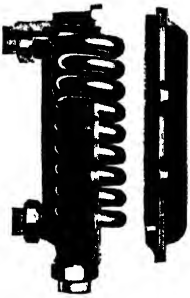
342 Madison Avenue, **New York 17, N. Y.**

TACO HEATERS OF CANADA, LTD., 24 Adelaide St., W., **Toronto**

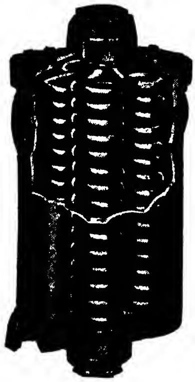
THERE is a storage or tankless, Biltin or external, type Taco indirect water Heater for every job.

Biltin Taco Heaters, which are standard equipment on leading heating boilers, are catalogued only in the boiler catalogs. For additional information on Biltin Tacos write boiler manufacturer or Taco.

Complete catalog information on external Taco Heaters is also available for home, apartment house, hotel and housing projects.

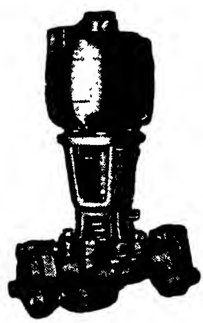


Domestic Tacos



Multi-coil Taco

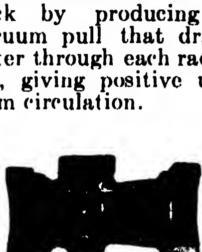
Tankless Taco Nos. 12, 14, 16, 18 and 23



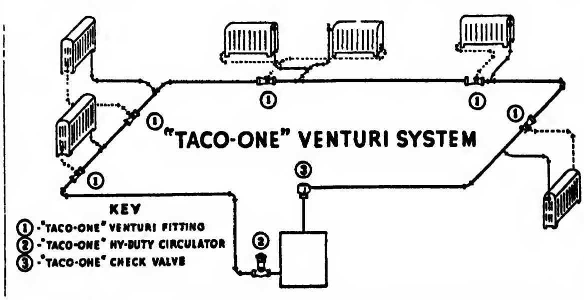
"Taco-One" Venturi System

A forced circulating warm heating system using a single pipe main from boiler to radiators and back again. Small pipes and one pipe main make a neater job, reduce installation costs. Radiators can be placed above and below main. 85 per cent of residential jobs need only one circuit which requires no balancing valves. This revolutionary heating system is made possible by the Taco Venturi Fitting and remarkable Taco Hy-Duty Circulator.

Taco Venturi Fitting—Acts as a suction pump in which the only moving part is water. Water does the trick by producing a vacuum pull that draws water through each radiator, giving positive uniform circulation.



Taco Venturi Fitting (Cross Section)

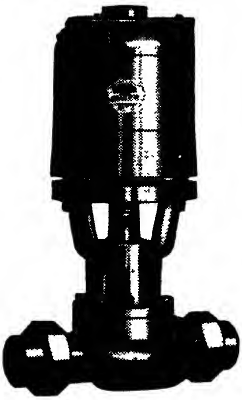


H. A. Thrush & Company

Peru, Indiana

Representatives in Principal Cities

FORCED CIRCULATING THRUSH FLOW CONTROL SYSTEM OF HOT WATER HEATING AND HEATING SPECIALTIES



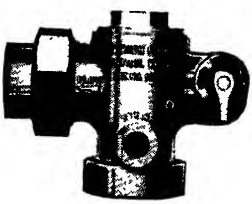
Flexible, economical of fuel, forced circulating Thrush Summer-Winter System of Hot Water Heat is the most satisfactory way to heat buildings. Thrush Systems, Water Circulators, Water Heaters, Pressure Relief and Pressure Reducing Valves and other fuel-saving heating specialties are available. Write today for information or engineering assistance.

THRUSH WATER CIRCULATORS

Five sizes, 1 in., $1\frac{1}{4}$ in., $1\frac{1}{2}$ in., 2 in. and 3 in., for forced circulation in Hot Water Heating and Domestic Water Systems. Save fuel, insure uniform heating.

Made of Iron or Brass

Patent Nos. 2,054,009, 2,111,441, 2,137,791, 2,207,183, 2,207,208, 2,257,867, 2,356,482, 2,358,670



Patent Reissue No. 19,873

THRUSH FLOW CONTROL VALVES

For use with Thrush Water Circulators

This patented valve prevents circulation when not required. Six sizes, 1 in., $1\frac{1}{4}$ in., [illegible] and 3 in. Work automatically by pressure head, generated by Thrush Circulator.

THRUSH PRESSURE REDUCING VALVES

Types for High and Low Pressures

Sizes, $\frac{1}{2}$ in., $\frac{3}{4}$ in. and 1 in. Low pressure reducing valves to reduce pressure of water entering heating system and maintain water supply in system automatically. High pressure reducing valves for protecting house plumbing and heating equipment from excessive city line pressures.

Number 4 Illustrated

THRUSH LOW PRESSURE WATER RELIEF VALVES

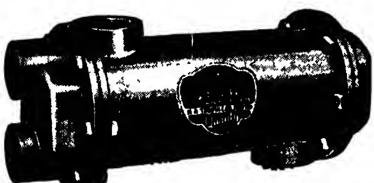
Protect heating boilers from excess pressures. Made in angle or straight types, of iron or brass, sizes $\frac{1}{2}$ in., $\frac{3}{4}$ in. and 1 in. Unfailing dependability has been proved by over a quarter of a century of successful operation.

THRUSH AIR-TIGHT PRESSURE TANKS

An essential part of every hot water heating system. Conserves water and fuel, because the heated water expands into the Thrush Pressure Tank and returns to the system as it cools. Adds to the continued operating efficiency of the heating system over a long period of time.

THRUSH HIGH PRESSURE WATER RELIEF VALVES

Protect water supply or range boilers and gas or electric water heaters from excess pressure and temperature (some types). Made in angle or straight types, of iron or brass, sizes, ½ in., ¾ in. and 1 in., for pressure relief only or combination pressure and temperature relief.



Patent No. 2,180,620

THRUSH WATER HEATERS

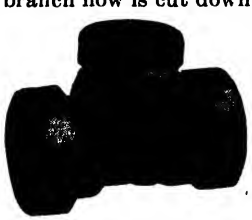
Highly efficient heat exchangers or converters. Sixteen sizes, for Hot Water or Steam. Pressure up to 150 lb water, 75 lb steam. Straight tubes readily cleanable. Provide Domestic Hot Water at low cost. Also used industrially for heating or cooling liquids.

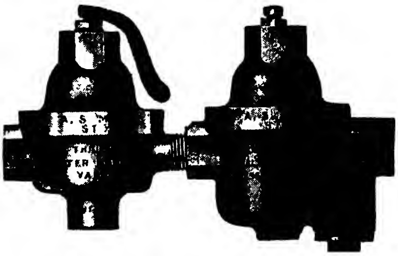
THRUSH SUPPLY TEES FOR ONE PIPE SYSTEMS NON-ADJUSTABLE

Assure positive diversion to radiators. Cast iron sizes, 1 in., 1¼ in., 1½ in. and 2 in., with branch outlets of ½ in., ¾ in. or 1 in. Only one Supply Tee required for each up-feed radiator.

THRUSH ADJUSTABLE SUPPLY TEE FOR ONE PIPE SYSTEMS

Balances flow of water through each radiator, resulting in uniform heating. When branch flow is cut down, main flow is increased not retarded. Easily adjusted. Sizes in cast iron, 1 in., 1¼ in., 1½ in., and 2 in., with branch outlets of ½ in., ¾ in., and 1 in.

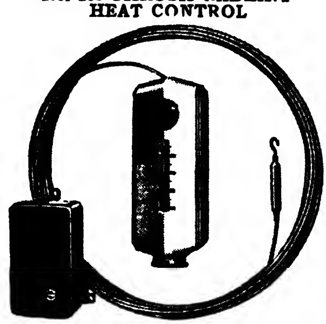




THRUSH DUAL CONTROL UNITS

Provide automatic pressure relief, reduce line pressure, automatically fill and maintain water supply in hot water heating system. Built-in strainer. Made in four types, brass or cast iron, ½ in. or ¾ in.

No. 201 THRUSH RADIANT HEAT CONTROL



No. 200 *No. 201*

Automatically maintains room temperature within a fraction of a degree. Controls both room and water temperature in the radiators, compensating to prevent variation in room temperature or a lack of radiant heat. Requires No. 200 Thrush Relay Transformer for low voltage.

THRUSH BRONZE SUPPLY TEES FOR COPPER ONE-PIPE SYSTEMS

Adjustable and Nonadjustable

Bronze Supply Tees are available in both Adjustable and Nonadjustable types for copper piping. Have solder connections. Made in four sizes, ¾ in., 1 in., 1¼ in., and 1½ in., all with ½ in. branch outlets. Adjustable Bronze Tee illustrated. Nonadjustable Bronze Tee similar to cast iron Tee at left above.

WRITE FOR COMPLETE CATALOG

American Hydrotherm Corp.

215 East 27th Street, New York 16, N. Y.

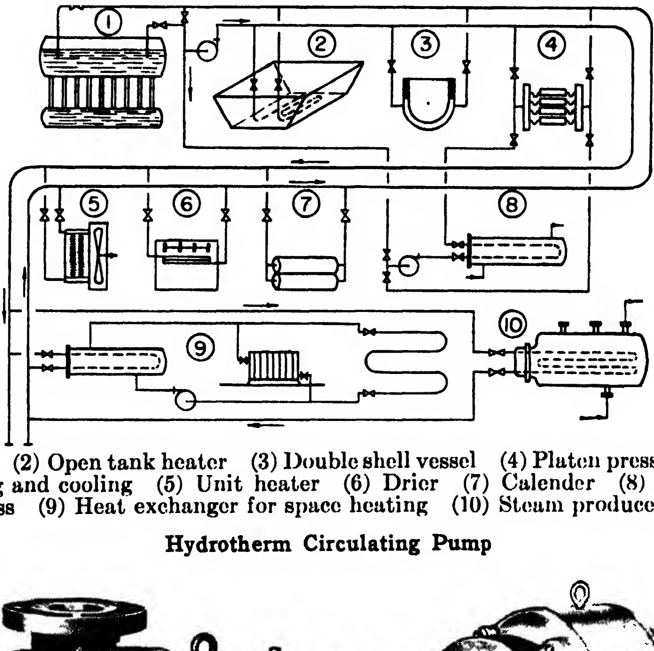
HYDROTHERM

HIGH TEMPERATURE FLUID HEATING UP TO 1200 F

The Hydrotherm high-temperature heating system makes use of liquid under pressure as the heat transfer medium. This liquid may be either water or chemical and is circulated in an absolutely closed system. A boiler or heat exchanger is used to heat or cool the fluid.

Small investment, low repair and maintenance costs and large fuel savings are among the system's many advantages. The large temperature drop between flow and return permit very small piping. Uniform, close tolerance temperature control and long distance heat distribution are also important features. Circulating pump power requirements are generally less than those for the boiler feed pumps in an equivalent steam system. The heat accumulation capacity of the system permits the use of reduced boiler size.

Typical Schematic Flow Chart for Hydrotherm System



(1) Boiler (2) Open tank heater (3) Double shell vessel (4) Platen press connected for heating and cooling (5) Unit heater (6) Drier (7) Calender (8) Cooler for platen press (9) Heat exchanger for space heating (10) Steam producer

Hydrotherm Circulating Pump

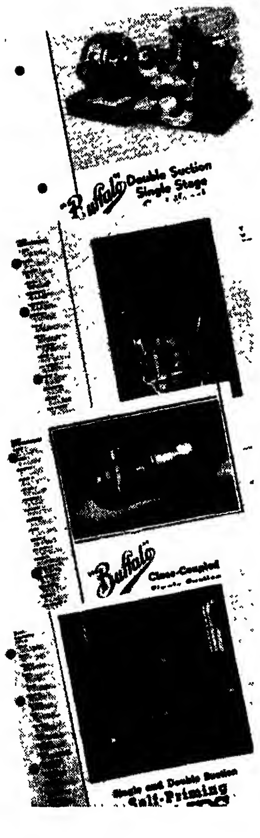
Buffalo Pumps, Inc.

450 Broadway, **Buffalo, N. Y.**

Manufacturers of a Complete Line of Centrifugal Pumps, Single and Double Suction, Single and Multistage, For All Types of Heating and Air Conditioning Installations. Write Us For Engineering Bulletins on Your Problem, or Contact Your Nearest Engineering Representative:

REPRESENTATIVES:

ALBANY 7, N. Y., R. B. Taylor, 966 Broadway; *ATLANTA, GA.*, J. J. O'Shea, 305 Techwood Dr., NW; *BALTIMORE 1, MD.*, C. A. Conklin III, 1014 Cathedral St.; *BOSTON 76, MASS.*, E. D. Johnson, 507 Main Street, Melrose Station; *CHICAGO 6, ILL.*, Emmert & Trumbo, 20 N. Wacker Drive; *CINCINNATI 2, OHIO*, F. W. Twombly, 626 Broadway; *CLEVELAND 13, OHIO*, Weager & Sherman, 518 Rockefeller Bldg.; *DALLAS 1, TEXAS*, Mr. T. H. Anspacher, Mgr., 1801 Tower Petroleum Bldg.; *DAVENPORT, IOWA*, D. C. Murphy Co., 305 Security Bldg ; *DENVER 17, COLO.*, Stearns-Roger Mfg. Co., 1720 Calif. St.; *DES MOINES 14, IOWA*, D. C. Murphy Co., 840 Fifth Avenue; *DETROIT 16, MICH.*, Coon-DeVisser Co., 2051 W. Lafayette Blvd.; *EL PASO, TEXAS*, Stearns Roger Mfg. Co., P. O. Box 38; *GREENVILLE, S. C.*, Roy A. Stipp, 222 N. Main St.; *HOUSTON 2, TEXAS*, D. M. Robinson, 407 Scanlon Bldg.; *INDIANAPOLIS 4, IND.*, S. E. Fenstermaker & Co., 937 Architects & Builders Bldg.; *KANSAS CITY 6, MO.*, W. K. Dyer, 1808 Federal Res. Bank Bldg.; *LOS ANGELES 13, CALIF.*, Frank Halladay, 804 Pershing Sq. Bldg.; *LOUISVILLE 2, KENTUCKY*, H. M. Lutes, 633 South Fifth Street; *MEMPHIS, TENN.*, Humphrey-Wynne Co., 713 Sterick Bldg.; *MINNEAPOLIS 2, MINN.*, E. Floyd Bell, 2102 Foshay Tower; *NEWARK 2, N. J.*, G. C. Norman, 27 Washington St., Room 205; *NEW ORLEANS 12, LA* , Devlin Brothers, 1003 Maritime Bldg.; *NEW YORK 7, N. Y.*, Koithan & Johnson, 29 Cortlandt St.; *OMAHA 2, NEBR* , Wain Engineering Co., 415 Brandeis Theatre Bldg.; *PHILADELPHIA 2, PA.*, Davidson & Hunger, 1200 Cunard Bldg.; *PITTSBURGH 22, PA* , H. L. Moore, 345 Fourth Avenue; *PORTLAND, ORE.*, Arthur Forsyth Co., 921 SW Oak St.; *ROCHESTER 4, N. Y.*, R. Moyer, 846 Sibley Tower Bldg.; *ST. LOUIS 3, MO.*, J. W. Cooper, 2118 Pine St.; *SALT LAKE CITY 1, UTAH*, Stearns Roger Mfg. Co,. 405 Kearns Bldg.; *SAN ANTONIO 6, TEXAS*, Langhammer Rummel Co., 436 Main St.; *SAN FRANCISCO, CALIF.*, Chas. W. Lockhart, 1214 Central Tower Bldg.; *SEATTLE, WASH.*, Arthur Forsyth, 3150 Ellicott Avenue; *TOLEDO 2, OHIO*, Carl M Eyster, 1118 Madison Avenue; *WASHINGTON 5, D. C.*, G. S. Frankel, Mgr., 310 Woodward Bldg.



SINGLE STAGE DOUBLE SUCTION PUMPS. For clear water service from 10 to 10,000 U. S. gpm. These pumps provide substantial savings in power and maintenance over years of service, because they incorporate all accepted features of modern centrifugal pump design. BULLETIN 955-M.

AUTOMATIC SUMP PUMPS. Compact units, shipped complete, with very little installation work necessary. Ball bearing thrust. Unusual efficiencies. BULLETIN 936-F.

CLOSE-COUPLED SINGLE SUCTION PUMPS. Compact and easily serviced. Suited to handling hot water with low submergence. Close-coupled design means permanent shaft alignment. BULLETIN 975-B

SELF-PRIMING SINGLE AND DOUBLE SUCTION PUMPS. Maintain positive prime without foot valves. All parts are oversized for longer service. The high-efficiency rotors are vibrationless. BULLETIN 970-A

Chicago Pump Company

2330 Wolfram Street **BRunswick 8-4110** **Chicago 18**

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

"CONDO-VAC"

Return Line Vacuum and Boiler Feed Pump for Heating Systems

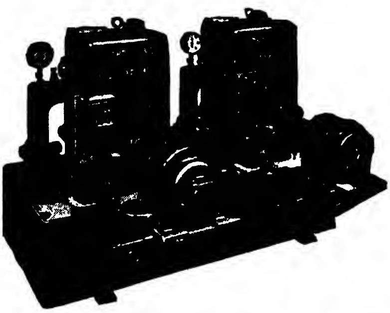


Fig. 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for bulletin 270.*

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

Fig. 2130—Close-Coupled, side suction pump

Capacities range from 3 to 600 Gpm against heads up to 189 ft. Motors from ⅛ to 20 Hp. Discharge 1 to 3 in. Closed and open type impellers. *Bulletin 108.*

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 75,000 Sq Ft Radiation

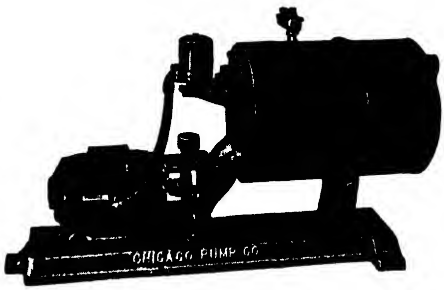


Fig. 1946

"Sure-Return" Condensation Pumps and Receivers are built for systems up to 75,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

Vertical Condensation Pumps

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

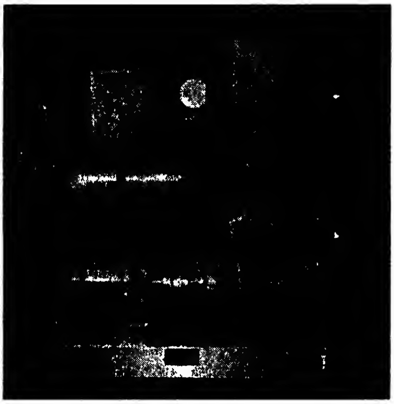
Fig. 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245, 253 and 255.*

The Nash Engineering Company

234 Wilson Road South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities



Return Line Vacuum Heating Pump

Standard with the heating industry for over eighteen years. Removes air and condensation from return lines of vacuum steam heating systems, discharging air to atmosphere and returning water to the boiler.

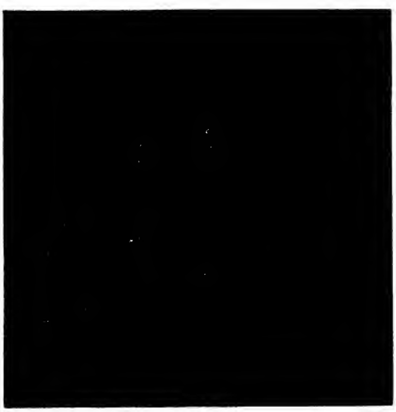
Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. Pump is bronze fitted throughout.

Supplied direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation. Standard in capacities up to 300,000 sq ft E.D.R. Larger units special. Bulletins Nos. 307, 308, 309, and 310 on request.

Vapor Turbine Vacuum Heating Pump

Jennings Vapor Turbine Heating Pumps combine all advantages of the standard return line heating pump with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E.D.R. Larger units special. Bulletin No. 290 on request.



Condensation Pump and Receiver

Removes the condensation from radiators in return line steam heating systems, particularly radiators set below the boiler water line level, and pumps the condensation back to the boiler. Pump is bronze fitted with enclosed centrifugal impeller of improved design. By making the pump casing a part of the return tank, and bolting the motor base to the tank, floor space is conserved. The rectangular construction permits installation in a corner against the wall.

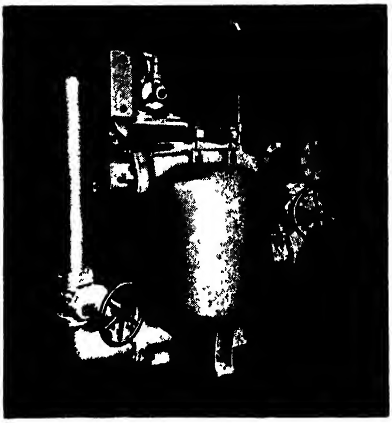
These pumps are furnished in standard sizes with capacities ranging from 1½ to 225 gpm of water. For serving up to 150,000 sq ft of equivalent direct radiation. Bulletin No. 319 on request.

The Nash Engineering Company

234 Wilson Road . South Norwalk, Conn., U. S. A.

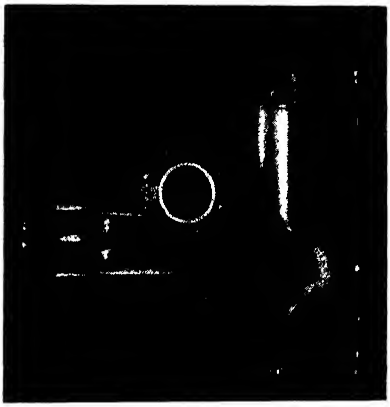
Sales and Service Offices in all Principal Cities

SEWAGE EJECTOR



For pumping unscreened sewage or drainage from basements below the street sewer level, handling crude sewage from low level districts, pumping effluent, sludge and other heavy liquids. The Jennings Sewage Ejector is of the pneumatic type. Air, compressed only to the pressure at which it is used, by a Nash Hytor Air Compressor, is motive power to pump the accumulated sewage from a pot to the sewer. There are no air storage tanks, reciprocating air compressors or screens, no air valves. Furnished in several standard sizes up to 1500 gpm against heads up to 100 ft. Bulletins on request.

Suction Sump and Sewage Pumps



Jennings Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. Sewage Pumps are equipped with non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are mounted on same shaft that carries rotor of the driving motor, forming a single moving element, rotating without metallic contact.

Will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit, affording perfect accessibility for inspection or cleaning. Capacities to meet all requirements. Bulletins Nos. 159, 161, and 338 on request.

Air Compressor and Vacuum Pump

Nash Air Compressors operate on a unique and different principle. The one moving part rotates in casing without metallic contact. There is nothing to wear, and no internal lubrication.

Nash Compressors deliver absolutely clean air; ideal for agitation of liquids, pressure displacement, and handling gases. Vacuum pumps ideal for priming pumps, blood sucking pumps in hospitals, and wherever non-pulsating vacuum is required.

Pressure 75 lb or vacuum 27 in. of mercury. Furnished for any capacity; special for higher vacuums and pressures. Bulletins Nos. 282, 325, 331 and 337 on request.

Ingersoll-Rand

11 BROADWAY, NEW YORK 4, N. Y.

BIRMINGHAM	DALLAS	KNOXVILLE	PHILADELPHIA	SCRANTON
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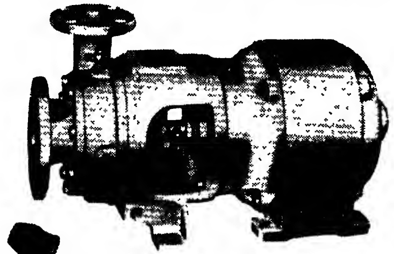
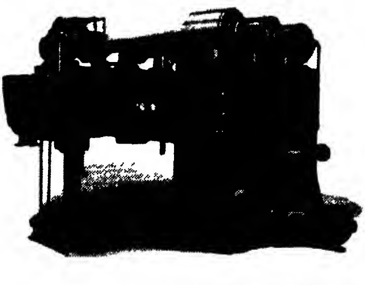
BUTTE	DULUTH	NEW ORLEANS	SALT LAKE CITY	TULSA
CHICAGO	EL PASO	NEW YORK	SAN FRANCISCO	WASHINGTON
CLEVELAND				

Offices and agents throughout the world





CENTRIFUGAL PUMPS

The MOTORPUMP is a compact, "package" unit, mounted integrally with motor on a rigid, oversize shaft and over-size bearings. It is highly adaptable to many services, needs no special foundations, and operates equally well in any position. It is available in several materials for pumping various liquids. Capacities from 10 to 1800 gpm, heads to 600 ft.

Other I-R pumps are offered for all hydraulic services, with any type of drive.

ALL PURPOSE PORTABLE TOOLS

Electric Ipactool are light-weight, portable, tools for running and removing nuts, screws and studs, drilling, reaming, tapping, wire brushing, drilling brick and masonry, driving wood augers, hole-saw work, and the 101 jobs encountered in installation work. Plug into any wall outlet. Capacities: ⅜ in. drills, tapping to ¾ in., running nuts to ⅝ in. thread size. 110 or 220 volts, universal motor.

LIGHT-WEIGHT JACKHAMER

The J-10 Jackhamer is the smallest of the self-rotating, rock-drill line, weighs 14 lbs, and is especially designed for maintenance and installation work. Its uses include drilling masonry for conduit, sprinkler hangers, foundation bolt holes, pipe lines and drains as well as tearing out brick-work for doors and windows and similar demolition jobs. The J-10 is air-powered.

STEAM-JET REFRIGERATION

Where refrigeration is needed down to 35°F, and a supply of steam is available, the I-R system of water-vapor refrigeration with Steam-Jet Coolers is offered. In this system water is the only refrigeration medium. It is cooled by direct evaporation in a high vacuum created by steam-jet booster ejectors. There are no moving parts, no vibration nor noise. Sizes run from 20 to 1200 tons of refrigeration and can be built to operate at any one of a wide range of pressures down to 2 lb per sq in.

I-R COMPRESSORS OF ALL TYPES AND SIZES

I-R Compressors are offered in all sizes and types from ½ to 3000 hp, in pressures from a few ounces to 15000 lbs, and in stationary or portable models. Air-cooled units range from ½ to 90 hp. I-R Water systems are available for industrial and domestic uses.

INTERNATIONAL HEATING & VENTILATING EXPOSITION
THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, **New York 17, N. Y.**

EXPOSITIONS HELD

Chicago, 1949; New York, 1948; Cleveland, 1947-1940; New York, 1938; Chicago, 1936; New York, 1934; Cleveland, 1932; Philadelphia, 1930.

FUTURE SCHEDULE

Southwest Air Conditioning Exposition, Dallas, Texas, Jan. 23-27, 1950.

UNDER AUSPICES OF A.S.H.V.E.

These Expositions have been and will be held co-incident with the annual meetings of the American Society of Heating and Ventilating Engineers and under their auspices. Management is by International Exposition Company with permanent headquarters at Grand Central Palace, New York 17, N. Y.

EXHIBITORS

Comprise leading firms in each phase of the industry; number has varied from 150 to 400 exhibitors.

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of The A.S.H.V.E. Guide.

1. *The* Combustion *Group:*
   Furnaces, burners (coal, oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries.
2. *The* Oil Burner *Group:*
3. *The* Hydraulic *Group:*
   Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc.
4. *The* Steam Heating *Group:*
   Vapor heating, steam specialties.
5. *The* Hot Water Heating *Group:*
6. *The* Air *Group:*
   Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc.
7. *The* Air Conditioning *Group:*
   Equipment which circulates and filters the air, in summer dehumidifies and cools; in winter heats and humidifies, and does all these in proper season for complete, all year-round air conditioning.
8. *The* Control *Group:*
   Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured.
9. *The* Refrigerating *Group:*
   Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants.
10. *The* Central Heating *Group:*
    Apparatus and materials especially designed or adapted to the uses of central heating and central heating station supplies.
11. *The* Insulating *Group:*
    Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, etc.
12. *The* Miscellaneous *Group:*
    Electric Heaters, boiler and pipe repair alloys, liquids and compounds, tools of all kinds, and equipment not specifically included in the above groups, but related thereto.
13. *The* Machinery and General Equipment *Group.*
14. Books and Publications.

VISITOR ATTENDANCE

Attendance is by invitation and registration only, thereby presenting a selected audience. Included are contractors, dealers, jobbers, supply houses, home owners, industrial users, professional and service organizations, public utilities, real estate management concerns, etc. A detailed analysis of registered attendance is available on request.

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value. This Exposition stands among the leaders in Industrial Expositions in America. It is an educational institution which brings together the research developments and improvements in equipment and materials for use in heating, ventilating and air conditioning all types of buildings.

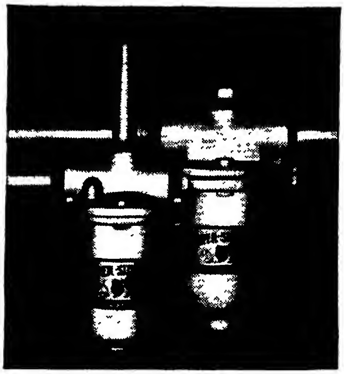
The V. D. Anderson Company

1942 West 96th Street, **Cleveland, Ohio**

Representatives in all principal cities

SUPER-SILVERTOP STEAM TRAPS

Are inverted bucket steam traps of an improved, thoroughly tested design. These steam traps automatically remove both condensate and air from the steam system—no manual operations are necessary.



Shows how trap is installed either as an elbow or straight in-line, using a minimum of fittings.

APPLICATIONS

Super-Silvertops are used on any steam-using equipment where it is desired to remove condensate and air automatically in order to produce maximum heating efficiency.

FEATURES

Simplified Piping—Connected either as an elbow or straight-in-line—only one nipple needed since the U-tube is inside the trap. Saves as much as three elbows, three nipples and 60 minutes of time, compared to other inverted bucket steam traps.

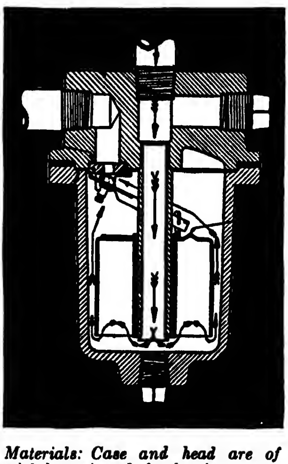
Precision Parts Alignment—The bucket does not swing free. It is controlled in true engineering fashion, guided on a hexagonal tube. This makes a knife-edge contact, eliminating almost all friction. The unusual guide arrangement keeps all parts in proper alignment and prevents the bucket from hitting the sidewalls of the case. Positive closing of the valve is insured.

Self-Cleaning—The reversing of the condensate flow on entering the trap produces a scrubbing action. This stirs up any sediment and dirt which is then carried away in the discharging condensate.

No Restricting Passages—Even in the smallest sizes there are no narrow cored passages to become clogged with scale.

Cannot Air-bind—As the air is automatically discharged ahead of the condensate in each cycle of operation.

Vacuum or Pressure—These traps do not leak steam. No danger of vacuum being destroyed—trap is recommended for vacuum operation.



Materials: Case and head are of nickel semi-steel, bucket is copper, guide tube of admiralty bronze, valve and seat of special chrome alloy, all other parts of stainless steel. All interior parts can be furnished in stainless steel when so ordered.

INSTANT HEAT

Anderson Super-Silvertops can be equipped with Thermal Air Eliminators where instant heat is desired. **Other Anderson products are:** combination open float and thermostatic steam traps, float-type steam traps, air release valves, and pipeline strainers.

SIZES, LIST PRICES AND CAPACITIES

No. of Trap	110	118	119	120
Code Word for Telegrams	Vay.	Ves.	Vys.	Vyt.
Pipe Connections, In. (See Note)	½" only	½" only	½ or ¾	½ or ¾
Shipping Weight, Lbs.	2¾	5	6	7
Diameter, In.	4	3⅞	3⅞	3⅞
Height, In.	4 1/16	3⅞	5¾	6 1/16
Maximum Gauge Pressure	100	150	200	200
List Price	$10.00	$9.00	$10.00	$12.00

WITH THERMAL AIR ELIMINATOR

Code Word for Telegrams	Vayel	Vesel	Vysel	Vytel
List Price	$11.60	$10.60	$11.60	$13.60
Differential Pressure	CAPACITY IN POUNDS OF WATER PER HOUR			
1	137	161	225	380
2	192	230	323	545
3	238	280	392	660
4	270	322	450	758
5	305	360	510	860
10	435	505	705	1190
15	525	595	832	1400
20	300	700	980	1650
30	370	515	710	1110
40	410	585	810	1260
50	455	650	900	1400
60	490	550	757	1160
70	375	586	810	1240
80	400	625	860	1320
90	420	435	645	1140
100	440	455	675	1190
125	.	500	740	1310
150	. .	450	580	1000
200	. .	...	655	1100
225	. .	...	...	....
250		.	.	....

Capacities based on continuous flow. When ordering be sure to specify maximum steam pressure.
Note: Pipe shown in heavy type are standard and traps will be shipped tapped standard unless otherwise specified. Pipe sizes shown in light type furnished at no additional cost but only from Cleveland stock.

SUPER-SILVERTOP STEEL SERIES

. . . for higher pressures and temperatures—come complete. No need for extra, expensive fittings, no pipe to bend, and no labor cost for making up extra pipe joints. Installation may be straight-in-line or as an elbow. Only two threaded joints required in either installation. The case and head are made of steel; and the gasket joints are of the recess type, preventing any possibility of gasket blowouts; all interior parts are stainless steel, except the valve and seat which are special chrome alloy.

CAPACITIES AND LIST PRICES

	No. of Trap		
Size	33	34	36
List Prices	$77.00	$120.00	$230.00
Code Word	FSH	FSB	FSC
Pressure, Lbs.			
150	2800	6000	11000
200	3100	5500	9000
250	3500	6000	10000
300	2500	6400	11000
350	2800	4500	8300
400	2500	4500	8700
500	2800	4750	9000
600	3000	4900	8500
700	2000	4500	8500

Armstrong Machine Works

851 Maple Street, Three Rivers, Mich.

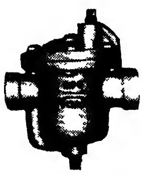
Representatives in Principal Centers

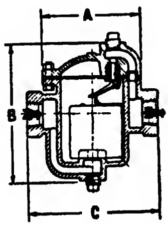
Steam Traps . . . Air Traps . . . Humidifiers
High Side Floats . . . Refrigerant Purgers

ARMSTRONG INVERTED BUCKET STEAM TRAPS

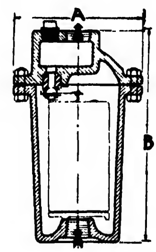
Armstrong offers a complete line of traps for draining low and medium pressure steam headers, pipe coil radiation, unit heaters, air conditioning equipment, process equipment, etc., to gravity or vacuum return systems. Armstrong design and construction results in fast heating, uniformly high temperatures and long trap life with low maintenance.

Automatic Air Discharge. Standard traps automatically discharge normal amounts of air along with condensate. Where large amounts of air must be vented quickly when steam is first turned on, Armstrong BLAST Traps with thermic bucket vents are recommended. Depending on size, air handling capacity ranges from 500 to 1500 cu ft free air per hour.





No. 800–813



No. 211–216

CAPACITIES, PRICES DIMENSIONS

Note that capacities of No. 811, 812 and 813 traps are the same as No. 211, 212 and 213 respectively. On most installations, the engineer therefore has a choice of side inlet—side outlet or bottom inlet—top outlet body styles.

Side Inlet Traps	Lb. Pressure	No. 800	No. 811	No. 812	No. 813			
Pipe Connections		½″ or ¾″	½″ or ¾″	½″ or ¾″	¾″ or 1″			
List Price (Regular)		$7.00	$10.00	$16.00	$22.00			
List Price (Blast Trap)		$8.50	$11.50	$18.00	$24.00			
Telegraph Code (Regular)		Aloe	Brown	Cherry	Dawn			
Telegraph Code (Blast Trap)		Aloette	Brownette	Cherette	Dawnette			
Dimension A		3¾″	3¾″	5⅝″	7″			
" B		5⅜″	6⅞″	8{11/16}″	11¼″			
" C		5″	5″	6½″	7¾″			
Number of Bolts		6	6	6	6			
Diameter of Bolts		¼″	¼″	⅜″	½″			
Weight		4½ lbs.	5½ lbs.	13½ lbs.	25 lbs.			
Maximum Pressure, lbs.		125	250	250	250			
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information see the Capacity Chart in Armstrong Steam Trap Book.	5	450	830	1600	2900	4800	7600	14500
	10	560	950	1900	3500	5800	9000	17300
	15	640	1060	2100	3900	6500	10000	19200
	20	690	880	1800	3500	6000	8500	18500
	30	500	1000	2050	4000	6800	9800	18000
	50	580	840	1900	4100	6300	9000	18200
	70	660	950	2200	3800	6000	9200	18300
	100	640	860	1800	3600	6200	10400	18000
	125	680	950	2000	3900	6700	10900	20000
	150		810	1500	3500	5700	9500	18500
	200*	...	860	1600	3200	5300	9200	17500
	250		760	1300	3500	5700	7000	19000
Bottom Inlet Traps			**No. 211**	**No. 212**	**No. 213**	**No. 214**	**No. 215**	**No. 216**
Pipe Connections		....	½″	½″ or ¾″	½″ or ¾″	1″	1″ or 1¼″	1½″ or 2″
List Price (Regular)		....	$ 9.25	$15.00	$20.75	$29.00	$38.00	$55.00
List Price (Blast Trap)		....	$10.75	$17.00	$22.75	$31.50	$40.50	$60.00
Telegraph Code (Regular)		....	Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)		....	Aspette	Birette	Walette	Hemlette	Larette	Tamrotte
Height........Dimension B		....	6⅜″	8″	10¼″	12½″	14″	16¾
Diameter ..... " A		....	4⅛″	5″	6⅜″	7½″	8½″	10⅜″
Diameter of Bolts		....	¼″	¼″	⅜″	⅜″	½″	½″
Number of Bolts		....	6	8	6	8	8	12
Weight		....	5½ lb	10½ lb	19 lb	32 lb	47 lb	80 lb
Maximum Pressure, lb		....	250	250	250	250	250	250

High Capacity, Compact Size. The high leverage of the patented free-floating lever makes it possible to open discharge orifices which are very large for over-all trap size.

Positive Action. The discharge valve is either wide open or closed tight. Fast opening and closing prevent wire-drawing.

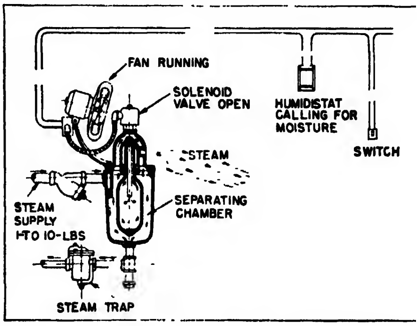
Self-Cleaning. Swirling action of condensate during discharge carries dirt through trap. There are no dead spots for dirt to collect.

High Quality. 18-8 stainless interior parts. Valve and seat are chrome steel, hardened, ground and lapped. Low pressure trap parts same material and quality as those used for 1500 psi, 900° F.

Armstrong Steam Trap Book. 36 pages of data on traps, selection, installation and maintenance. A usable handbook for any engineer dealing with traps. Free copy on request.

ARMSTRONG STEAM HUMIDIFIERS

For Stores, Offices, Hospitals, Factories, Laboratories

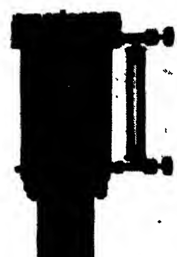


Operation. These units provide automatic, closely controlled humidification by introduction of steam directly into the atmosphere. Installation is comparable to that of unit heaters. Steam at 10 psi or less is required. A solenoid discharge valve on the humidifier is controlled by a sensitive humidistat. A fan mounted on the humidifier aids in steam dispersal, or a venturi nozzle can be supplied. Where an electric spark might represent an explosion hazard, compressed air operated models are available. In addition to the unit humidifier illustrated, there is a large model for installation in large air ducts and central heating systems. Capacities range from 31 to 630 lbs of steam per hour.

Advantages. Installation cost of Armstrong Humidifiers is as much as 80 per cent less than some types of equipment. The small C-2 unit lists at $120.00 complete with fan and motor, humidistat, strainer and steam trap. Operation, using steam at around $1.00 per ton, is economical. There is no extra load on the heating system. No dripping—any moisture in the steam is re-evaporated in the steam-jacketed separating chamber. Control is accurate within a few per cent R. H. There is almost no noise with open nozzle types.

Bulletin No. 1771 gives complete data on required relative humidities, selection and installation of Armstrong Humidifiers. Available on request.

ARMSTRONG NON-CONDENSABLE GAS PURGERS



Armstrong Purgers *automatically* remove air and other noncondensable gases from refrigerating systems, with minimum loss of refrigerant. Refrigerant gas is condensed in the cap of the purger, causing non-condensables to separate out at the top from where they discharge through a needle valve. These purgers are suitable for systems from 12 to 500 tons capacity. Body is forged steel, interior parts stainless steel. Purger for Ammonia, Freon, etc., lists at $200. Heavier model for CO_2 lists at $350. BULLETIN NO. 192 describes benefits of purging, gives complete data on installation and operation of Armstrong Purgers. Available on request.

HIGH SIDE FLOATS. For compression type refrigerating systems where all of the liquid refrigerant can be carried on evaporating side. Identical to Armstrong Steam Traps except for bucket weight. Ask for BULLETIN 178.

All Armstrong products are sold on a basis of satisfaction or your money back.

Barnes & Jones

INCORPORATED

New York Office: 101 Park Avenue

129 Brookside Avenue Boston 30, Mass.

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves; Adjustable-Orifice Radiator Valves; Packless Quick-Opening Radiator Valves; Thermostatic Radiator Traps; Thermostatic Trap Replacement Units; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers; Damper Regulators; Gages; Systems of Zone Control for Steam Heating. Complete Catalog on Request

Series K

The Series K valve is of the modulating type equipped with dial and pointer to indicate whether the valve is open or shut or in an intermediate position.

Lever or wheel handle.

Series F

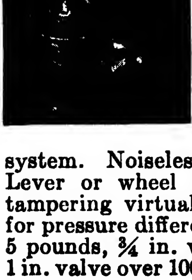
The Series F valve has no dial or pointer and is therefore furnished only with wheel handle or lock shield. Of same quality as Series K but lower priced.

Both Series K and Series F valves are quick opening, nonrising stems, renewable disc seats, of packless design. Made angle pattern in ½, ¾, 1 and 1¼ in. sizes; straightway (globe) pattern in ¾ and 1 in. sizes. For use in vapor or vacuum systems.

Capacities—Sq Ft E. D. R.

Size valve	½"	¾"	1"	1¼"
2 ounces pres.	25	50	90	160
8 ounces pres.	45	90	160	280

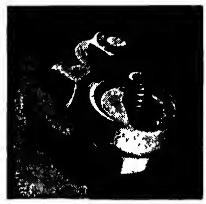
Type H



The Type H valve is of the adjustable orifice type so arranged that the dial indicates at all times the size of the radiator for which the valve is adjusted. May be adjusted with steam on the system. Noiseless in operation.

Lever or wheel handle, unauthorized tampering virtually impossible. Made for pressure differentials of 1, 2, 3, 4 and 5 pounds, ¾ in. valve up to 100 sq ft. 1 in. valve over 100 sq ft.

Thermostatic Radiator Traps



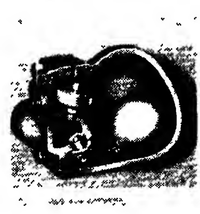
The B & J precision made thermostatic trap contains the unique, patented cage unit which is a complete operating unit in itself. The cage assembly contains the double thermostatic element calibrated in the factory, and locked in correct adjustment with the trap seat, which is also an integral part of the cage unit. Suitable for pressures up to 15 lb.

Capacities—Sq Ft of E. D. R.

Size	No.	Pres. Diff.—Lbs. per Sq In. ⅛	¼	1	2	5	15
½	122	70	92	168	228	320	610
¾	134	120	152	320	450	710	1260
1	147	200	300	590	760	1200	2200

Medium and High Pressure traps suitable for pressures between 15 lb and 100 lb can be furnished.

Float and Thermostatic Traps



Made in five sizes for handling large and sudden loads of condensate that are greater and more variable than can be handled efficiently by thermostatic traps. Suitable for pressures up to 15 lb.

Capacities Lbs Water per Hour

Size	No.	Pres. Diff.—Lbs. per Sq In. ¼	½	2	5	15
¾	T41	70	100	200	210	230
1	T42	175	250	500	525	575
1¼	T43	425	600	1200	1260	1380
1½	T44	850	1200	2400	2520	2760
2	T45	1775	2500	5000	5250	5750

Medium pressure F. & T. traps in ¾ in. and 1 in. sizes can be furnished for pressures up to 65 lb.

Barnes & Jones, Inc., manufactures a complete line of control systems operated either manually from a central station or automatically from an outdoor thermostat.

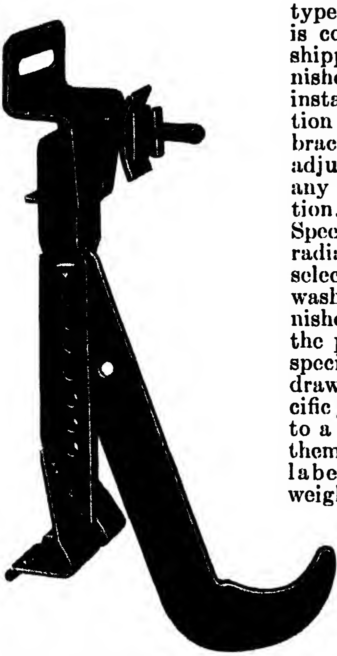
Carty & Moore Engineering Co.

Established 1919

511 West Larned Street • • **Detroit 26, Mich.**

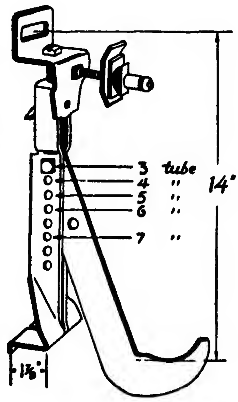
STEEL RADIATOR BRACKETS **STEEL CONCRETE INSERTS**

For over a quarter of a century Carty-Moore Speed Brackets have been recognized by engineers and contractors as a superior product and there are nearly a million in use today. Specify C & M Brackets on your next job and note the substantial savings in labor due to the quick-mounting features.



MODEL NO. 22 BOTTOM HUNG SPEED RADIATOR BRACKETS

Illustrated is the No. 22 Speed Bracket for hanging all types of wall, tube or thin-tube radiation. This bracket is completely assembled when shipped and all parts are furnished, ready for quick and easy installation. A brief instruction card is enclosed with each bracket explaining the simple adjustments required to hang any particular type of radiation. Preparation of a C & M Speed Bracket for hanging a radiator consists simply of selecting the right hold-back washer (there are three furnished) and bolting the hook in the proper hole. Roughing-in specifications are shown by the drawing on the right. For specific jobs 40 brackets are packed to a burlap bag—jobbers stock them six in an attractively-labeled carton. Shipping weight approx. 3½ lbs.



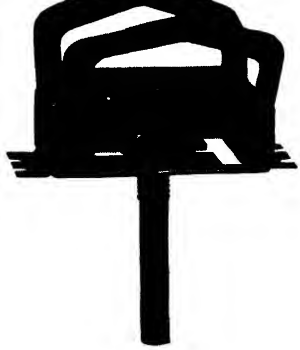
This bracket is also available with double hook and reinforced frame for double row wall radiation—specify No. 222.

MODEL NO. 33 TOP HUNG BRACKETS

A completely assembled top hung bracket with 1¾ in. outset from the wall, especially approved for government jobs. Also available with 2-⅞ in. outset for hospitals, institutions, etc. where frequent cleaning behind the radiator is required—specify No. 333. When ordering top hung brackets, specify model number and the type of radiation to be hung.

MODEL NO. 44 CONCRETE INSERTS

Carty-Moore No. 44 Concrete Inserts have long been used by leading heating contractors and are designed to meet the most exacting requirements. The C & M Insert is made of heavy gage pressed steel stampings for ⅜ in., ½ in., ⅝ in. and ¾ in. nuts. A long travel slot permits ample adjustment yet the nut cannot pull out and the wide wingspread allows the insert to become deeply imbedded in the concrete so it cannot tear out under heavy strain. 50-⅜ in. or ½ in. packed in a nicely-labeled carton—25-⅝ in. or ¾ in.

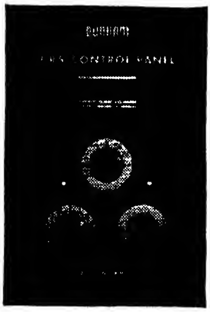


C. A. Dunham Company

Administrative and General Offices

400 W. Madison Street, **Chicago 6, Ill.**

Factories: Marshalltown, Iowa; Michigan City, Ind.; Toronto, Canada; London, England
TORONTO 4, 1523 Davenport Road. LONDON, Lombard Rd., Merton, S.W. 19



"Dunham Heating Service"

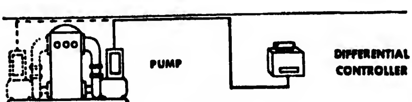
is as close to you as your telephone in most principal cities; see classified telephone directory. Expert heating engineers are available to architects and engineers for counsel at every one of these offices.

CONTROL VALVE

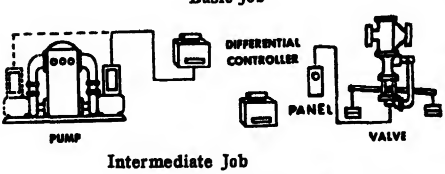
THE DUNHAM VARI-VAC DIFFERENTIAL "JOB SCALED" SYSTEMS

This System provides a complete Differential System for every size job and budget. The Vari-Vac System is a vacuum return job plus exclusive features which permit utilization of varying vacuums to provide the right amount of heat under any weather condition. It exactly balances heat supply with heat demand. It provides a continuous flow of steam—continuous heat distribution—at controlled temperatures. In coldest weather steam as hot as 212 F or higher can be obtained. In mild weather steam as cool as 133 F can be circulated. The simplified Dunham "Metro System of Piping" greatly reduces installation and operation costs, from the simplest basic installation to the complete'y automatic controlled job.

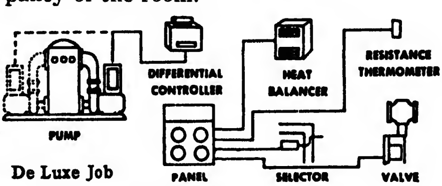
The basic control of the Differential Temperature Control Equipment balances the heat supply (measured by the Heat Balancer) with the heat demand (measured by the Selector). Compensating or limiting features are provided by the room Resistance Thermometer units. Since this room thermometer is not depended upon to perform the function of the conventional room thermostat, it is not as limited in ts application. It can be used to register room temperatures from any location regardless of the type of occupancy of the room.



Basic Job



Intermediate Job



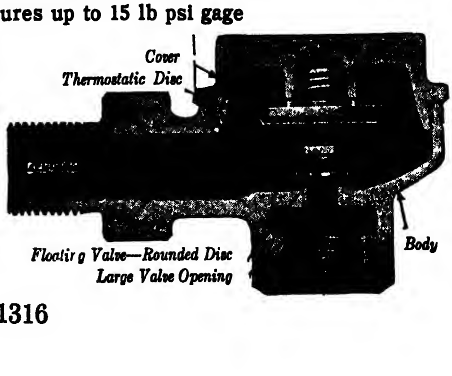
De Luxe Job

DUNHAM THERMOSTATIC TRAPS

For Operating Pressures up to 15 lb psi gage

Cast bronze body with valve seat and cast bronze cover containing the expansion thermostat. Disc constructed of monel metal sheet, valve of cuprous alloys. Thermostatic elements for traps of same size interchangeable without adjustment.

No. 1C-½ in., AP., SW., RH., LH., VS., 200 sq ft EDR. No. 2E, ¾ in. AP., SW., 400 sq ft. No. 3C, 1 in. AP 700 sq ft.



DUNHAM THERMOSTATIC STEAM TRAPS—TYPE TH

For Working Pressures 5 Lb to 100 Lb

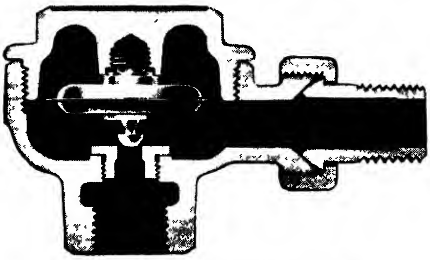


Fig. 3518A—Sectional Type THI-A

These traps are of the thermostatic fluid expansion type, operating in response to the pressure caused by partial vaporization of the liquid within the thermostatic disc. The trap consists of two principal parts, a body with renewable valve seat and a cover containing the thermostatic element. The trap is non-adjustable; permanent adjustment for correct operation is built into the thermostatic element.

The bodies and covers as well as the union nut and nipples are brass. The thermostatic element is fabricated from monel metal sheet using a special welding process, and corrugations are designed to reduce the movement at the rim, distributing the disc motion uniformly.

The valve and seat are of special heat treated stainless steel. The spherical valve is swiveled to insure its seating tightly without causing localized stresses on the thermostatic element.

The valve opening is exceptionally large and the passage of water or dirt is not obstructed by a guide.

CAPACITY POUNDS CONDENSATE PER HOUR

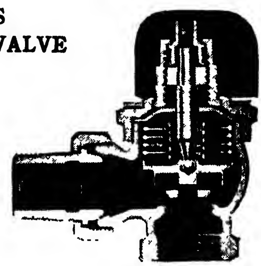
Trap Size	Working Pressure (Lbs. per sq. in. gage)				
	5	25	35	50	100
TH1A	125	355	445	560	840
TH2A	195	565	700	880	1300
TH3A	290	790	980	1200	1780

Note—TH2A and TH3A have a double thermostatic disc.

Traps shall start discharging with a temperature differential (i.e. difference between temperature of discharging water and that of saturated steam of same pressure as that on the trap of about 30 F to 40 F.

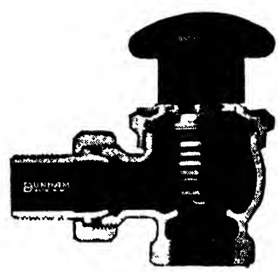
DUNHAM RADIATOR VALVES

"ORIFLEX" PACKLESS RADIATOR VALVE



A self-contained adjustable orifice valve for proportioning the steam supply to each radiator. The adjustable orifice within the valve controls the steam flow. No need to disconnect valve when making an adjustment. Merely remove medallion, insert the key on the adjustment stem, adjust orifice (calibrated guide surrounds stem) to the exact setting needed for desired balance. Sizes ½ in., ¾ in., 1 in., 1¼ in. Capacity 30, 120, 160, 240 sq ft EDR.

PACKLESS RADIATOR VALVE, SERIES 1140



Suitable for all types of low pressure steam heating systems. The bellows construction, the non-rising stem, low bonnet, heat-resistant, composition handle requiring less than one turn to open the valve fully, recommend this valve for long-wearing service and quality. Sizes ½ in. to 1¼ in.

Bodies and bonnets are brass castings, rough finish, furnished with heavily constructed brass union nuts and nipples. Valves are made packless by means of a built-up type expansion member consisting of a series of corrugated phosphor bronze diaphragms. The expansion member prevents leakage of steam, air and water, but also prevents steam, water and dirt from clogging and corroding the spindle nut and screw.

700 SERIES SPRING PACKED RADIATOR VALVE

Bodies are brass castings, rough finish. Heavily constructed brass union nuts and nipples. All pipe threads and tappings are standard gages. Non-rising stem. Requires less than one turn of handle to open the valve fully. Angle, S.W., R.H., or L.H., corner patterns in sizes from ½ in. to 2 in. inclusive.

DUNHAM INVERTED BUCKET TRAPS

Type OB For Operating Pressures up to 250 lb

Type OBS For Operating Pressures up to 150 lb

This high pressure trap vents air and drains water from steam lines, heat exchangers, sterilizers and processing equipment operating at their respective pressure ranges. It is particularly adapted to clearing high pressure distribution lines of condensate and to draining high pressure applications such as coffee urns, cooking utensils, laundry equipment, etc. It increases the efficiency of heat transmission and holds equipment capacity to a maximum. Type OBS built only in ½ in. and ¾ in. sizes, right hand female tappings. Type OB sizes are ½ in. to 1¼ in. inclusive. Body and cover are of high test semi-steel castings. Valve and seat (renewable and interchangeable) are constructed of especially hardened, corrosion resisting steel. Bucket is formed from sheet copper. Cover cap screws are steel.

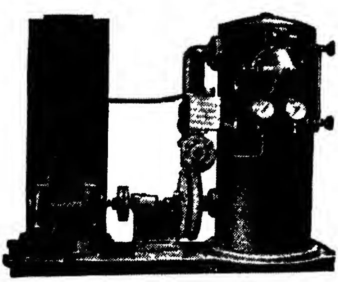
DUNHAM FLOAT AND THERMOSTATIC TRAPS

30 Series—For Operating Pressures up to 15 lb Gage

Comprised of a cover, mechanism assembly and body. Thermostatic disc and valve controls flow through a cored passage between trap body and discharge tapping for release of air. The float is cuprous material. Float valve and seat are monel metal. Thermostat elements are interchangeable. Trap body readily removed without disturbing piping connections to fully expose working parts for inspection. All traps are tapped so that gage glass set may be applied, but traps are shipped plugged. Bottom plug can be used for flushing out trap.

Dunham Closed Float Traps In Same Capacities

Vacuum Pumps

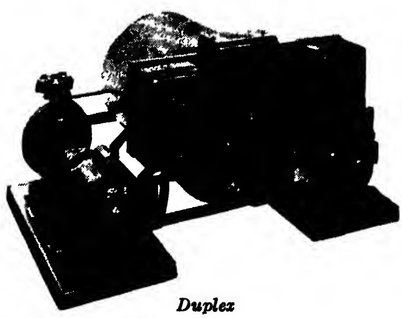


Type VR

Made in 9 sizes from 2500 to 65,000 EDR to be applied in accordance with EDR load, no allowance necessary for piping. Furnished as single or duplex units with a separate accumulator tank to take care of low returns. Each automatic pump has its own control panel wired through to motor. Selector switch on panel permits pump to operate on full vacuum and float control.

Pump and motor assembled on rigid cast iron base. Bronze fitted centrifugal pump has non-corrosive shaft. Enclosed type Impeller. Liberal size ball bearings.

Condensation Pumps



Duplex

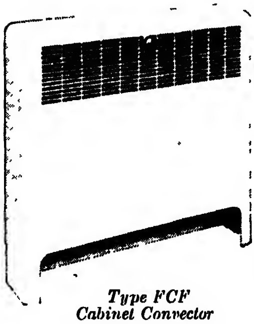
Pump and motor assembled on rigid steel base. Bronze fitted centrifugal pump has non-corrosive shaft. Enclosed type impeller. Liberal size ball bearings.

Type CH-Model C, Single and Duplex—Capacities 2000 to 50,000 sq ft EDR discharge pressures 20 to 50 lbs; 60 cycle d.c. or a.c. 1750 rpm; 25 or 50 cycle a.c., 1450 rpm.

Type CHH-Model C, Single and Duplex—Capacities same as type CH. Discharge pressures 20, 30, 40, 50 and 70 lb 60 cycle d.c. or a.c. 3450 rpm; 25 or 50 cycle a.c., 2850 rpm.

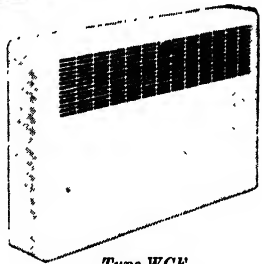
DUNHAM CABINET CONVECTORS

With Non-Ferrous Heating Element



Type FCF Cabinet Convector

Type "T" Heating Element



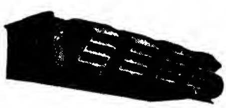
Type WCF Cabinet Convector

DUNHAM CONVECTORS

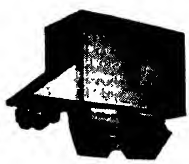
can be used on steam or vapor and gravity or forced hot-water heating systems. EDR capacities with 1 lb steam, 65 F entering air range from 12 sq ft EDR to 108 sq ft EDR. Made in 4 in., 6 in., 8 in. and 10½ in. widths; 20 in., 24 in. and 32 in. enclosure heights; casing lengths from 18 in. to 64 in. Casings are constructed of No. 18 gage steel with removable fronts. Outlet grille is punched in the casing front. Heating element is copper or aluminum fins on seamless drawn, round copper tubes brazed to bronze headers. These elements withstand hydrostatic test pressures of 400 lb per sq in and are suitable for operation on 150 lb steam or water working pressure. Cabinets also available in recessed types, semi-recessed types and in sloping top.

DUNHAM UNIT HEATERS

Type R—Wall Type



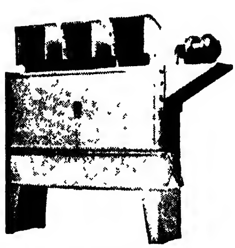
Type R—Ceiling Type



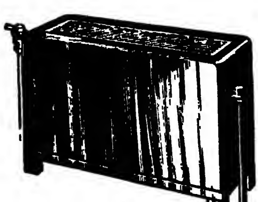
Type R—Inverted Wall Type

Type C—Built in various sizes up to 2000 sq ft EDR.

All units have belt driven centrifugal type fans. Constant speed 1750 rpm.



Type R—Floor Type with mixing damper also furnished with elongated nozzles



Type M—Floor Mounting

Type V—Horizontal propeller fan type. Built in various sizes from 65 up to 1500 sq ft EDR.

Dunham Type M Unit Heaters can be installed in any position, discharging their air directly upwards, directly downwards, or horizontally. Units must be ordered for the position in which they are to be set, so that proper mounting can be provided. Non-Ferrous Coil.

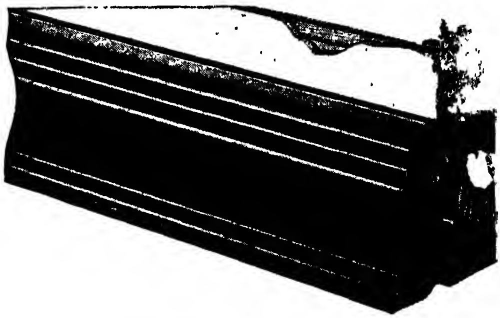
DUNHAM AIR CONDITIONING UNIT

This single unit, when correctly installed, supplies heated air in winter and cooled and dehumidified air in the summer, with filtering, ventilation and circulation of air the year 'round. These combined services are rendered with a minimum requirement of space and a high functional efficiency which guarantees genuine satisfaction and genuine economy.

DUNHAM TYPE OTS HEATING ELEMENT

An extended surface heating element made entirely of steel. Light in weight, and has unusual heating capacity. Each element is made up of 1¼ in. steel pipe with No.22 gage heavy fins mechanically attached, eliminating the use of a solder bond without sacrifice of heat transfer. Each fin, when pressed on the pipe, interlocks with the preceding fin—forming an exceedingly tight and permanent mechanical joint. The complete unit is painted with heat-resisting zinc chromate black enamel. Lengths from 18 in. up to 144 in. can be supplied in 6 in. increments. Standard units are threaded at each end with standard pipe threads.

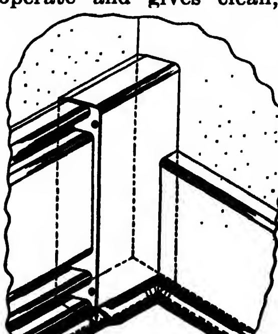
DUNHAM BASEBOARD



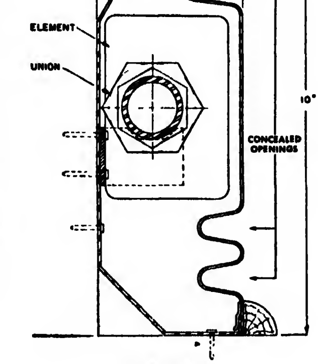
The Dunham Baseboard Simplicity Heating System is completely automatic. A room thermostat starts and stops the water circulator to provide just the right amount of heat. An aquastat starts and stops the heating device on the boiler to maintain a minimum water temperature and provide adequate hot water for washing, bathing and other domestic purposes. This automatic operation, plus the high efficiency and complete simplicity of the Dunham System, cuts installation and operation costs to a *fraction* of what you pay for old-fashioned heating systems.

Smart design and easy maintenance are combined. The specially engineered baseboard comes in an attractive style with completely hidden louvres. It's easy to keep clean, too, because the Dunham Baseboard has no outside openings to catch dirt—and as it goes completely to the floor there is no hard-to-clean pocket of accumulated dust. The baseboard cover is easily removed and all parts are readily accessible. Attractive in appearance—easy to install—simple to operate and gives clean, comfortable, satisfying, healthful warmth . . . from wall to wall . . . from floor to ceiling.

It wraps a house in a blanket of warmth. Designed and tested at odd moments during the long war years, an independent laboratory says, "Its performance was better in a basementless house than any other device that has been tested."



Corner Detail Showing Butting of Interior Partition



ELEMENT SECTION

William S. Haines & Company

1010–12–14 Wood Street, **Philadelphia 7, Pa.**

Manufacturers of

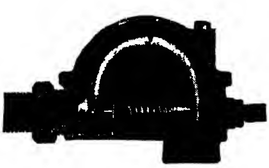
EQUIPMENT FOR VAPOR AND VACUUM HEATING SYSTEMS

PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Traps, Air Eliminators, High Pressure Thermostatic Traps, Boiler Return Traps, Radiator Valves.

HAINES F & T TRAPS

Designed to handle large quantities of condensation. For dripping steam mains, unit heaters, hot water generators, etc. Cannot become air bound as it has a thermostatically controlled air by pass. Sizes ¾ in., 1 in., 1¼ in. For pressures to 30 pounds.

HAINES MEDIUM PRESSURE TRAPS

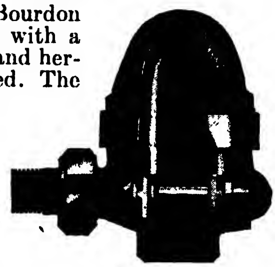


A ruggedly constructed bolted case trap. Ideal for hospital and kitchen equipment and all process work operating on pressure up to 100 pounds.

HAINES MODULATING VALVES

A packless valve assuring positive and leak proof performance. Completely opens or closes or less than a full turn of handle. Can be furnished with wheel or lever handle or lock-shield.

HAINES RADIATOR TRAPS

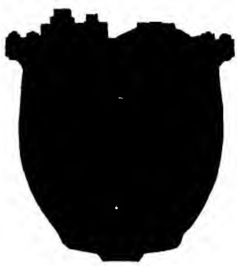


The thermostatic element in all Haines Traps is a Bourdon tube, charged with a volatile fluid and hermetically sealed. The expansion and contraction of the fluid, under varying temperatures, furnishes the operating power. The vertical seat of this trap prevents it from becoming inoperative from scale or other foreign matter.

HAINES HIGH PRESSURE TRAPS

For dripping high pressure mains, laundry equipment and all process fixtures with working pressures up to 125 pounds.

HAINES BOILER RETURN TRAPS



For vapor and atmospheric heating systems. Assures positive circulation by venting the air and returning the water of condensation to the boiler. Has no stuffing boxes or packed joints to leak air or water.

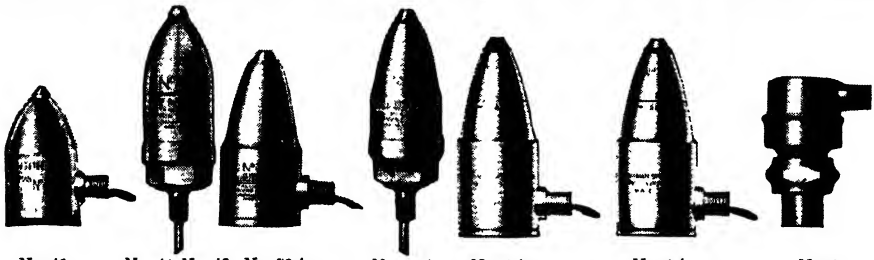
Each device is individually tested, factory adjusted and guaranteed.

Hoffman Specialty Company

General Office and Factory
1001 York Street, **Indianapolis 7, Ind.**
Sales Representatives in Principal Cities

Manufacturers of Radiator Air Valves, Quick Vents and Air Eliminators for all types of Steam and Vacuum Heating Systems—Steam Traps of all kinds—Radiator Supply Valves—Vacuum and Condensation Pumps—and Hot Water Automatic Heat Control Systems.

RADIATOR AIR VALVES FOR STEAM AND VACUUM SYSTEMS

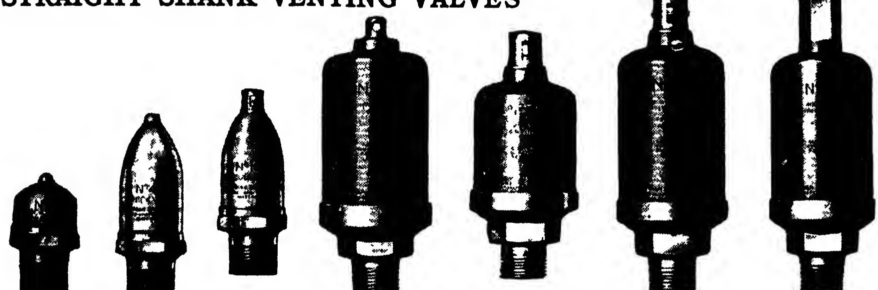


No. 40 — No. 41 No. 43 and No. 45 — No. 70A — No. 71A — No. 2A — No. 1A — No. 3

No. 40 Steam—Hoffman patented tongue syphon—1/8 in. connection—fixed port.
No. 41, 43 and 45 Steam—Straight shank for convectors—telescopic syphon—1/8 in., 1/4 in., and 3/4 in. male, 1/2 in. female connections.
No. 70A Steam—Tongue Syphon—Non-adjustable, single port—1/8 in. connection.
No. 71A Steam—Straight shank for convectors—telescopic syphon 1/8 in. connection.
No. 1A Steam—Tongue Syphon—ADJUSTABLE air opening—1/8 in. connection.
No. 2A VACUUM—Tongue Syphon—ADJUSTABLE air opening—1/8 in. connection.
No. 3 Steam—For Airline or PAUL systems—1/8 x 1/4 in. conn.—union tailpiece.

All radiator air valves operate on 10 lb max. press.

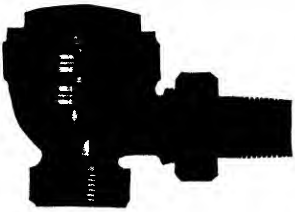
STRAIGHT SHANK VENTING VALVES



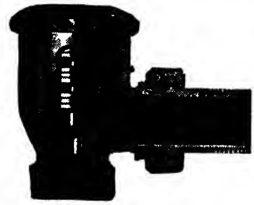
No. 4 — No. 4A — No. 16A — No. 74 — No. 75 — No. 76 — No. 79

No. 4 Steam Mains—Will not close against water—3/4 in. male, 1/2 in. female connection.
No. 4A Steam Mains—Float closes against water—3/4 in. male, 1/2 in. female connection.
No. 16A VACUUM Mains—Float closes against water—3/4 in. male, 1/2 in. female connection.
No. 75 Steam Mains—Medium systems—has float—3/4 in. male, 1/2 in. female connection.
No. 76 VACUUM Mains—Medium systems—has float—3/4 in. male, 1/2 in. female connection.
No. 75A Steam Mains—Large systems at low pressure—has float—3/4 in. male, 1/2 in. male connection.
No. 76A VACUUM Mains—Large systems low pressure—has float—3/4 in. male, 1/2 in. female connection.
No. 74 Unit Heater Vent Valve—Operates 0 to 35 lb. Vents air at any pressure and whether rising or falling—can be used on steam mains—3/4 in. male, 1/2 in. female connection.
No. 79 Hot Water Vent Valve—Positively removes air from piping of any hot water system. Max. pressure 75 lb—3/4 in. male, 1/2 in. female connection at base and tapped at top for 1/4 in. pipe connection.

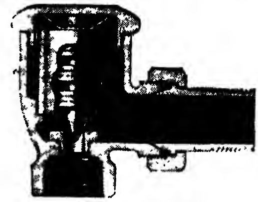
LOW, MEDIUM AND HIGH PRESSURE THERMOSTATIC TRAPS



Low Pressure



Medium Pressure



High Pressure

Low pressure traps have brass bodies, caps, and union nut and tail piece. 17C is made in Angle, Swivel and Vertical patterns. 8C is made in Angle and Straightway patterns. 9C is made in Angle pattern only. Thermostat and seat both renewable.

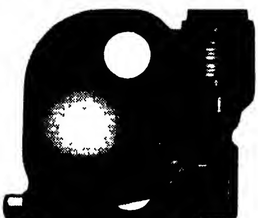
No. 17C Capacity 200 sq ft EDR 15 lb pressure ½ in. connection
No. 8C Capacity 400 sq ft EDR 25 lb pressure ³/₄ in. connection
No. 9C Capacity 700 sq ft EDR 25 lb pressure 1 in. connection

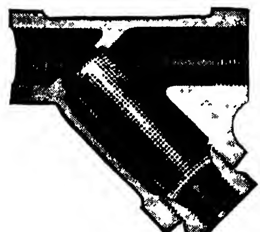
Medium Pressure Nos. 8 & 9 and High Pressure Nos. 8H & 9H have all bronze bodies and caps with union nut and tailpiece. Thermostats are 6 diaphragms of special non-corrosive metal. Thermostats and seats are renewable. ½ in. sizes are furnished in Angle, R.H., L.H., and Straightway patterns, others in Angle only. Medium Press. 50 lb limit. High Press. 125 lb.

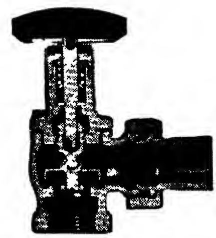
Capacities—Lb Condensate per Hour—Working Pressure—Lb per Sq In Gage

Traps		5	15	25	50	Traps		25	50	100	125
8	⅜"	100	180	235	400	8H	⅜"	235	400	550	590
8	½"	125	225	300	490	8H	½"	300	490	650	720
9	¾"	225	350	450	650	9H	¾"	450	650	875	950
9	1"	325	500	625	850	9H	1"	625	850	1125	1250

FLOAT TRAPS, DIRT STRAINERS AND SUPPLY VALVES







50 Series Float and Therm. Traps, are available in large capacities and four pressures, 15, 30, 60 and 125 lb. Used for venting and draining risers, steam mains, unit heaters, blast coils, etc. 50 Series Traps are made for easy servicing with all working parts mounted on cover. Remove four bolts to expose all parts. Pipe sizes are from ³/₄ in. to 2 in.

Radiator Supply Valves are made in sizes from ¹/₂ to 2 in. in Angle, R.H., L.H., and Straightway patterns. Brass bodies, union nut and tailpiece. Nos. 180 and 185 have non-rising handles. Both are packless. No. 186 is especially suited to vacuum systems.

Hoffman Dirt Strainers are self-cleaning Y type. Brass strainer cylinders and cast iron body. Sizes ¹/₂ to 2 in. for 125 lb pressure. Should be used in line ahead of all float and thermostatic traps. Also available with monel metal strainers.

CONDENSATION AND VACUUM PUMPS

Condensation Pump

Hoffman-Economy pumps are available in varying capacities, D.C. and A.C. current, single, two, or three phase, and in pressures up to 200 lb. Also made in single and duplex units for different capacities and pressures.

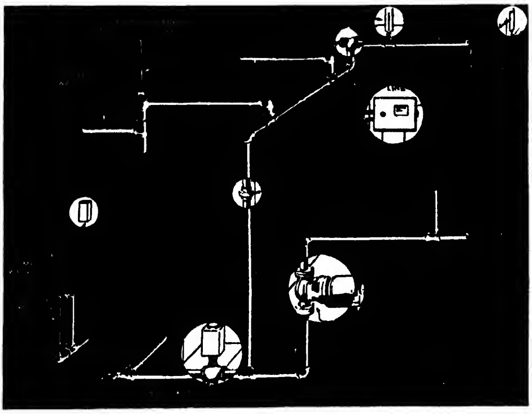
Vacuum Pump

HOFFMAN "PANELMATIC" HOT WATER SYSTEM CONTROLS

Designed for all Types of Radiant Heating

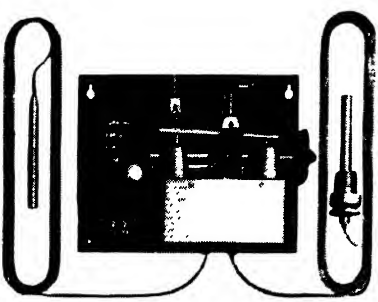
The Hoffman method vastly improves the ordinary forced hot water system by the application of *Continuous Circulation.* This method permits a smoothly modulated regulation of the heat supply. The Control Valve closes and the circulating stream by-passes the boiler as long as the heat requirements are satisfied.

When the circulating water begins to lose heat, the Control Valve is slowly opened by the Controller, permitting hot boiler water to enter the system. Thus, the delicate precision of the Hoffman Controller smoothly varies the temperature of the *continuously circulating* water so that the heat supply is always equalized with the heat loss and room temperature remains constant throughout all changes in the weather.



There is no type of building to which Hoffman Hot Water Controlled Heat cannot be applied. The system permits *zoning* of apartments, institutions, large residences and factories, thereby assuring a distribution of heat in direct relation to either personal temperature preference or to the functional activities of the building.

HEATING COMFORT IS ACHIEVED BY THESE SIX HOFFMAN SPECIALTIES



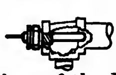
The Panelmatic Controller

The brain of the Hoffman system. It automatically maintains a constant comfort condition regardless of the outdoor temperature. Its accurate balancing mechanism electrically opens or closes the Hoffman Hot Water Control Valve as required.

Outdoor Temperature Anticipating Bulb

This Bulb transmits changes in the outdoor temperature to the balancing mechanism of the Comfort Controller.

Hot Water Temperature Bulb



Accurately relays temperature changes of the supply main water to the balancing mechanism of the Hoffman Comfort Controller.

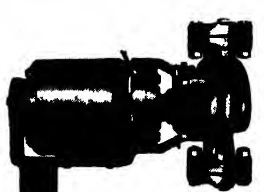
Hot Water Control Valve

Opens only sufficiently to supply the correct amount of water to maintain the proper water temperature being circulated through the system.

Thermometer. Should be installed about 6 in. from submerged Water Temp. Bulb.

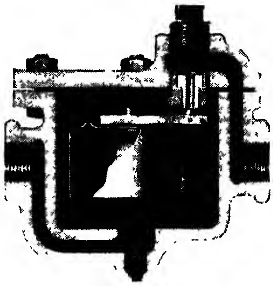
Balancing Orifice

A calibrated orifice used to balance the circuits through the boiler and through the Hoffman circulating pipe.



Hoffman Circulator

A centrifugal pump of prescribed capacity and low in power consumption. Usually installed in the return main and continuously circulates the water through the system. Continuous operation puts less strain and wear on the pump motor than intermittent operation as any engineer will attest.

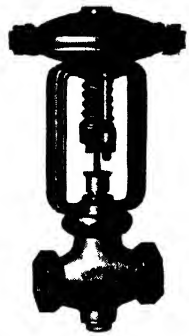


Series 600

HOFFMAN INVERTED BUCKET TRAPS

Working Pressures to 200 lbs

Simple mechanism assures high operating efficiency. Features include—straight-through pipe connection; simple seat adjustment; all working parts connected to bonnet and easily removed with it; stainless steel seat and pin.



Series 710

HOFFMAN PRESSURE REDUCING VALVES

Series 710	**Series 700**
Max. High Pressure. . 250 psi.	Max. High Pressure. . 200 psi.
Low Pressure Range 5 to 80 psi.	Low Pressure Range 1 to 25 psi.

Choice of types specifically engineered for your needs—continuous service, or, for tight sealing against low pressure side when there is no demand—and with typical Hoffman features of design.

Series 1100

HOFFMAN TEMPERATURE REGULATORS

Series 1100—for steam pressure to 150 lb per sq in. Standard temperature range—80°F to 250°F. Other ranges available on special order. Self-motivated. Hydraulically formed bellows of selected material. Rugged construction. For water heaters, convectors, fuel oil preheaters and other similar applications.

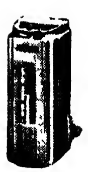
HOFFMAN "ZONET" ZONE CONTROL

for Steam and Hot Water System Service

Series 1200—Zonet systems or packages consist of heat anticipating Room Thermostat, motor operated Globe Type Valve, Transformer, Fuse Block and fuse, and in the case of steam zoning, a Vacuum Breaker. The motor operated, two position, single seated, packed globe type Valve (illustrated) is available in sizes from 1/2 in. to 6 in. Maximum service pressure: steam 100 lbs per sq in, water 150 lbs per sq in.

HOFFMAN ELECTRIC CONTROLS

for Heating System Equipment



Hoffman Room Thermostat

Hoffman offers a comprehensive selection of fine quality electric controls for steam, hot water and warm air heating systems. These controls are especially suited for use with other famous Hoffman heating system specialties and provide one source of supply, and one responsibility for satisfactory performance. Illustrated are two popular control units, typical of the complete Hoffman line.

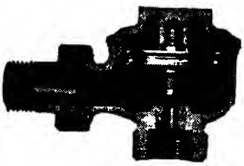
Hot Water Limit Control Immersion Type

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago 8, Ill.

**Representatives
In Principal Cities**

Illinois Thermo Radiator Traps



Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes thoroughly and seats perfectly at all times. Valve and seat are of hardened steel alloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test.
Ask for Bulletin.

Vapor Gauge

Vapair Vent Trap

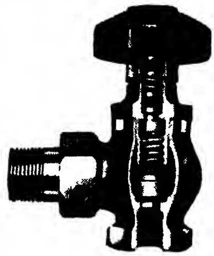
Boiler Return Trap

Illinois Selective Pressure Control Systems

Selective Controller

An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system individually engineered to meet exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation.

Illinois Radiator Supply Valve



Quick - opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

Illinois Vapor System

A two pipe low pressure steam circulating system which may be installed in any type of building, where the condensate can return to the boiler by gravity.

A sensitive damper regulator or other means of automatic control is used to control initial steam pressure above, at or below atmospheric pressure. Steam is regulated at the radiators by Illinois Packless Supply Valves. Condensate and air are discharged from the radiator through Illinois Thermostatic Radiator Traps. In the boiler room a Vapair Vent Trap and Boiler Return Trap are installed near the boiler. The vent trap eliminates air from the system and the Return Trap insures return of condensate to the boiler.

The system and the piping arrangement are simple. No metering orifices or vacuum pumps are needed. This system will be found suitable for many installations where low first cost and low operating cost are of prime importance. May be used with unit heaters or any type of radiation.

Illinois Float and Thermostatic Traps

Series G

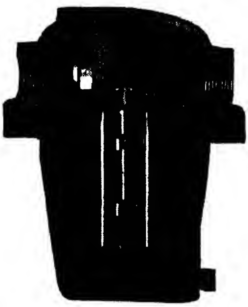
Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago 8, Ill.

**Representatives
In Principal Cities**

ILLINOIS STEAM TRAP



Series 30

Valve and stem are separate from the bucket and operated only by the bucket at extreme top and bottom of travel — result—*valve is always either full open or tight closed.* Provided with continuous thermostatic air vent. No wire drawing or cutting of valve and seat which are of hardened steel alloy.

Illinois Thermostatic Traps for High Pressures

Series HG

Maximum working pressure 150 pounds. Used where neat appearance and compactness are desirable, as for trapping sterilizers or water stills in hospitals; steam jacketed kettles, coffee urns, warming tables and for process work. Also used extensively for air vents on blast type drying heaters. Multi-diaphragm of phosphor bronze. Heavy duty bronze body. Made in three sizes.

These traps are also furnished for medium pressures. *Write for literature.*

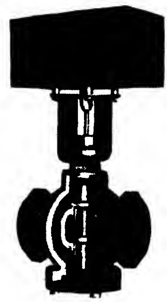
Steam and Oil Separators

Vertical Steam Separators

Eclipse steam separators are made in both horizontal and vertical type, standard or extra heavy.

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point.

Illinois Motorized Valves (on and off)



Type E3

For automatic control of steam temperatures and pressures to prevent overheating and conserve steam; to control fluid levels; and to regulate flow in hot water heating systems. May be operated by any automatic contact device or by manual switches.

Furnished in three types.

Pressure Regulating Valve, Semi-Steel Bodies, Bronze Trim

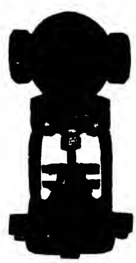


Fig. 121

Furnished in either single seat or double seat type as service requires, for the control of steam, air or gas. Control spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with proper size diaphragm and proper length spring to give satisfactory service under all operating conditions. Furnished also in weight loaded type, Fig. 71.

Write for Bulletins.

Non-return Valves

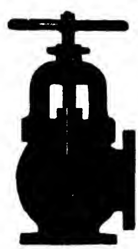


Fig. 260

Placed between boiler and header to prevent return of steam to boiler. Sensitive in operation. Extra heavy semi-steel bodies with bronze trim for 250 lbs steam working pressure. Bronze dash pot and water sealed pistons prevent valve sticking. Globe and angle patterns from 4 in. to 12 in.

Jas. P. Marsh Corporation

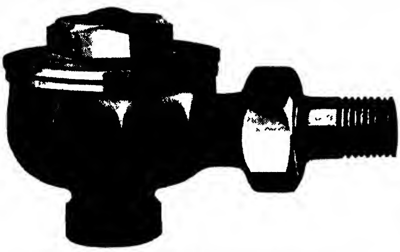
Dept. 5, Skokie, Illinois

Branches in Principal Cities

Marsh products include: Pressure, Vacuum and Compound Gauges; Dial Thermometers; Steam Traps; Vents; Packless Radiator Valves and other heating specialties.

Thermostatic Diaphragm Radiator Traps

These efficient traps are equipped with a phosphor bronze diaphragm, consisting of two wafers of tinned phosphor bronze, drawn and spun to per-

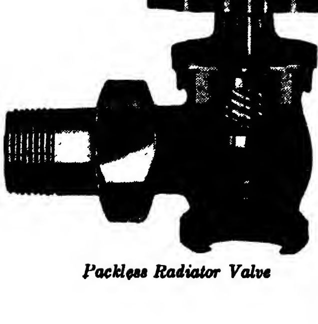


Thermostatic Diaphragm Radiator Trap

fection of temper. The wafers are spun together and soldered to form a seamless, sensitive, powerful expanding member—not easily fouled by dirt and foreign matter. Diaphragms are charged with a volatile fluid making them self-equalizing for use on pressures below atmosphere to 15 lbs gauge. Traps are factory adjusted.

Packless Radiator Valves

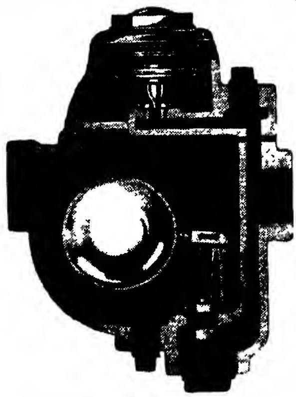
Marsh all-metal packless valves are truly packless. These valves contain



Packless Radiator Valve

no packing to deteriorate, wear or crack, and are simple in design, with ample strength where strength is required. The principles upon which they are designed have been proved sound over many years of service. Valves operate easily—opening or closing tightly with less than one full turn. All valves are individually tested. Adaptable for use on hot water—forced or gravity systems—as well as all steam heating systems.

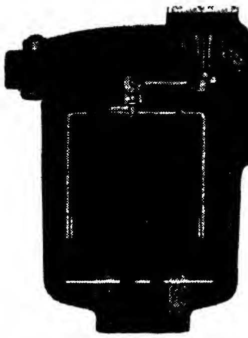
Marsh No. 12 Float and Thermostatic Traps



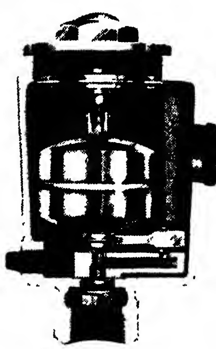
Float and Thermostatic Trap

Marsh Heavy Duty Float and Thermostatic Traps are designed for removal of air and condensate from steam mains, branches, or risers, unit heaters, steam coils, etc. The size and weight of the trap permits installation in the piping without any other means of support. Condensation is discharged through a float-operated valve located at the lowest point inside the trap body. Air vent is located in a by-pass in the cap or cover of the trap. Air passes through a passageway and out through the trap outlet. Construction permits removal of mechanism without disturbing the piping.

Marsh No. 500
Inverted Bucket Type Trap



This type is ideal for all types of hospital and kitchen equipment or similar service where a considerable volume of condensate is handled. The trap is self-venting, which, combined with the large water capacity assures unusually high efficiency in removing condensate, air and gases.



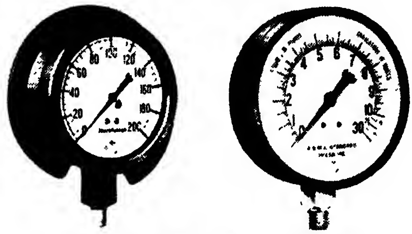
Marsh No. 17
Float and
Thermostatic Trap

This trap is designed for removal of air and condensate from short steam mains, branches or risers. Operating characteristics adapt it for unit ventilators, unit heaters, and other equipment subjected to freezing temperatures when heating system is not in operation. Outlet discharge is water sealed at all times. Air vent is located in trap bonnet and air is normally discharged through a port directly to the outlet connection. All working parts are made accessible by removing bonnet. The piping is all the support required for the No. 17 Trap.

Marsh Low Pressure Gauge

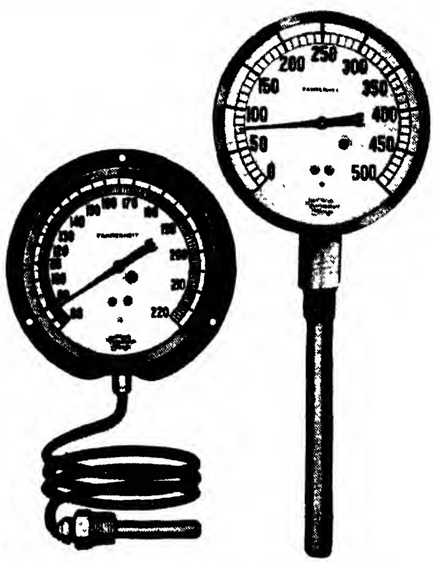
The Marsh *A.S.M.E.* standard, low pressure gauge will contribute to the economy and improve the operation of any type of steam boiler. It is finely built throughout and is available with the Marsh "Recalibrator" for quickly and easily resetting the hand to zero when the gauge is knocked out of adjustment.

Marsh Gauges include vacuum and compound types in a wide range of designs covering all services and pressures. Over 75 years of gauge manufacturing has reached its highest achievement in the Marsh "Mastergauge" for use where high pressures and temperatures are present and where maximum stamina and accuracy are essential.



Marsh Dial Thermometers

Have the same basic refinements found in Marsh Gauges. Typical Marsh Thermometers of bourdon tube type in self-contained and distant reading types are illustrated. They are available in either vapor tension or gas-filled types. Bi-metallic types of dial thermometers are also available. Practically all temperature ranges up to 750° F. are covered. The "Recalibrator" is standard in all Marsh bourdon tube type thermometers. The Marsh line also includes recording thermometers.



Ask for complete information covering any problem involving traps, vents, gauges, dial thermometers, packless valves, etc.

Sarco Company, Inc.

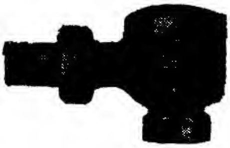
Empire State Bldg., **New York 1, N. Y.**

Branches in Principal Cities

SARCO CANADA LIMITED, 496 CHURCH ST., TORONTO 5, ONT.

PRODUCTS—A complete line of Specialties for Steam and forced hot water Heating Systems, and automatic control for same, combined with a competent engineering service to architects and heating engineers to assist them in providing modern heating.

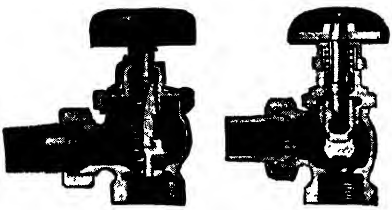
SARCO RADIATOR TRAPS



Radiator Trap, Type H

Type H is the standard radiator trap for vapor and vacuum systems. It is equipped with the well known Sarco heavy walls bellows, drawn from flat blanks and helically corrugated in our own plant. It operates noiselessly and positively at pressures from highest vacuum to 25 psi.

Body and cap are of cast brass, rough brass finish; self-aligning valve head and renewable seat of hard bronze; union connection on inlet.



Radiator Valves

Type 45 *Type SM*

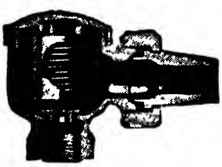
Available in ½ in. and ¾ in. sizes, angle, straightway or corner offset patterns; also 1 in., angle style only. *Catalog No. HV-150.*

SARCO RADIATOR VALVES

Sarco offers two types of valves; bellows packless type 45 wherein the valve stem is sealed by a standard Sarco bellows, positively preventing air leakage into the heating system; also "spring-loaded-packless" type SM. Both can be furnished with the modulating feature, including proportioning disc and indicating dial.

Valves are made in angle and straightway patterns, wheel handles or lock shield tops. Also available for hot water systems.

Bodies of all valves are cast brass, rough brass finish; outlet fitted with unio connection; sizes ½ in. to 1½ in. *Catalog No. HV-150.*



N-100 Medium Pressure Trap

SARA N-100 TRAP

For high pressure radiators and heating coils in stationary and marine service, and for hospital and kitchen equipment. Has full length protecting shield and stainless steel valve head and seat. Sizes ⅜ in. to 1 in. pressures to 100 lb. *Catalog HV-190.*

Also S-65 for pressures to 65 psi.

Float-Thermostatic Trap

SARCO FLOAT-THERMOSTATIC TRAPS

For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ in. to 2 in. Pressures up to 200 psi. *Catalog HV-450.*

Inverted Bucket Trap

SARCO INVERTED BUCKET TRAPS

Are recommended for high pressure unit heaters and sometimes preferred for kitchen and laundry equipment. Strainers are built right into these sturdy traps. Seats and valves are stainless steel and renewable. Automatic air vents can be furnished for extra rapid removal of air. Available in sizes ½ in. to 2 in. for pressures up to 900 psi. *Catalog HV-350.*

SARCO ALTERNATING RECEIVER


Alternating Receiver

A complete line of boiler return traps for vapor systems. Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

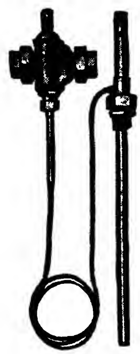
Made in four sizes for up to 14,000 sq ft of radiation. *Catalog HV-165.* Same type available as a pumping trap for pressures to 100 psi.

SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement. Available in three sizes, for systems up to 15000 sq ft radiation. All are equipped with float valves to stop water escaping through the vent and with check valves to prevent ingress of air when system is under vacuum.

Also several types for hot water heating systems. *Catalog HV-170*

SARCO SELF-CONTAINED TEMPERATURE REGULATORS


Type TR-21 Standard for hot water storage tanks, fan units, etc.


Water Blender Type MB

Sarco Temperature Regulators are simple self-operated valves—the only self-contained units that use the irresistible force of liquid expansion. No stuffing boxes to leak no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 300° F. *Catalog HV-600.*

SARCO WATER BLENDERS AND TEMPERING VALVES


Water Blender Type DB

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. *Catalog HV-800.*

SARCOTHERM HOT WATER HEATING SYSTEM

A simple, all-mechanical system for the control of radiator temperatures in direct relation to outside temperatures. Radiation is balanced by Sarcoflow fittings in the radiator outlets.

The Sarcotherm three-way valve recirculates a varying proportion of water around the boiler and back to the system as dictated by the thermostatic bulb outside the building. Write to Sarcotherm Controls, Inc., 280 Madison Ave., New York 16, N. Y. for *Catalog No. HV-1.*


Sarcotherm Weather Control Valve

SELF-CLEANING STRAINERS

For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 600 psi, with brass, iron or monel screens. Available in sizes ¼ to 8 in. *Catalog No. HV-1200.*

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

Webster HEATING SYSTEMS

Main Office and Factory:
Camden, New Jersey

Webster Nesbitt UNIT HEATERS

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Licensees and Manufacturers for Canada and Newfoundland:
DARLING BROS., LTD., P. O. Box 187, Montreal, Canada

THE COMPANY

Warren Webster & Company have specialized for sixty years in the field of steam circulation and steam distribution, particularly vacuum, vapor and low pressure steam heating of buildings, and medium pressure steam in industrial and process heating applications.

The specialized experience of the Company is available through engineers at the Home Office and through the Representatives listed above. These engineers are thoroughly conversant with methods and equipment to control steam distribution and heat transfer, assuring maximum heating effectiveness with minimum fuel or steam consumption. Webster Representatives are prepared to supply on request full technical, availability and price information on all Webster Systems and Products.

WEBSTER HEATING SYSTEMS

There is a type of Webster Heating System to meet practically every need and purpose. These include Webster Steam Heating Systems for larger buildings of almost every type, and Webster Baseboard Heating, a hot water system particularly suited for small residences and other small buildings.

Webster Steam Heating Systems are low pressure, two-pipe systems in which steam is delivered to radiators and other heating surfaces through supply piping and water of condensation and air are removed through separate return piping. Webster Radiator Valves and Thermostatic Traps are installed respectively on the supply and discharge connection of each radiator. Webster thermostatic or float and thermostatic traps assure removal of water of condensation and air from the piping.

Available with vacuum return, or with open return (vented to the atmosphere) with either Condensation Pump or Boiler Return Trap and Vent Trap to return water to the boiler, or with Vent Trap alone where condensate is wasted to the sewer, or in appropriate small installations.

Webster Vacuum System—A conventional vacuum heating system in which the return mains are joined together and connected to the suction end of one or more vacuum pumps which remove air and water of condensation and assists circulation by maintaining a lower pressure in the return than in the supply piping.

Webster Type "R" System—A two-pipe, low pressure or vapor heating system. Water of condensation is returned to the boiler by gravity, prompt return being assured regardless of variations in boiler pressure through the operation of a Webster Boiler Return Trap and Vent Trap in combination. Equipment is available in sizes to care for systems ranging from the smallest to 16,000 sq ft EDR. Where desired, or where gravity return is not possible, a Condensation Pump may be substituted for the Boiler Return Trap Combination.

Webster Type "V" System—Employs only a Webster Vent Trap. For installations of 1000 sq ft EDR or less with oil burner, stoker or gas burner; with vaporstat having cut-in pressure of about ¼ lb and cut-out pressure of about ¾ lb (not pressurestat), lockswitch or protector relay and one or more key room thermostats. Vent Trap at ample height above water level. Boiler Protector, or at least a low water cut-out.

Webster Moderator Systems—These are all Webster Steam Heating Systems with vacuum or open return *to which are added* (a) accurately sized metering orifices in radiators and other heating surfaces to balance distribution and permit "partial filling" of all radiators practically simultaneously and at various rates of steam flow, (b) Automatic control by Outdoor Thermostat for variations in outdoor temperature, (c) Manual Variator to provide for convenient adjustments for heating up, reduced night heating, shut off, etc.

"E" Series Moderator Controls—In this series the Outdoor Thermostat and Variator position a motor-operated Steam Control Valve through an Electronic Differential Pressure Control Cabinet to produce continuous steam delivery and heating effect at the radiators with automatic variation in heating for changes in outdoor temperature and automatic adjustment to compensate for variations in steam supply pressure.

"EH" Series Moderator Controls—In this series the Outdoor Thermostat and Variator may control a motor-operated steam valve or directly control oil or gas burner or stoker through a cycling Control Cabinet. Steam delivery is intermittent but in short cycles so that heating effect is substantially continuous, particularly with cast iron radiation.

Webster Continuous Flow Control—for forced circulation Hot Water Heating Systems. Outdoor Thermostat and Variator control throttling-type valve or directly control oil burner, gas control or stoker motor. Water flows continuously through the system. Heating is continuous and adequate at all times. Applicable to Baseboard, Convector, Radiator or Panel Heating.

Webster Baseboard Heating—A patented forced circulation hot water heating system in which the heating element fits behind a specially built metal baseboard. Air enters at the floor line, passes over the finned heating element, is warmed and comes out of slots at the top of the base-board. The heating element is a copper tube with copper fins running in a continuous loop around the exposed walls of the house—a separate loop for each floor.

Uses less material and less labor than conventional radiator heating systems, while providing all the advantages claimed for forced hot-water, plus radiant effect from warmed baseboards and walls, plus natural convected air movement essential to comfort. Temperatures vary less than 2 deg from floor to ceiling.

STEAM HEATING AND PROCESS SPECIALTIES

Radiator Valves—Choice of spring retained packing, Type BW-P or Sylphon Bellows Packless Series 600-S. ½ in., ¾ in., 1 in., 1½ in. sizes. In angle, right and left hand; straightway, with single or double union. Spring retained packing. For low pressure vapor and vacuum steam heating service.

Double Service Valve—¾ in. and 1 in. sizes. Incorporates Webster Thermostatic Trap and Webster Radiator Supply Valve in a single compact unit. Drips down-feed riser and serves as supply valve to radiator or convector. Saves six fittings and installation work.

Thermostatic Traps—Series "7" (diaphragm type) and Series "5" (bellows type) for radiators and drips. ½ in., ¾ in. and 1 in. sizes. There are six body models for the ½ in. size alone. Maximum pressure, 25 lbs per sq in. For low pressure vapor and vacuum steam heating service. Series "78" for process. ⅜ in., ½ in., ¾ in. and 1 in. sizes. Class 2 for pressures of 60 to 150 lbs. Used to discharge air and water from heating coils of any apparatus using steam at process pressures.

Heavy Duty or Drip Traps—Series "26" Float-and-Thermostatic for heating and air conditioning. Most used sizes: 00026, 0026, 026. Pressures up to 15 lbs per sq in. Made for the pressure and capacity conditions encountered at all drip points. Series "79" Float-and-Thermostatic for process. For pressures up to 150 lbs per sq in. For use where large volumes of hot condensate must be handled more quickly than is possible by thermostatic traps alone.

Dirt Strainers—½ in. to 6 in. sizes. Maximum working pressure 150 lbs per sq in. Placed ahead of traps in return lines of steam-using equipment and steam heating systems to catch dirt and other particles, preventing them from impairing the tightness of the traps.

Boiler Protectors—One size, with ¾ in. connections with or without electrical cut-out switch. Maximum pressure 15 lbs per sq in. Maximum cold water main pressure, 150 lbs per sq in., minimum not less than 25 lbs per sq in. Prevents breakage in low pressure heating boilers when the water level becomes inadequate.

Vacuum Pump Governors—In sizes ¾ in. to 3 in. Standard valve furnished for pressures up to 150 lbs per sq in. Special valve available for higher pressures. Govern flow of steam to steam-driven vacuum pumps.

Suction Strainers—Maximum working pressures 15 lbs per sq in. Installed ahead of vacuum pump to prevent dirt from damaging pump.

Vacuum Breakers—Sizes ¾ in., 1 in. and 1¼ in. adjustable on job. Size ⅛ in. for radiators is factory adjusted. For automatic breaking of vacuum at predetermined setting in heating systems, feed water heaters, hot water generators, industrial pressure chambers, etc.

Expansion Joints—Crosshead or internally guided. In single slip and double slip models in most widely used pipe sizes. For pressures up to 200 lbs per sq in. For steam, hot water, hot oil, hot gas, and condensate return lines.

RADIATION AND HEATING SURFACE

Webster System Convector Radiation—Non-ferrous convector radiation. Each Webster System Radiator includes a complete enclosure of furniture steel with baked prime coat. Contained within the enclosure is a prefabricated unit, combining heating surface, valve, trap and union connections, shipped ready to connect to supply and return piping.

Webster System Radiation was first offered in 1932. From its introduction until withdrawal from the market because of war conditions, more than 1,000 installations were made.

Now, available in an improved design, using the same basic material, copper tubing and aluminum fins. Increased rigidity of the tubing and the development of a new method of manufacturing has produced a fin surface of unusual rigidity, free of expansion and contraction noises.

Webster Type WI Extended Surface Radiation—Covered by patents and patents pending. Completely non-ferrous, being made up of specially annealed copper tubing with rib-reinforced, square pressed aluminum fins. Available in two fin sizes. 3 in. fin size in 2, 3, 4, 5, and 6 ft; 4 in. fin size in 2, 4, 6 and 8 ft lengths. Can be used to advantage in many buildings with steam or hot water heating, particularly where floor space or vertical wall space is limited.

WEBSTER-NESBITT UNIT HEATERS

Are manufactured by John J. Nesbitt, Inc., Philadelphia 36, Pa., and are distributed solely through Warren Webster & Company, Camden, New Jersey. Designed to circulate large volumes of air at comparatively low temperatures, assuring quick heating.

Ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with standard test code of Industrial Unit Heater Association and A.S.H.V.E.

PROPELLER FAN UNIT HEATERS

Fig. 1. Standard Propeller-Fan Type.

Designed to incorporate four characteristics essential to both proper application and satisfactory performance: 1.) *Selective range of sizes.* Manufactured in nine sizes. Heating capacities vary from 34,700 to 338,000 Btu per hour. Air deliveries from 470 to 4800 cfm. 2.) *Quiet Operation.* All fans have blades of exceptionally large areas and of a shape to impart a gradual acceleration to the air stream. Ample spacing is maintained between the fan and heating element. Motors are of sleeve bearing type and equipped with isolators. 3.) *Durable lightweight Heating Elements.* Extended fin-and-tube type, constructed of copper condensing tubes and plate-type aluminum fins. 4.) *Modern Casing Design.* Compact suspended type. W-N 126.

SERIES "R" UNIT HEATERS

A neat, furniture steel cabinet enclosing a copper tube, aluminum fin heating element adaptable for steam or forced hot water systems; and two to five centrifugal fans belt-driven from an electric motor. Universal design offering wide flexibility. Especially adaptable where low noise levels are desirable. Variable-pitched motor sheave permits low or high speed operation of fans. Available in four sizes. Air deliveries with standard drive range from 518 to 1890 cfm. Steam heating capacities range from 158 to 588 EDR. Send for Catalog W-N 133.

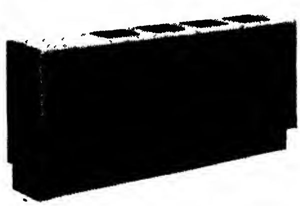


Fig. 2. Series "R" Unit Heater.

GIANT UNIT HEATERS

Fig. 3. Blower-Fan Type

Sturdy blower-fan units for the economical heating of large areas.

Standard (Non-Thermadjust) type. Used principally where heating is by recirculation only, and where constant heat output is desired during period of operation.

Thermadjust Type Employs dampers in front of casing and over face of heating element to provide mixing of unheated and heated air, producing heat output in accordance with requirements and continuous circulation of air volume.

Valve Controlled Type. Unit is of standard casing arrangement but equipped with Nesbitt Heating Surface and Steam-distributing Tubes which allows for automatic control of heat output.

Floor mounted, wall mounted, ceiling suspended, from 101,000 Btu, 2580 cfm, to 1,008,000 Btu, 17,800 cfm. Pub. W-N128.

LITTLE GIANT UNIT HEATERS

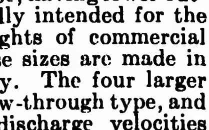


Fig. 4. Down-Blow Type.

Adaptable to a wide variety of applications and field conditions. Seven basic sizes, each with a choice of two (some units three) heating elements. The three smaller sizes are of the blow-through type, having lower outlet velocities generally intended for the lower mounting heights of commercial installations. These sizes are made in down-blow type only. The four larger models are of the draw-through type, and produce the high discharge velocities necessary to blow long distances. These four models are available in either horizontal or vertical down-blow arrangements.

Non-ferrous all-purpose heating elements designed for steam pressure up to 200 lb. gauge, saturated, and sturdy casings of modern design. Heating capacities range from 28,500 to 348,000 Btu basic steam ratings. Send for publication W-N 134.

Morehead Manufacturing Co.

2455 W. Warren, Detroit 8, Mich.

MOREHEAD BACK-TO-BOILER SYSTEMS

TILT-TYPE RETURN TRAPS — **VACUUM AND CONDENSER TRAPS**

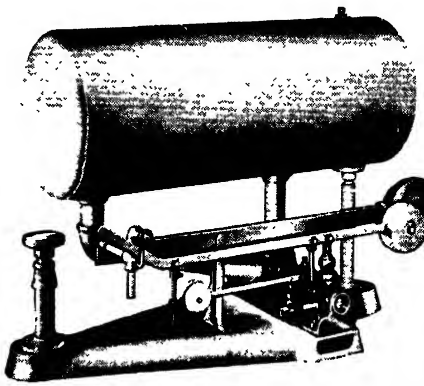
TILT-TYPE NON-RETURN TRAPS — **THREE-VALVE LIFT TRAPS**

RETURN TRAP. Completely efficient method of draining all types of heating, cooking, drying, evaporating and processing machines. Returns hot condensate to boiler as boiler feed water at its original high temperature. Consumes only about 10 per cent of the steam required by a pump. Deaerates boiler feed water. Automatic operation. All parts are outside readily accessible.

NON-RETURN TRAP. Primarily designed for drainage of steam headers, separators and all apparatus where the condensation is to be disposed of by means of the steam pressure from the space drained. Condensation flowing to the non-return traps forms a continuous path—once the tank tilts to empty, it will continue emptying as long as water continues to flow.

THREE-VALVE LIFT TRAP. Designed for draining condensate from low pressure, vacuum or varying pressure apparatus. Will also drain low pressure machines into higher pressure lines. Efficiently used as a lift trap in a double trap system. Lifts two feet for each pound of pressure. Will not steam or air bind. Not affected by temperature of incoming condensate.

WRITE DIRECT TO THE FACTORY for additional information and the name of your nearest Morehead representative.



Morehead Return Trap. All traps are similar in appearance and size.

DIMENSIONS APPLICABLE TO ALL TYPES OF MOREHEAD TRAPS

Trap Size	Height Inches	Width Inches	Length Inches	Net Weight Pounds	Shipping Weight Pounds
1	24¼	16½	31¼	120	165
2	28½	17½	35	160	225
3	34	21	44	240	320
4	36½	21½	51½	285	350
5	39½	30	58½	500	690
6	40	30	58½	520	700

DIMENSIONS AND CAPACITIES* OF MOREHEAD TRAPS

Trap Size			Size of Drum Inches	Size of Inlet and Outlet Connections Inches	Size of Steam Pipe Connections Inches	Capacity of Water in Lbs. Per Hour		
Return Trap	Non-Return Trap	Three Valve Lifting Trap				Return Traps	Non-Return Traps	Three Valve Lift Traps
1	21	31	10 x 24	1	1	1050	2800	3000
2	22	32	12 x 30	1¼	1	1850	5000	5000
3	23	33	14 x 36	1½	1¼	4000	11000	10000
4	24	34	16 x 40	2	1¼	6000	13500	14000
5	25	35	18 x 42	2½	2	11000	14700	20000
6	26	36	18 x 42	3	2	15000	17600	

* Based on 50 lbs pressure per sq in.

Yarnall-Waring Company

Manufacturers of YARWAY **Steam Specialties**

133 Mermaid Ave., **Philadelphia 18, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—The Yarway Impulse Steam Trap is unique in that there is only one moving part, the simple valve (F). This trap is made of bar stock throughout, no castings used. For pressures up to 400 lb, body and bonnet of cold rolled steel, cadmium plated; cap of tobin bronze, valve and seat of heat treated stainless steel. For pressures 400 to 600 lb, trap is all stainless steel.

Operation—Movement of valve (F) is governed by changes in pressure in control chamber (K). At lower temperatures, condensate bypassing continuously through orifice in center of valve reduces chamber pressure below inlet pressure and valve opens, allowing free discharge of air and condensate through seat. As condensate approaches steam temperature, low chamber pressure causes vaporizing and the increased volume builds up pressure in control chamber, closing valve (F).

Other Advantages

Light Weight—Yarway traps need no support—½ in. trap weighs only 1⅜ lb. 2 in. trap weighs 8⅝ lb.

Small Size — Can be installed in cramped quarters—½ in. trap measures 2¼ in. long—2 in. trap, 4¾ in. long.

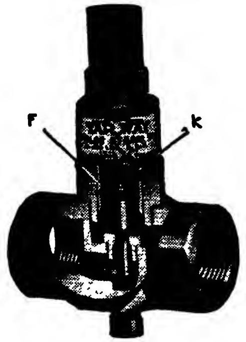
Will not air bind. *Require no priming.*

Insure quick heating.

Low Price—Often cheaper than repairing old traps.

Factory set to operate at all pressures up to 400 lb (or 600 lb) without change of valve seat.

Send for descriptive Bulletin T-1739.



List Prices, Weights and Dimensions

No. 60 Series—up to 400 lb and

No. 120 Series—up to 600 lb

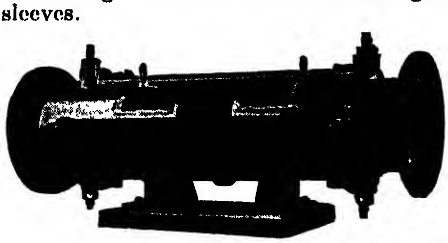
Size	Trap Complete	Weight Pounds	Length Inches
½" Nos. 60 or 120	$15.00	1¼	2⅝
¾" Nos. 61 or 121	22.00	2	3
1" Nos. 63 or 123	31.00	2½	3⅜
1¼" Nos. 64 or 124	48.00	4	3¾
1½" Nos. 66 or 126	68.00	5¾	4¼
2" Nos. 67 or 127	90.00	8½	4¾

Yarway Fine-Screen Strainers offer better protection against rust, scale and dirt for all steam equipment.

Made in ten standard sizes from ¼ in. to 3 in. Cadmium plated bodies. High grade Monel woven-wire screens. Many thousands in use. Write for Bulletin S-201.

YARWAY EXPANSION JOINTS

All-steel welded construction; light but strong. Chromium covered sliding sleeves.



Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types: Gun-pakt (illustrated) fitted with screw guns which permit addition of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. Also all-brass joints, ¾ in. to 2 in. For additional details send for Bulletin EJ-1912.

The Dole Valve Company

Main Offices and Factory: 1933 Carroll Avenue, **Chicago 12, Ill.**

WATER MIXERS | **THE ALL STAR LINE**

THERMOSTATIC AIR CONTROL | **AIR AND VACUUM VALVES**

"DOLE THERMOSTATIC AIR CONTROL"

FOR FORCED WARM AIR HEATING SYSTEMS

PROVIDES INDIVIDUAL ROOM TEMPERATURE CONTROL

- Operates thermostatically from room air temperature.
- Extremely Sensitive: Modulates output to meet heat requirements.
- Completely self-contained; no wires to run—no bulbs to locate—simple to install. Replaces standard forced warm air registers.

Temperature Selector Dial

- Simple setting of the thermo-dial assures room temperature as desired—corrects many unsatistory heating installations. Materially improves any forced warm air system. An automatic balancer.
- A fully automatic zone control for every room. Dole Air Controls are available in two sizes and will fit the following stackhead openings:

10″ will fit—	12″ will fit—	With an adapter 12″ will fit—
10″ x 4″	12″ x 4″	14″ x 4″
10″ x 5″	12″ x 5″	14″ x 5″
10″ x 6″	12″ x 6″	14″ x 6″

An adapter is available for baseboard installation of these Controls.

Dole No. 20 Fully Automatic Hot Water Air Valve

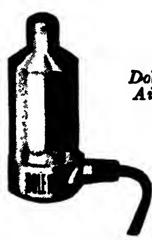
DOLE AIR AND VACUUM VALVES

The Dole line covers every venting need on one pipe steam and hot water heating systems and offers a complete choice for every purpose.

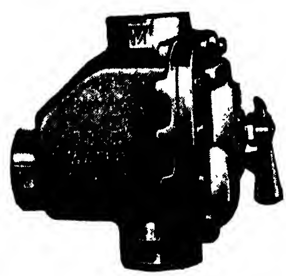
DOLE WATER MIXERS

Dole Water Mixers provide safer, tempered domestic hot water on all tankless heater and storage tank installations. Available in 3 sizes, ½ in., ¾ in., and 1 in.

Dole No. 1A Vari-Vent Air Valve



Dole No. 3 Air Valve



Dole Water Mixer

Jenkins Bros.

80 White Street, New York 13, N. Y.

BRIDGEPORT, CONN.; BOSTON, PHILADELPHIA, CHICAGO, SAN FRANCISCO, ATLANTA

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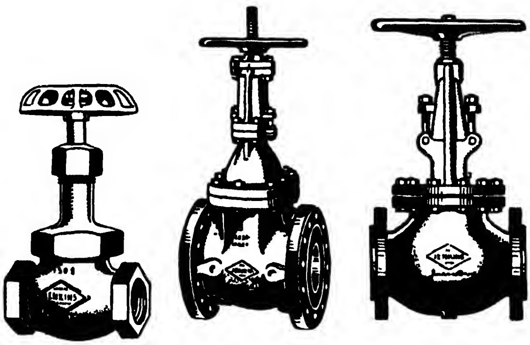
Leading Supply Houses Everywhere Stock Jenkins Valves

Jenkins VALVES for LIFETIME SERVICE



Bronze **Iron** **Steel**

FOR EVERY NEED

JENKINS CATALOG LISTS OVER 500 VALVES

Consult Jenkins Catalog for complete details on more than 500 different valves that cover practically all industrial plumbing and heating, and engineering requirements. Below is a brief list.

All-Iron Valves,—globe, angle, gate; **Angle Valves,**—bronze, steel, and iron body with bronze mounting or trimming.

Blow-Off or **Y Valves,**—bronze and iron.

Electrically Operated Valves, Gates, Globes, Angles, Fire Line Valves; Floor Stands; Foot Valves for gasoline service.

Gate Valves,—bronze, iron, steel; with solid wedge or double disc paralle seats; with removable bonnet and renewable bushing.

Globe Valves,—bronze, iron and steel; one piece and union bonnets; renewable and integral seats; rubber composition or metal discs and plugs.

Horizontal Check Valves,—bronze, iron and steel; **Hose Valves; Indicator Posts; Lock Shield Valves.**

Needle Valves; Non-Return Valves; Quick-Opening; Self-Closing Valves.

Radiator Valves; Rapid Action Valves; Regrinding Valves; bronze and iron body with bronze trimming; renewable plug seats and bevel seats of a special nickel alloy in globe, angle, check and swing check patterns.

Selclo Valves; Stop and Check Valves,—combination or automatic equilizing; Swing Check Valves,—bronze, iron and steel.

Stainless Steel Valves,—globe, angle, gate and check.

Underwriters' Pattern Valves,—check and gate; **Whistle Valves; Waterworks Valves.**

Carey PRODUCTS

The Philip Carey Mfg. Company

Lockland, Cincinnati 15, Ohio

District Offices In All Principal Cities

CAREYDUCT is recommended whereever quietness, ease of installation, fire safety, fume resistance and good appearance are desirable or essential. Widely used in air conditioning systems. Careyduct has proven itself on some of the largest governmental industrial and commercial installations in the country.

Write for engineering performance and installation data.

ACOUSTICAL. Careyduct is a natural sound absorber and non-conductor of sound. Quiets fan noise; won't pick up and "telegraph" other outside noises.

INSULATED. High-efficiency insulation assures delivery of hot or cold conditioned air to outlets with minimum change in temperature.

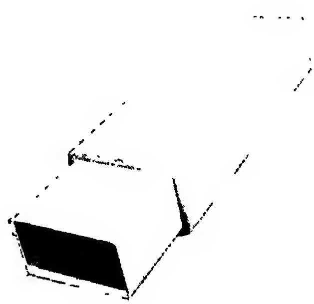
AIRTIGHT. Won't "breathe" or vibrate at high velocities. Slipjoint construction prevents leakage.

SAVES SPACE. Being 40% to 50% quieter than ordinary duct, Careyduct handles higher velocities, permitting the use of smaller sized ducts.

***EASY TO INSTALL.** Prefabricated Careyduct units are easy to install—particularly in tight places. Simple low cost fittings can be made in the shop or on the job.

FIREPROOF. Being 100% asbestos construction Careyduct won't smoulder or burn. Approved by *Underwriters' Laboratories, Inc.*

* *Installation under jurisdiction of International Sheet Metal Workers, A. F. of L.*



GOOD LOOKING. Surfaces are smooth and free from unsightly raised seams or joints. No stiffeners or braces. Blends well with modern interiors.

5 TYPES OF CAREYDUCT

Insulated and Acoustical Type. 100% asbestos construction—combines duct and insulation.

Key-lock Type. For high temperature applications up to 500 F. Impervious to water.

Single-wall Type. For heating and ventilating systems.

Reinforced Corner Type. Fabricated of Carey Firefoil or insulated sheathing. An ideal duct for large industrial installations.

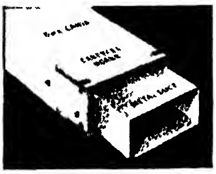
Asbestos-cement Type. Made of asbestos-cement wallboard in different thicknesses. Sizes: 23½ in. and up.

The Philip Carey Mfg. Company

Lockland, Cincinnati 15, Ohio

District Offices In All Principal Cities

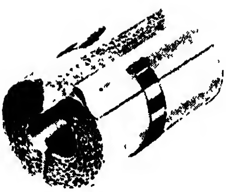
Carey PRODUCTS



CAREYCEL FOR AIR DUCTS

Uses: A fireproof, low cost, high efficiency asbestos board for insulating ducts and all types of air conditioning equipment.

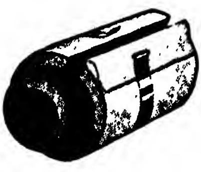
Description: Composed of 12 to 14 laminations of indented (not corrugated) asbestos felt per inch of thickness. Weight: approximately 1½ lb per board foot. Sheet Size: 36 in. x 36 in., or cut to order. Blocks: 6 in. x 36 in. Thickness: ½ in. up.



CAREYCEL FOR HEATING SYSTEMS

Uses: Pipe coverings and blocks for pipes, boilers, ovens and other apparatus where the temperature doesn't exceed 300 F.

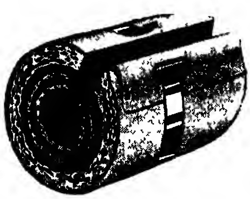
Description: Pipe covering sections 36 in. long by 1 in. thick, finished with cotton duck jacket and bands. Blocks: 6 in. x 36 in. Sheets: 36 in. x 36 in., or cut to order. Thickness: ½ in. up.



CAREY IMPERVO FOR COLD PIPES

Uses: A high efficiency insulation for cold or ice water pipes—keeps the water cold and prevents sweating.

Description: Laminated insulating felt with waterproof liner and jacket. 36 in. long in ½ in., ¾ in., double ½ in. and double ¾ in. thick sections, finished with cotton duck jackets and bands.



CAREY PROTECTO TO PREVENT FREEZING

Uses: Designed especially to reduce the danger of freezing of exposed water pipes.

Description: Consists of two inner layers of hair felt, a waterproof felt liner and an outer layer of insulating felt (wool felt). For severe conditions use two thicknesses. 36 in. long sections with cotton duck jacket and bands. One thickness only—approximately 1½ in.

Grant Wilson, Inc.

141 West Jackson Blvd.
Chicago 4, Illinois

DUX-SULATION

Designed for efficient insulation of metal ducts in Heating, Ventilating and Air Conditioning equipment.

DUX-SULATION is composed of fine, flexible fibres fabricated into a strong felt with millions of dead air spaces, producing a thermal insulation efficiency of 70 per cent (K Factor .27 Btu). Saves 75 per cent of bare duct heat loss.

A heavy asbestos membrane is woven into the felt body below the surface, giving the protection of fire proof asbestos.

The outer surface is heavy woven fabric. A high sound deadening surface—absorbs 70 per cent of air borne noises—with low frictional resistance (F = 0.0001322).

Complete Package:—A flexible insulating blanket ½ in. and 1 in. thick. Comes complete with corner tape and adhesive for cementing on to sheet metal duct work **DUX-SULATION** comes 36 in. wide in a roll containing 100 sq ft. Will not rot, chip or crack.

Surface Temperature of ½ in. DUX-SULATION applied to Metal Duct

Outside Duct Temperature	Room Temperature—Deg F			
	30	50	70	90
40°F	33	48	63	78
60°F	37	53	68	83
80°F	42	57	72	88
100°F	47	62	81	93
120°F	52	68	82	97
150°F	60	75	90	105

Dew Point Temperature—Deg F

Relative Humidity	Room Temperature—Deg F (Dry Bulb)			
	30	50	70	90
20%	0	12	28	44
40%	10	27	45	64
60%	18	37	56	75
80%	25	44	64	84

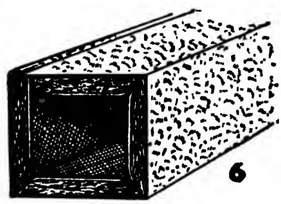
Note: As determined through using the two Tables above, the Surface Temperature of the Dux-Sulation must be HIGHER than the DEW POINT to prevent condensation.

Asbestos Protected DUX-SULATION is applied to round pipes, rectangular ducts or irregular surfaces. It bends in any direction, even to very sharp and abrupt angles. The tape is applied to the corners and joints as illustrated.

Acoustical Values

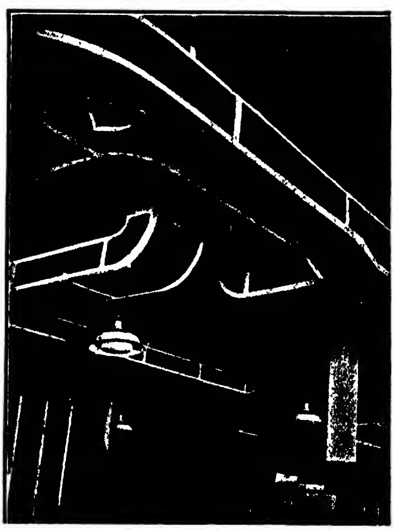
70 Per Cent Reduction in Loudness

Frequency	Feet
1035	9.5
2048	9.7
4100	11.4
517	15.2
259	31.0
129	34.0



6

DUX-SULATION is also applied to the inside of ducts, as illustrated, for high sound absorption efficiency.



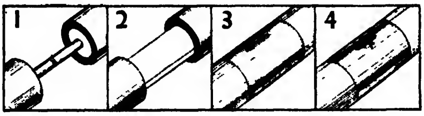
TRADE-MARK
REG. U. S. PATENT OFFICE

Durant Insulated Pipe Company

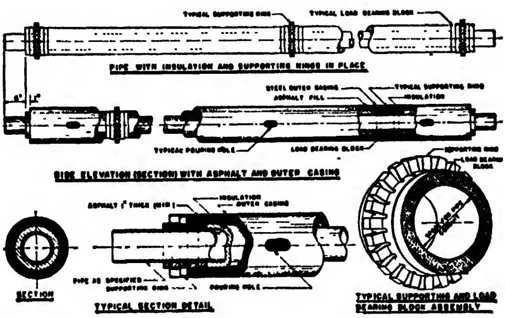
1015 Runnymede Street, P. O. Box 88

Palo Alto, California

4 SIMPLE STEPS IN SEALING FIELD JOINTS



1. Field joint ready for inspection.
2. Joint covered with standard pipe insulation.
3. Special Durant joint casing in place ready for asphalt.
4. Asphalt poured in slot—to seal.



ADVANTAGES

Completely waterproof.
"Elimination" of electrolysis and corrosion.
Requires no sub-drains.
● In multiple lines, individual Durant Pipes can be added, removed or replaced without disturbing others.
Minimum trenching & field work.
No rollers or pipe supports required.
Tile or masonry protection not required.
Lower field costs.
● Insulation protection is absolutely dependable.

CONSTRUCTION OF D.I.P.

The Durant Construction Principle involved consists of encasing the piping, or the insulation covering around the piping, with a heavy layer of High Melting-Point Non-Porous Asphalt. The asphalt is poured hot into a sheet metal form which is spaced concentrically around the pipe or insulation. For all underground conditions the thickness of the asphalt casing is one inch thick minimum for insulated piping. The insulation for all pipes carrying heat should be 85 per cent magnesia or Unibestos and for cold pipes either zero, Unibestos or moulded cork covering should be used. The thickness of the insulation will vary with the conditions of operation. See details in diagram illustration above.

Load bearing supports are provided within D.I.P. structure to prevent weight of pipe from resting on specified insulation. These load bearing supports are full circles slipped on between insulation sections at regular spacings during the fabrication of D.I.P. After covering and load bearing supports have been fitted to pipe, all joints are taped and sealed. A cylindrical jacket of heavy galvanized steel with a diameter large enough to provide a minimum 1 in. air space outside of the insulation, is placed concentrically around insulating pipe. This metal jacket has openings at regular intervals or a full length slot along the top. The insulated pipe is then ready to receive the asphalt protection which consists of a special grade of high-melting point asphalt heated to proper temperature and poured through openings in top of metal jacket.

INSTALLATION OF D.I.P.

Since the insulation and protection are factory-applied to D.I.P., field operations are limited to placing the pipe lengths into position, connecting them and then, after heat pressure tests, insulating and sealing the joints. In underground pipe systems, back-fill can be made at once, and can be flooded with water to pack it. The system is completely waterproof as soon as the joints have been finished.

Foundations are practically never required for underground D.I.P. installations. The pipe needs merely to be placed on the bottom of the graded trench. Only comparatively narrow trenches are required and the individual lengths of D.I.P. can be readily lowered into the trench with rope slings. In the trench the pipe can be turned and slid into proper position and the exposed pipe ends provide plenty of clearance for workmen to connect and finish the joints.

Therm-O-Tile REG. U.S. PAT. OFF.

H. W. Porter & Co.

INCORPORATED

817-G Frelinghuysen Ave., **Newark 5, New Jersey**

Permanent Protection and Insulation for Underground Pipe Lines

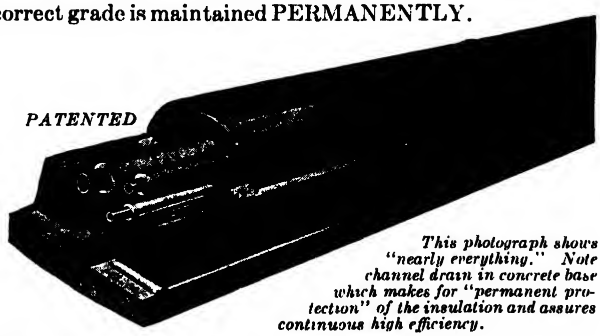
BALTIMORE, MD. CHARLOTTE, N. C. RICHMOND, VA.

Also sold and installed by Johns-Mansville Construction Units in all principal cities.

Therm-O-Tile assures the following:-

1—Much better insulation. A truly DRY conduit. Higher efficiency.
2—Much stronger than required by *A.S.T.M.* Arched construction. Spread-footing foundation.
3—Much longer life. The correct grade is maintained PERMANENTLY.
4—No leakage. Positive sealing throughout.
5—Most helpful installation—on a concrete "sidewalk."
6—No condensation pockets at any time.
7—Lifetime economy.
8—Experienced engineering.
9—A first cost that is competitive, and, definitely lower ultimate cost.

PATENTED

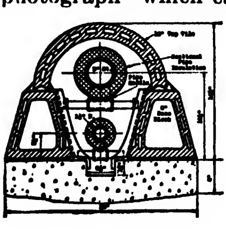


This photograph shows "nearly everything." Note channel drain in concrete base which makes for "permanent protection" of the insulation and assures continuous high efficiency.

THERM-O-TILE STEAM CONDUIT SYSTEMS

"Permanent"—Now that Therm-O-Tile has been on the market so many years, and its design is so well known by all leading heating and ventilating engineers, we again wish to emphasize the importance of "Permanent protection." It is not difficult to provide TEMPORARY protection and insulation for underground pipe lines. Threads and joints don't fail immediately, and foundations don't sag immediately, but unless the job is properly done in the first place it won't be long before water seeps in and ruins the insulation. So, watch out. With wet and spoiled insulation, efficiency drops drastically. Unless the conduit is built on a truly solid foundation there will be sagging and collection of water in pockets.

Thoroughly Drained, Always—In the Therm-O-Tile concrete base there is a drainage channel—clearly visible in photograph—which carries off all water that may enter the conduit from any source, thereby keeping the insulation PERMANENTLY dry. Drainage is entirely internal. The channel is accurately and PERMANENTLY sloped so that condensate or other pockets cannot form. Open to thorough inspection at any time at manholes. Amply large to keep the pipe space dry at all times.



Single or Multiple Pipe Lines Using Sectional Pipe Insulation.

Showing a typical Therm-O-Tile piping arrangement when there are two pipe lines. Note the channel drain which "permanently protects" the insulation.

"Spread Footing" Foundation—Positively prevents settling or sagging. The Therm-O-Tile "Spread Footing" foundation base is a thick concrete slab poured directly in the trench bottom. Steel reinforced or placed on piles when installed over filled or boggy ground to insure PERMANENT protection.

See Previous Issues—In previous issues of this GUIDE we gave details regarding the Tile Envelope which produces 27 different conduit cross sections. We told about the ideal accessibility of this conduit, its great strength, how it is water-proofed, and so on. For complete information ask for a copy of Bulletin 381.

Competitive in Cost—Despite the high efficiency, greater strength, dependability, and other features that are obtained in Therm-O-Tile, it is nevertheless competitive in total first cost. Final cost is much less, thanks to its permanence.

Cooperation—Our engineers have had exceptional experience in designing and installing steam conveying equipment for nearly every purpose.

The Ric-wiL Company

INSULATED PIPE CONDUIT SYSTEMS

Union Commerce Bldg., Cleveland, Ohio • **Agent in Principal Cities**

There is a Ric-wiL insulated conduit system engineered to your specific needs—the transmission of steam, hot water, oil, hot or refrigerated process liquids—providing heat transfer with the lowest possible loss.

1. Ric-WiL Insulated Pipe Unit—Single or Multiple Pipes
Prefabricated complete units with one or more pipes—in any specified combination—in helical corrugated conduit, coated and wrapped with asphalt saturated asbestos felt. 21-ft lengths for easy installation. Insulation is applied to any or all pipes in any thickness specified.

2, Ric-WiL Insulated Pipe Unit—For Process Liquids
An adaptation of the multiple system used where a steam or hot water line heats fluids in other lines. Pipes are insulated from the exterior but not from each other. Sizes and specifications as required—conduit same as for other insulated pipe units.

3. Ric-WiL Foilclad Pipe Units—For Overhead Lines
Pipe and insulation are protected and waterproofed by a double coating of machine-applied, high temperature asphalt. Unit is then wrapped with asbestos felt and covered with a final spiral wrapping of copper or aluminum foil for maximum insulation and protection.

4. Ric-WiL Standard Tile Conduit—Type F
Vitrified glazed *A.S.T.M.* Standard Tile Housing—acid and waterproof—with foundation type base drain supporting weight of piping through correctly engineered pipe supports. Positive locked-in-place cement seals on sides and ends. For single or multiple pipes.

5. Ric-WiL Super Tile Conduit—Type F
Same advantages as Standard Tile but with walls approximately double thick for strength under heavy traffic or where overhead load is above normal. Will support concentrated static load of 6 tons per wheel under actual installation conditions. Base drain of extra-heavy tile.

6. Ric-WiL Cast Iron Conduit—Type F
Heavy reinforced cast iron conduit for use where underground pipe lines run close to or under railroad tracks. Durable, water-tight and vibration-proof. Positive locked-in-place cement seals on sides and ends with metal clamps for extra tightness.

7. Ric-WiL Tile Conduit—Universal Type
Where installation conditions dictate the use of a concrete pad Ric-wiL Universal Tile is recommended. Side walls are double-cell vitrified trapezoidal block design. Arch may be Standard Tile, Super-Tile, or Cast Iron.

Ric-wiL accessories are available in all type systems; standard and special fittings, factory fabricated or field fabricated expansion devices, alignment guides, and anchors. Descriptive bulletins on request. **Write: The Ric-wiL Co., Dept. 279.**

American Structural Products Company

Toledo 1, Ohio

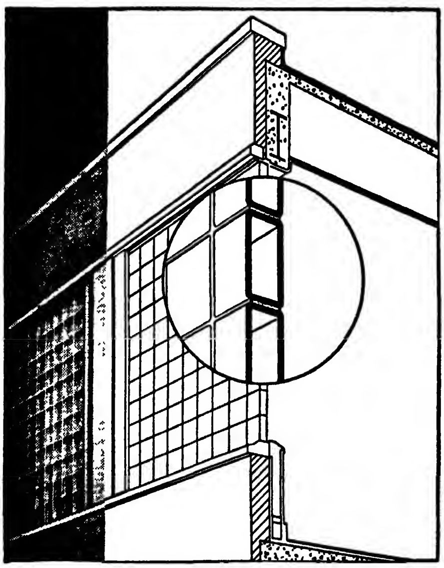
Subsidiary of Owens-Illinois Glass Company

INSULUX®

Glass Block

Insulux Glass Block Give Better Control of Interior Conditions

Insulux Glass Block are hollow, partially evacuated units, $3\frac{7}{8}$ in. thick. Faces are smooth or ribbed. Solid panels of these block, laid in mortar, make a light-transmitting wall of high insulating value. Their proper use aids control of interior conditions to a point where initial and operating cost of heating or cooling equipment is reduced.



Conductivity

The *U* factor for ribbed glass block is 0.46; smooth face, 0.49. For design purposes, these factors may be used as constant for either 6 in., 8 in. or 12 in. block. See Chapter 6 of this volume for additional data.

This *U* factor is only 43 per cent of that of ordinary light-transmitting materials. The reason lies in the two heavy glass surfaces separated by partially evacuated and hermetically sealed dead air space.

Surface Condensation

Because of the low overall air-to-air heat transfer, the room condensation point of a glass block panel is much lower than that of ordinary windows. This will permit higher humidities for both comfort air-conditioning and in industries where high humidity is part of the process. Glass block are not subject to deterioration caused by moisture.

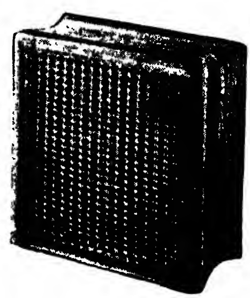
Infiltration

Insulux Glass Block are sealed in the building. They form a barrier against infiltration of dust, air and vapor. Winter drafts and summer vapor leakage are cut. Natural ventilation requirements can be met by installation of windows either inset in the panels or installed directly below the panels.

Solar Heat Gain

A comparative test showed 94 per cent more solar heat through steel sash than through glass block. However, as with sash, glass block transmit less solar heat when properly oriented and shaded. Data in Table 21, Chapter 15, of this GUIDE are for standard block. Other designs, such as the No-glare and Directional block afford further reduction. Data will be sent on request.

Design, Sizes, Erection



Insulux Glass Block is made in 9 face designs for residential or industrial uses. Sizes are: $5\frac{3}{4}$ in. x $5\frac{3}{4}$ in., $7\frac{3}{4}$ in. x $7\frac{3}{4}$ in. and $11\frac{3}{4}$ in. x $11\frac{3}{4}$ in. All are $3\frac{7}{8}$ in. thick. Blocks are erected by laying in mortar like any masonry material. ***Complete technical data, description, etc., sent to you on request.***

American Structural Products Company

Toledo 1, Ohio

Subsidiary of Owens-Illinois Glass Company

KAYLO®

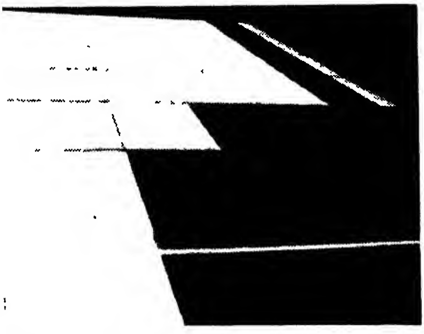
Insulating Products

Kaylo Insulating Roof Tile

Kaylo Insulating Roof Tiles are pre-cast units for construction of non-combustible, structural roof decks on steel or wood framing. Each tile measures 2⅝ in. x 18 in. x 36 in. and weighs only about 23 lbs; yet, Kaylo Roof Decks are more than strong enough to support typical roof loads.

Kaylo Roof Tile is predominantly a calcium silicate, or combination of calcium silicates, plus reinforcing fibers. Tiles are also reinforced with wire mesh. Density: about 20½ pcf.

Kaylo Roof Decks offer an exceptional combination of advantages: light weight that reduces dead load; fire protection; thermal insulation ("U" with built-up roofing, 0.19); little maintenance. Light reflectivity is more than 80 per cent and the underside of a Kaylo deck provides an attractive ceiling without finishing.



KAYLO *Insulating Roof Tile shown on partially completed roof deck. When all tiles are laid, grouting and conventional built-up tar or asphalt roofing are added.*

Kaylo Structural Insulating Block

Kaylo Structural Insulating Block combines with wood, paper, metal or plastics to produce laminated structures. Its many desirable properties include: lightweight (approx. 20½ pcf.); insulation value ("K", 0.65); fireproof; structural strength (average flexural strength, 175 psi).

Block sizes are: widths, 12 and 18 inches; lengths, 18 and 36 inches; thicknesses, 1½ to 2⅝ inches.

FIRE TEST *proves fireproof abilities of Kaylo Structural Insulating Block in door shown (left). Ordinary wood door (right) has already burned through.*

Kaylo Heat Insulating Block

Kaylo Heat Insulating Block is a new type lightweight mineral insulation, efficient from ordinary room temperatures up to 1200 F. One coverage with Kaylo insulation handles applications which often require two thicknesses of other materials.

The "K" of Kaylo Heat Insulating Block (0.41 at 100 mean) gives it high efficiency for medium high temperatures, and experience shows the efficiency improves after exposure to service temperatures.

Kaylo Heat Insulating Block, weighing only approximately 11 pcf., has an average flexural strength of 50 psi and an average compressive strength of 150 psi. This strength means easy handling and application, long service and low maintenance. Blocks can be cut and fitted with ordinary tools.

Standard sizes: 1 to 3 inches thick; 6 and 12 inch widths; 36 inch lengths. Additional information on all Kaylo products available on request.

TYPICAL KAYLO *Heat Insulating Block installation in an apartment house boiler room. This block can be used in hundreds of heat insulating applications.*

American 3 Way-Luxfer Prism Co.

431 S. Dearborn St.,
Chicago-5, Ill.

3-WAY Daylighting Products

26-20 Jackson Ave.,
Long Island City, N. Y.

AMERICAN GLASS BLOCK SKYLIGHTS

Light-Up—the AMERICAN *Way!*

American 3-Way Rooflights make use of Glass Blocks of special design and strength, manufactured by the proven process, incorporating designs on inner surfaces of plates, resulting in uniform even light distribution over wide areas, leaving top and bottom surfaces smooth for easy cleaning.

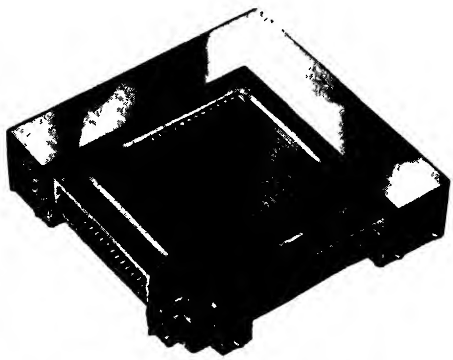
Glass Blocks are 9 in. x 9 in. x 2½ in. and spaced approximately 10⅝ in. on centers.

Low Heat Transfer

Tests conducted by methods suggested by the A.S.H.V.E. Code show that Glass Block Rooflights have about two and one-half times the insulating value of sheet metal skylights with no heat losses by "escape," since the construction is air-tight.

Solar Heat Transmission

Reduction in total solar heat gain as compared with ordinary windows is indicated by relative values given in Table 21 and Table 23 in Chapter 15.



Section of American Glass Block Skylight Showing Method of Block Application

American Glass Block Skylight

Triple Plates of Glass

Magnalite Diffusing Glass units may be attached to under side of glass block construction thus making for effective uniform light diffusion and even distribution; also very effective in condensation problems.

Insulated Construction

Construction of rigid reinforced concrete grids can be arranged with insulation materials sufficient to approximately equal the performance of the glass blocks.

Condensation

Due to the nature of the grid construction where insulation materials are employed with semi-vacuum glass blocks assemblies there is little or no tendency for condensation to form on the under side. Should relative high humidities or abnormal conditions exist, further insulation treatment can be provided.

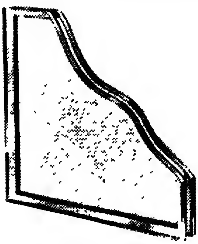
Glass Block Assemblies for all off-vertical arrangements are available. Details will be furnished on request.

3-Way Glass Block Skylights may also be furnished without special insulation treatment in reinforced concrete grid construction.

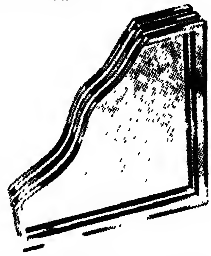
Write for Complete Information

Libbey • Owens • Ford Glass Company

Room 1038 HVA, Nicholas Building, Toledo 3, Ohio



THERMOPANE*
Insulating Glass



L-O-F Thermopane is a transparent factory-fabricated insulating glass unit composed of two or more panes of glass separated by ¼ in. or ½ in. of dehydrated captive air, hermetically sealed at the edges in the factory with a metal-to-glass bond.

Thermopane is made in more than 70 standard sizes, all of which can be used vertically or horizontally.

Thermopane Reduces the coefficient of heat transmission; increases the room-side surface temperature, thus promoting radiant comfort and lowering the dew point; furnishes a control of light quantity and quality through combinations of various types of glass, and deadens sound transmission to some degree.

Thermopane Uses are many, but may be briefly summarized as below:

Double-glass Thermopane: Glazing of wood or metal windows, doors and window walls for practically any purpose in structures requiring heating or air conditioning.

Triple-glass Thermopane: Large stationary units such as insulated glass walls in homes, apartments, public and commercial buildings and in show windows for refrigerated display where temperature differential must be considered.

Quadruple-glass Thermopane: Engineered to meet low temperature and high humidity conditions.

Thermopane Units provide a high resistance to heat flow, varying with the number of panes and the thickness of the air space used. In summer the low heat transmission coefficient reduces the load on air conditioning systems. In winter it saves heat. The greater efficiency of Thermopane makes it possible to incorporate larger windows in houses and keep the cost of fuel constant. For example, a house could have 115 sq ft of Thermopane instead of 58 sq ft of single glazing and not lose any more heat. Thermopane permits the influx of solar heat in exterior glazing of buildings without a prohibitive compensating loss from conduction.

The over-all heat transmission coefficient U varies with the ranges of temperature at which the coefficient is determined. For most practical heat loss calculations coefficient U can be the value determined at 10° outside temperature, 70° inside temperature, 15 mph outside air velocity, 0.25 mph average inside air velocity. The following table gives such values from actual tests with glass sizes 30 in. x 30 in.

Number of Panes	Glass Thickness	U Value		
		Air Space		
		None	¼"	½"
Single Glass	⅛" ¼"	1.16 1.15		
Double Thermopane (one air space)	⅛" or ¼"		.65	.58
Triple Thermopane (two air spaces)	⅛" or ¼"		.47	

The Room-Side Surface Temperature of Thermopane is considerably higher than that of single glazing. Usually radiators or registers are near glass areas in buildings to offset conducted heat loss from a room and radiant loss from the bodies of persons near cold glass areas. The higher glass surface temperature of Thermopane greatly reduces the amount of heat which must be supplied, permitting more flexibility in room design.

Another Important Benefit from Thermopane is the prevention of frost or condensation from forming on the room-side surface of a single pane of glass in winter due to higher room humidity. The absence of condensation on the room-side surface of glass is of considerable importance where clear visibility is a factor as in residences, all types of commercial or industrial buildings and refrigerated display spaces.

More Complete Information on Thermopane is available by writing to Libbey-Owens-Ford, or its district office nearest to you, and requesting technical data sheets prepared by Don Graf, a general booklet about the product, and a brochure which discusses Solar Housing.

* Reg. U. S. Off. Pat.

Pittsburgh Corning Corporation

Room 505–8, 632 Duquesne Way Pittsburgh 22, Pa.

PC FOAMGLAS INSULATION

When installed according to our specifications for recommended applications, PC Foamglas retains its original insulating efficiency *permanently*.

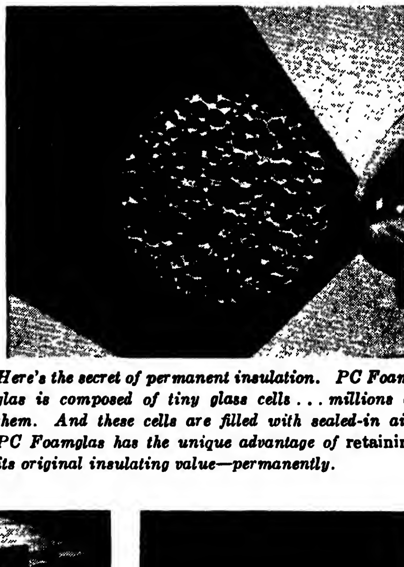
PC Foamglas is a cellular glass insulating material having unique characteristics. It is not a fiber, not a wool, not a board nor a batt. Consists of millions of glass enclosed air cells, in the form of big, rigid, light-weight blocks.

Being glass, PC Foamglas is highly resistant to the fumes, acid atmospheres, vapors and other elements that so often cause deterioration of other insulating materials.

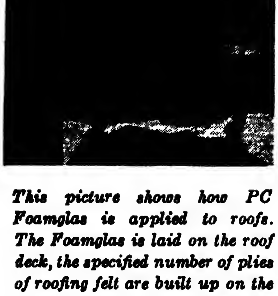
PC Foamglas has proved its ability to help maintain desired temperature levels and to minimize condensation—without maintenance, repairs or replacement—resulting in worthwhile economics. It is widely used on ducts and other industrial equipment, as well as in corewalls, on roofs and floors. As a pipe insulation, PC Foamglas can be used to insulate both hot and cold piping, indoors and outdoors. Note the illustrations of some typical applications.

When you are figuring on insulation, for any purpose, make sure that you have the latest information on PC Foamglas. Send for our free booklets. They tell you what you want to know. Pittsburgh Corning Corporation also makes PC Glass Blocks.

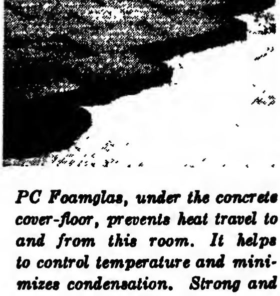
When you insulate with **PC Foamglas** you insulate for good



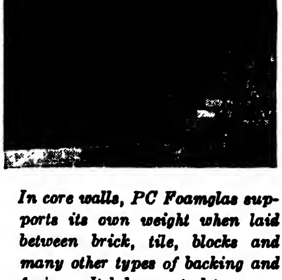
Here's the secret of permanent insulation. PC Foamglas is composed of tiny glass cells . . . millions of them. And these cells are filled with sealed-in air. PC Foamglas has the unique advantage of retaining *its original insulating value—permanently.*



This picture shows how PC Foamglas is applied to roofs. The Foamglas is laid on the roof deck, the specified number of plies of roofing felt are built up on the firm base which is provided by big solid blocks of PC Foamglas.



PC Foamglas, under the concrete cover-floor, prevents heat travel to and from this room. It helps to control temperature and minimizes condensation. Strong and rigid, PC Foamglas will support heavier than normal floor loads.



In core walls, PC Foamglas supports its own weight when laid between brick, tile, blocks and many other types of backing and facing. It helps control temperature and minimize condensation. It will not pack, rot or check.

Pittsburgh Corning Corporation

Room 704–8, 632 Duquesne Way, **Pittsburgh 22, Pa.**

PC GLASS BLOCKS

Distribution through Pittsburgh Plate Glass Company Warehouses in principal cities and by the W. P. Fuller Company on the Pacific Coast and by Hobbs Glass Ltd. in Canada. Also makers of PC Foamglas Insulation.

Glass Blocks allow the economical use of large glass areas, reduce heat loss in cold weather and materially aid air-conditioning. This is because each PC Glass Block contains a sealed-in dead-air space that is an effective retardant to heat transfer.

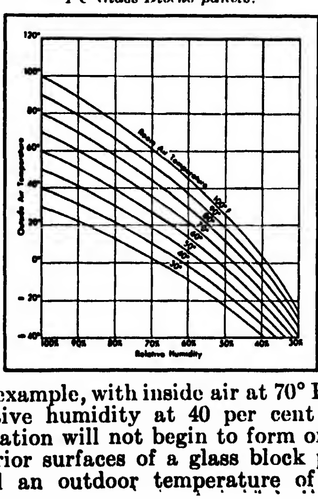
Thermal Insulation

Tests run by nationally recognized laboratories have established the value of glass blocks for insulation of light-transmitting areas. These tests have proved that with glass block panels, heat loss is slightly less than half that experienced with single-glazed windows. In computing heat losses through panels for most design purposes, it is recommended that a "U" value of 0.46 to 0.49 be used for all block sizes and face patterns. For complete data on heat transfer values see the section on heat transfer elsewhere in this Guide—page 142.

Surface Condensation

Due to high insulating value, condensation will not start forming on the room side of glass block panels until outside air has reached a temperature much lower than that necessary to produce condensation on single-glazed windows. The accompanying chart shows at what temperatures condensation will form.

Outdoor temperature required to produce Condensation on the room side surface of PC Glass Blocks panels.



For example, with inside air at 70° F and relative humidity at 40 per cent condensation will not begin to form on the interior surfaces of a glass block panel until an outdoor temperature of −1 deg is reached. Under similar conditions with single-glazed sash, moisture will begin to form when the outdoor temperature reaches +33°F.

Solar Heat Gain

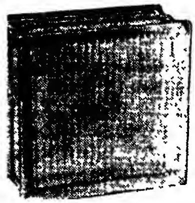
The use of glass blocks for light-transmitting areas results in a marked reduction in total solar heat gain as compared with ordinary windows. This factor is of considerable advantage in buildings that are properly air-conditioned, but does not eliminate the need for adequate ventilation or shading in non-air-conditioned rooms.

For data on solar heat gain through glass blocks see table 21 in the solar radiation section of this Guide—chapter 15. This table is for standard pattern glass blocks.

PC Glass Blocks Aid Air-Conditioning

Two of the chief aims of air-conditioning—temperature control and cleansing of air are aided by the use of PC Glass Blocks. Heat loss is less in winter—heat gain is less in summer. Ideal conditions are much more easily maintained without undue condensation. Solar heat transmission and radiation are reduced. Neither dirt nor drafts can filter in, for each panel is a tightly sealed unit.

Sizes and Shapes Available



PC Glass Blocks are available in many attractive patterns, some of which are designed for special control and direction of transmitted daylight. For complete information on the sizes and shapes of PC Glass Blocks, and for illustrations of the many patterns available, write the Pittsburgh Corning Corporation, Pittsburgh, Pa., or call the nearest Pittsburgh Plate Glass Company warehouse.

• • •

Additional technical data, including detailed figures on thermal insulation, solar heat gain, surface condensation, light transmission and construction data, will be furnished on request.

Armstrong Cork Company

Building Materials Division

Lancaster **Pennsylvania**

Offices

Albany	Cincinnati	Houston	Milwaukee	Providence
Atlanta	Cleveland	Indianapolis	Minneapolis	Richmond
Baltimore	Columbus	Jacksonville	New Orleans	Rochester
Birmingham	Dallas	Kansas City	New York	St. Louis
Boston	Denver	Los Angeles	Omaha	Salt Lake City
Buffalo	Detroit	Louisville	Philadelphia	San Francisco
Charlotte	Hartford	Memphis	Pittsburgh	Tulsa
Chicago				Washington, D. C.

Distributors

Charleston 23, W. Va.. Capital City Supply Co.
Eau Claire, Wis .... Horel-George Co.
El Paso, Texas .Case Industrial Service Company
Fort Wayne, Ind.....Midland Engineering Co.
Grand Rapids, Mich.
Tony Batenburg Insulation Co.
Jamestown, N. Y....Laco Roofing & Asbestos Co.
Joplin, Mo... .... Joplin Cement Co.
Little Rock, Ark...Fischer Cement & Roofing Co.
Manitowoc, Wis.
Northwestern Asbestos and Cork Insulation Co.
Portland 4, Ore.....Asbestos Supply Co. of Oregon
San Antonio, Tex.....General Supply Co., Inc.
Seattle 4, Wash....Asbestos Supply Co. of Seattle
South Bend 23, Ind.....Midland Engineering Co.
Spokane 12, Wash.
Asbestos Supply Co. of Spokane
Springfield, Mo......Southwestern Insulation Co.
Tacoma 2, Wash....Asbestos Supply Co. of Tacoma

For detailed technical information, samples, and descriptive literature, ask any office or distributor. Specifications appear in Sweet's Catalogs for Architects, Engineers, and Power Plants.

PRODUCTS—Armstrong's Corkboard, Cork Covering, Mineral Wool Board—Foamglas*, Heat Insulations, Armstrong's Insulating Refractories, Cushiontone®, Temlok®, Insulation Sundries.

Corkboard

The thermal conducitivity of Armstrong's Corkboard is 0.27 Btu per hour, per degree temperature difference, per inch thickness at 60°F mean temperature. It is furnished in rigid boards 12 in. x 36 in., 18 in. x 36 in., 24 in. x 36 in., and 36 in. x 36 in., in 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in. thicknesses. **Armstrong's Corkboard conforms in all details to Federal Specification HH-C-561b.**

Cork Covering

Armstrong's Cork Covering is made of pure cork in sizes to fit all standard pipe sizes. The inside surfaces of each piece are machined to assure an accurate fit, free from moisture-catching air pockets. Cork covering is rigid and will not sag. Thicknesses are: Light Duty (1.20 in. to 1.93 in.); Standard (1.70 in. to 3.00 in.); and Heavy Duty (2.63 in. to 4.00 in.).

Armstrong's Fitting Covers are rigid and are designed to fit accurately all types of standard ammonia and extra heavy fittings, screwed, flanged, and welded.

Mineral Wool Board

Armstrong's Mineral Wool Board equals or exceeds Federal Specification HH-M-371a for board or block form insulation; has low thermal conductivity; is moisture resistant, odorless; is easily handled and erected; possesses structural strength. Standard size 12 in. x 36 in.; thicknesses 1 in., 1½ in., 2 in., 3 in., 4 in.

Foamglas

Foamglas has a closed cellular structure which will not permit passage of air or moisture. It is efficient, moisture-proof, fireproof, and offers effective, lasting insulation. This type of insulation is made in standard 12 in. x 18 in. blocks; thicknesses 2 in., 2½ in., 3 in., 4 in., and 5 in. It may be used to insulate all types of low-temperature storage rooms.

Heat Insulation

The Armstrong Cork Co. distributes and offers contract service on a complete line of high temperature insulations. Included are: 85 per cent magnesia block and pipe covering; high temperature block and pipe covering; air cell block, sheet, and pipe covering; wool felt; hair felt; etc.

Engineering and Contract Service

All Armstrong offices and distributors maintain skilled erection crews. For aid in the solution of any technical problems involving insulation or acoustical treatment, and for literature and prices, get in touch with an Armstrong district office or distributor or the Armstrong Cork Co., Building Materials Division, Lancaster, Pennsylvania.

*T. M. Reg. U. S. Pat. Off., Pittsburgh Corning Corp.

Baldwin-Hill Co.

549 Breunig Avenue, **Trenton 2, N. J.**

Plants in

TRENTON, N. J. KALAMAZOO, MICH. HUNTINGTON, IND.

B-H No. 1 INSULATING CEMENT

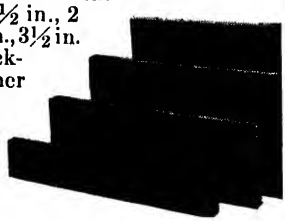
A plastic insulation produced from high-temperature-resisting, nodulated B-H *black* Rockwool combined with high-grade, long-fibre asbestos and colloidal clay. Effective up to 1800° F; reclaimable up to 1200° F. Suitable for insulating large or small irregular surfaces, including those not suited to molded types of insulation. Contains a special *rust inhibitor* which prevents corrosion taking place between insulated surface and cement. Makes a secure bond on either hot or cold surfaces. Instantly adhesive, easy and economical to apply. When mixed with water to trowelable consistency, the nodules of B-H *black* Rockwool retain their physical properties; when dried out, the dead air cells in these nodules provide maximum insulating efficiency. Packed in strong 50-lb bags; can be stocked without breakage or loss.

B-H KOLDBOARD

Effective from −150 to 300 F. Made from 100 per cent chemically stable B-H Rockwool fibres, felted and bonded together to form flat, semi-rigid blocks which do not disintegrate or break down structurally under severe service conditions. Koldboard will not support combustion nor smolder when flame is applied. Low moisture absorption: 0.68 per cent at relative humidity of 65 per cent at temperature of 75 F. Thermal conductivity: 0.29 Btu/sq ft/hr/F at a mean temperature of 70 F. Standard sizes: 18 in. and 36 in. long; 6 in., 12 in., 18 in., 24 in. wide; ½ in. to 4 in. thick; packed in high test fibre cartons.

B-H MONO-BLOCK

A one-block insulation effective over the full temperature range up to 1700 F. Fabricated of high-temperature-resisting B-H *black* Rockwool, felted by a special patented process. A strong, lightweight block, easily cut and fitted on irregular or flat surfaces. Low alkalinity factor insures stability under severe temperature conditions. Density—approx. 20 lb per cu. ft. Standard sizes: 18 in. and 36 in. long; 6 in., 12 in. and 18 in. wide; 1 in., 1½ in., 2 in., 2½ in., 3 in., 3½ in. and 4 in. thicknesses. Other sizes to order. Packed in high-test cardboard cartons.

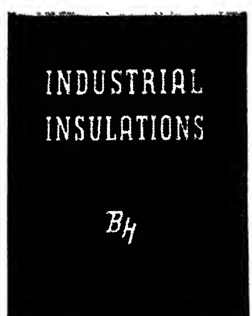


B-H DUCT SOUNDLINER

A rigid material felted of chemically stable Rockwool fibres. Provides an effective means of minimizing sound transmitted through the ducts, by installing it inside the ducts, also provides thermal insulation. Does not disintegrate nor support combustion. Easily cut and fitted around bends in the duct—attached with either bolts or specially prepared acoustical cement. Standard sizes: 24 in. x 36 in.; ½ in. or 1 in. thickness.

CATALOG ON REQUEST

We invite you to write for fully illustrated catalog giving more detailed specifications on these and other B-H Industrial Insulation Products of many types.



Bushings, Inc.

Coolidge at 14 Mile Road, **Royal Oak, Mich.**

Catalog No. Bu. 50

VIBRO-LEVELERS: Rubber-in-shear vibration dampner and precision leveler stop vibration and level machinery. Recommended for New or Existing: Air Conditioning Units, Refrigeration Units, Compressors, Blowers, Diamond Borers, Buffers-Polishers, Punch Presses, Broaching Machines, Vacuum Pumps, Forging Hammers, Tool Room Grinding Machines, etc.

Mouting Number	Load Capacity of Each Mounting (Pounds)
3010	10
3025	25
3050	50
3100	100
3200	200
3300	300
3400	400
3500	500
3750	750
4000	1000
4150	1500
4200	2000

Usually installed 4 per machine. Any number may be used to suit machinery requiring more (or fewer) points of suspension.

VIBRO-LEVELERS Have These 6 Important Features: (1) Rubber-in-shear Vibration Dampner *and* Precision Leveler; (2) Ease of Installation; (3) No special parts required; (4) No tapping or fitting (Vibro-Levelers come complete); (5) No floor cutting (and floor repair); (6) Low Cost.

Vibro-Leveler (Dual Purpose Machinery Mountings that stop vibration transmission and level machines) are as simple in design and construction as they are easy to install.

An inner cylinder with the stud, is insulated from the outer shell by a wall of mechanically bonded rubber. They have an ample safety margin. Overloaded to destruction they will allow a drop of less than ½ in.

Installation

Vibro-Levelers are usually installed under the machine. The machine is either raised or tilted enough to let the Vibro-Leveler be slipped under the base and to allow the stud to be inserted into the hold-down bolt hole in the base of the machine.

The lower nut is turned to bring the machine to exact level and the upper nut locks it in position. The Vibro-Levelers come complete even to the leveling and locking nuts; no fitting, no cutting, no threading of any kind is required.

Where the machine must be kept at or near the original level, brackets are either bolted or welded to the side of the base. Ordinary structural brackets are ideal for this.

The Celotex Corporation

General Offices

120 South LaSalle Street, **Chicago 3**

CELOTEX
REG. U. S. PAT. OFF.

Celotex Cane Fibre Insulation products are made by felting the long, tough fibres of bagasse into strong, rigid boards. They are manufactured under the Ferox Process (patented) which effectively protects them from destruction by termites, fungus growth, and dry rot. They are integrally water-proofed which insures a non-hygroscopic insulation of low capillarity and enduring insulating efficiency.

Celotex Insulating Sheathing

An insulating, weather-resisting sheathing for use under any type of exterior. Surfaces and edges are moisture-proofed with a surface impregnation of asphalt.

Sizes: $\frac{25}{32}$ in. thick: 4 ft wide: 8 ft, 9 ft, 10 ft and 12 ft long. Also 2 ft x 8 ft with long edges—center matched.

Celotex Insulating Lath

Regular Insulating Lath.—A cane fibre plaster base of high insulating efficiency. Surface provides a strong bond for plaster and the bevelled edges and ship-lap joint provide additional reinforcement.

Size: 18 in. x 48 in.; thickness: ½ in.

Celotex Roof Insulation Products

Regular Roof Insulation—A cane fibre product possessing superior insulating properties. It reduces roof heat transmission as shown by coefficients established in THE GUIDE; reduces roof movement due to contraction and expansion.

Size: 23 in. x 47 in. or 24 in. x 48 in.; thickness: ½ in., 1 in., 1½ in. and 2 in.

Preseal Roof Insulation—A cane fibre product coated with a special asphalt for moisture protection on the job. Size: 23 in. x 47 in., or 24 in. x 48 in.; thickness: ½ in., 1 in., 1½ in. and 2 in.

Vapor-seal Roof Insulation—An improved type of water resistant cane fibreboard coated with high grade asphalt and having offsets on all edges so that when applied a network of channels next to the roof deck provides a means of equalizing the air pressure therein. Size: 24 in. x 48 in.; thickness: 1 in., 1½ in. and 2 in.

Cemesto

A completely fabricated fire and moisture resistant structural insulating wall unit. Consists of a Celotex cane fiber core surfaced on both sides with a ⅛ in. layer of asbestos-cement board. The established low thermal conductivity of the Celotex core is maintained in the manufacture of Cemesto.

Sizes: 4 ft x 4 ft, 4 ft x 6 ft, 4 ft x 8 ft, 4 ft x 10 ft, 4 ft x 12 ft; thicknesses: 1⅛ in., 1$\frac{9}{16}$ in. and 2 in.

Celo-Block

Celo-Block Cold Storage Insulation consists of ½" low density cane fibreboards bonded together with asphalt mastic and surfaced front and back with asphalt. Celo-Block has excellent insulating qualities. Size: 12 in. x 36 in. or 18 in. x 36 in.; thickness: 2 in. and 3 in.

Celotex Rock Wool Products

Available in the following forms—Loose, Granulated, and Paper-backed Batts. Celotex Rock Wool is made from the clean fibres of molten rock. Incombustible and integrally waterproofed.

Q-T Ductliner

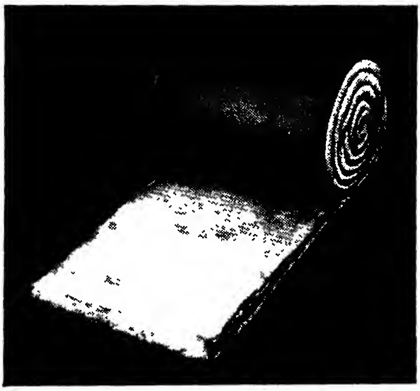
An acoustical material designed especially for duct lining in air conditioning systems. Absorbs duct noises. Made of rock wool and a special binder. Designed to withstand air duct humidity conditions. Is fire resistant and will not smoulder or support combustion. Thermal conductivity of 0.30.

Crawford-Austin Mfg. Co.,

6th & Jackson St., P. O. Box 209
Waco, Texas

FLAMEPROOFED—COTTON INSULATION

HIGHLY EFFICIENT



EFFICIENCY

The low thermal conductivity of Heat Stopper Insulation makes it ideally suited for all insulation requirements. **K factor: 0.24** Btu/hr/sq ft/deg F/in. (See table.)

EASY TO INSTALL

Requires a minimum of labor and expense for installation. Unrolls like a rug. Complete flexibility insures uniform distribution in all odd shaped spaces. An easy one man job.

LIGHTWEIGHT

The fluffy lightness of Heat Stopper Insulation gives maximum protection with minimum load factor. **Density:** ⅞ lb/cu ft.

NON-ABSORBENT

The natural wax-like coating on each of the cotton fibres, along with the aid of chemical treatment, makes Heat Stopper Insulation resistant to moisture.

FLAMEPROOF

Treated with a chemical formula for permanent fire-resistance. Effectively withstands 1800 deg blow torch test.

NON-IRRITATING TO THE SKIN

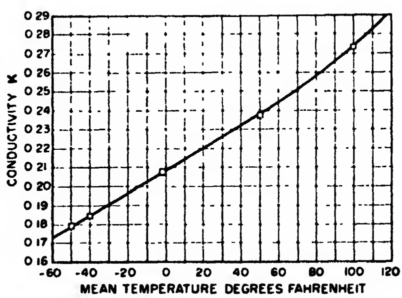
Heat Stopper Flameproof Cotton Insulation contains no splinter or needle like points to injure skin or breathing passages. Always clean and fresh, *has no odor*.

WILL NOT SETTLE OR PACK

Because of its natural resiliency, Heat Stopper Insulation will not pack down from vibration or age. If ever compressed for any reason, Heat Stopper quickly regains original thickness or better when exposed to air.

REPELS HOUSEHOLD PESTS

Heat Stopper Insulation will not harbor such household pests as insects, rats, mice, etc. Chemicals used in the treatment of Heat Stopper Insulation, although harmless to humans, is repulsive to household pests.



THERMAL CONDUCTIVITY OF 1" THICK HEAT-STOPPER INSULATION AT DENSITY OF 0.95 LBS. PER CUBIC FOOT

Heat Stopper Flameproof Cotton Insulation is available in a wide variety of widths and thicknesses to fit standard building requirements.

The Eagle-Picher Company

General Offices: American Building, **Cincinnati 1, Ohio**

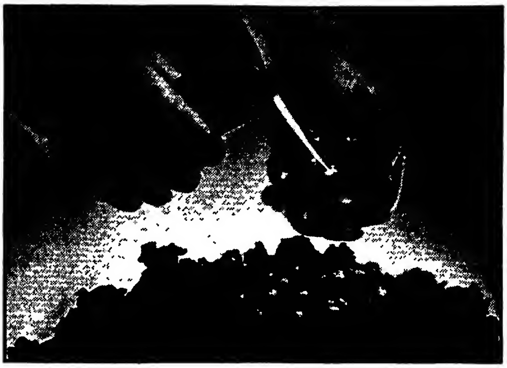
Offices in Principal Cities

EAGLE-PICHER

A Remarkable Insulating Wool Made From Minerals

Years ago Eagle-Picher pioneered a method of fusing and fiberizing carefully selected minerals into a dark gray insulating wool. This mineral wool is chemically inert. Fibers are mechanically strong, extremely resilient and flexible. They withstand expansion and contraction without loss of efficiency even at elevated temperatures.

From this mineral wool, Eagle-Picher has fabricated a long list of insulating products to meet a wide range of temperatures and operating requirements.



H-2 Loose Wool

A clean fill insulation that is highly efficient for temperatures to 1200°F. Averages considerably lighter in weight than many rock and slag wools—goes farther. Fibers are soft and flexible. Eagle-Picher Insulation is as fireproof as the minerals from which it is made. Retains physical and chemical stability in presence of water. Packed in 40-lb bags.

7-B Granulated Wool

Another grade of fill insulation that has all the advantageous properties of Eagle H-2 Loose Wool. It consists of small pellets averaging 1/8 to 1/2 in. in size. For all fill jobs in irregular spaces. May be poured. Packed in 40-lb bags.

Low Temperature Felt

A highly efficient insulating material for subzero and low temperatures (to 400°F). Available in densities of 4, 6 and 8 lb per cu ft. Recommended for refrigerator rooms, trucks, refrigerators, stoves, etc. Sheds water. Extensively used in marine field.

Paper Backed Batts and Blankets

These light-weight, sturdily constructed batts and blankets are easy to apply. One side is protected with asphalt coated paper which serves as an approved vapor barrier. Strong tacking flanges. Quickly cut with knife or shears. Two thicknesses—Full-Thick and Semi-Thick. For home use.

Super "66" Cement

A high-temperature insulating cement. Easy to apply and trowels to a smooth finish. Actively inhibits rust. Will stick on any clean, heated surface. Dry coverage approximately 50 sq ft per 100 lb. 100 per cent reclaimable up to 1200°F. Packed in 50-lb bags.

Supertemp Blocks

An all-purpose high-temperature block insulation which will withstand elevated temperatures up to 1700°F without loss of efficiency or structural strength. Fibers are water-repellent. Light weight. Easily cut to fit irregularly shaped surfaces. Blocks withstand all normal vibration and abrasion encountered in use for which they are recommended. All standard sizes.

Insulseal

A protective coating for Industrial Insulation Blankets, Supertemp, "66" Cement and other kinds of heat insulation. Provides a permanent seal that safeguards insulation against air infiltration, moisture, water, fumes; also against vibration and abrasion. Does not support combustion.

For more complete specifications and technical data on these and other Eagle Insulating Products, see Sweet's Engineering or Power Plant catalogs.

INSULITE

General Offices
500 Baker Arcade Bldg., **Minneapolis 2, Minnesota**

THIRTY-FIVE YEARS PROVEN DURABILITY

For 35 years engineers and architects have specified Insulite materials for structural uses, interior finish, low temperature duct lining, and for other thermal insulation and sound control work. Insulite materials have proved themselves practical through their performance on the job.

STRUCTURAL MATERIALS

Sealed Graylite Lok-Joint Lath—An insulating plaster base of Graylite, sealed on stud space side with an effective *moisture vapor barrier*. Has patented "Lok" on long edges.
Thickness: ½ in.
Size: 18 x 48 in.

Bildrite Sheathing is an asphalt-containing wood fiber insulating board manufactured under an exclusive process which provides increased strength and moisture resistance. It is $\frac{25}{32}$ in. thick and has a gray-brown color. Thermal conductivity maximum: 0.36 Btu per inch thickness. Each sheet is marked to indicate proper nail spacing. Available in sizes 4 x 8 ft up to 4 x 12 ft with all edges square. Also available in 2 x 8 ft size with interlocking joint on long edges. Used as a structural sheathing board and as roof boarding.

Condensation Control—Where low outside temperatures and high inside humidities may occur, authorities recommend "sealing the warm side and venting the cold side" of the wall to prevent condensation. An adequate vapor barrier, Sealed Graylite Lok-Joint Lath should be used on the warm (room) side of the wall thereby effectively reducing vapor transmission into the stud space. Bildrite Sheathing is designed to allow any surplus vapor in the stud space to "breathe" or be vented to the exterior air. If vapor is trapped within the stud space and cannot escape through the sheathing, destructive condensation may occur.

THE APPROVED INSULITE WALL OF PROTECTION

This construction consits of Bildrite Sheathing on the exterior of the frame work and Sealed Graylite Lok-Joint Lath on the interior. Transmission coefficient (U) is shown below.

Exterior Finish and Sheathing	Interior Finish — No Insulation Between Studding — Plaster (½ in.) on Sealed Graylite Lok-Joint Lath (½ in.)
Wood Siding, 25/32 in. Bildrite Sheathing	0.15 Btu/sq ft./hr/°F

The above value is typical of results which can be obtained by utilizing Insulite materials in frame construction. For further (U) values refer to Chapter 6 pages 130 and 131.

Applying Bildrite Sheathing

Applying Lok-Joint Lath

INTERIOR FINISH MATERIALS

Graylite Building Board—An integrally treated asphalt containing wood fiber board of grayish brown color—burlap and linen textured surfaces. Thermal conductivity nominal 0.35 Btu per inch thickness. Furnished in thickness of ½ inch and sizes of 4 x 6 ft to 4 x 12 ft.

Smoothcote Interior Board—Factory coated Insulating Board with smooth, finished surface one side, having 68 per cent light reflection. Furnished in ½ inch thickness only and in sizes of 4 x 6 ft to 4 x 12 ft.

Satincote Interior Board--Factory finished Insulating Board in colors light ivory and oyster white. Light reflection 80 per cent for the light ivory and 72 for the oyster white. Requires no further decoration. Resistant to abrasion and washable. In ½ inch thickness and in sizes of 4 x 6 ft to 4 x 12 ft.

Tileboard—Available in Satincote. TileBoard is furnished with the Lok-Grip Joint that permits concealed nailing and which together with the Lok-Pin (a flat diamond shaped metal dowel) definitely and mechanically safeguards against any falling units even though no face nailing is used.

Satincote TileBoard available in ½ inch thickness and sizes of 12 x 12 inches to 16 x 32 inches.

Plank—Available in Satincote. Plank has the Lok-Grip Joint which permits concealed nailing and is beveled and beaded on both long edges. Satincote Plank furnished in ½ inch thickness, widths of 8 to 16 inches and lengths of 8 to 12 ft.

Acoustilite—A high efficiency acoustical material for sound control. Coefficient of sound absorption, at 512 cycles, is 0.79 when mounted on solid background and 0.80 when on furring strips. Noise reduction coefficient is 0.65 when mounted on solid background and 0.75 when on furring strips. Factory painted in buff, (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ¾ in.; sizes, 12 x 12 in. to 16 x 32 in.

Acoustilite or Fiberlite effectively quiet and control sound

Fiberlite—An efficient sound absorptive and decorative material. Coefficient of sound absorption, at 512 cycles, is 0.53 when mounted on a solid background and 0.72 when on furring strips. Noise reduction coefficient is 0.55 when mounted on solid background and 0.65 when on furring strips. Factory painted in buff (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ½ in.; sizes, 12 x 12 in. to 16 x 32 in.

HardBoard Products

HardBoard materials are tough, durable, grainless, pressed wood fiber boards with a hard, smooth surface. Available in a range of densities from 55 to 68 lb/cu ft. Thicknesses are from 1/10 to 5⁄16 in. and sizes of 4 x 2 ft to 4 x 12 ft.

Industrial Insulation

Industrial Insulation is a wood fiber board for use in all types of manufacturing industries producing items such as refrigerators, coolers, showcases, brooders, partitions and cabinets.

It can be cut-to-size and fabricated to customer's specifications. Three types of industrial board are available.

Lowdensite Industrial Board—A 10 to 14 lb density board with an average tensile strength of 100 lb/sq in. and an average conductivity of 0.30 Btu/hour /sq ft/F/inch thickness.

Ins-Lite Industrial Board—A 14 to 18 lb density board with an average tensile strength of 250 lb/sq in. and an average conductivity of 0.33 Btu/hour/sq ft/F/inch thickness.

Graylite Industrial Board—Differs from two above products in that it has an integral asphalt treatment which provides increased strength and moisture resistance as well as minimum thickness and linear expansion. A 16 to 20 lb density board with an average tensile strength of 350 lb/sq in. and an average conductivity of 0.35 Btu/hour/sq ft/F/inch thickness.

Johns-Manville

Executive Offices: 22 East 40th Street, **New York 16, N. Y.**

Offices in All Large Cities

Johns-Manville Home Insulation

Applying J-M Super-Felt batts in new home

Johns-Manville Rock Wool Home Insulation is a light, fluffy mineral wool, highly efficient in heat-proofing practically any building, old or new. It is durable, rot-proof, fire-proof and odorless, and will not corrode or settle. Full stud thickness of this material will save up to 30 per cent on fuel in winter and help keep rooms up to 15F cooler in hottest weather. J-M Rock Wool Home Insulation is furnished in two forms: for new construction, in easily handled batts; for existing buildings, in nodulated form to be installed pneumatically.

For New Construction
J-M Super-Felt* Batts

Super-Felt Home Insulation is furnished in pre-fabricated batts of uniform thickness and density, in both full stud thickness and semi-thick, in sizes 15 x 24 in. and 15 x 48 in., designed to fill completely the space between studs, joists and rafters on the usual 16 in. centers. The sturdy felted "wool" is strong enough to be handled rapidly without damage. The batts are backed with a vapor-seal paper, extending on both the long sides in 1½ in. wide flanges, by which the batt is fastened in place and which also aid in sealing the joints. This backing protects against passage of abnormal humidity, that may be present in the house, into the structure.

For Existing Homes and Buildings
Type A "Blown" Rock Wool

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors. Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in.; the density, approximately 5 to 8 lb per cu ft, assures maximum thermal efficiency. This type of insulation is installed only by Johns-Manville or by Approved J-M Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

J-M Airacoustic* Sheets for lining Air-Conditioning Ducts

J-M Airacoustic Sheets, for duct linings of air conditioning systems, are flame-proof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Airacoustic sheets are furnished 24 x 36 in., ½, 1 and 1½ in. thick.

Johns-Manville Pipe and Boiler Insulation

J-M Pre-Shrunk Asbestocel*
Pipe Insulation

Cellular type of insulation for pipes carrying low pressure steam or hot water.

Made up of alternate layers of plain and corrugated, specially-treated, moisture-resistant, asbestos felts. Three finishes: Glazed White for quick application, will not carry flame; asbestos paper; and regular canvas cover.

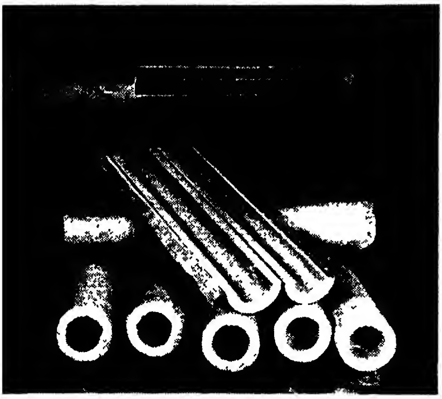
Furnished in 3-foot sections, in thicknesses of 2 through 8 plies, each ply approximately ¼ in. thick.†

* Reg. U. S. Pat. Off.

JOHNS MANVILLE PIPE AND BOILER INSULATION, Cont'd

J-M 85% Magnesia

Recommended as the most widely used insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all standard pipe sizes†, in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, 1 in. through 4 in. thick. Minimum thickness for curved blocks, 1¼ in.



J-M 85% Magnesia Pipe Insulation

J-M Pre-Shrunk Wool Felt Pipe Insulation

J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. Prevents sweating on cold water pipes. Made of a specially indented wool felt and provided with a dual service liner.

Supplied in canvas finish in 3-ft sections in thicknesses of ½ in., ¾ in., 1 in., Double ½ in., and Double ¾ in., for all standard pipe sizes.†

J-M Asbesto-Sponge* Felted Pipe Insulation

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft sections up to 3 in. thick, for all standard pipe sizes.†

* Reg. U. S. Pat. Off.

J-M Superex* Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and 85% Magnesia insulations are both furnished in sectional and segmental pipe covering, and in block forms.

J-M Asbestocel* Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating low pressure boilers, feed water heaters and warm air ducts. Temperature limit 300 F. Furnished 6 to 36 in. wide by 36 to 96 in. long, from ½ in. through 4 in. thick.

J-M Rock Cork* Sheets and Pipe Insulation

Rock Cork is made of mineral wool and an asphaltic binder molded into sheets and pipe insulation for all low temperature service to *minus* 400 F. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating value is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1, 1½, 2, 3, and 4 in. thicknesses. Lagging, for curved surfaces, supplied 18 in. long by 1½, 2, 3, and 4 in. thick, 2 to 6 in. wide, depending on diameter. Pipe covering furnished in Ice Water, Brine and Heavy Brine thicknesses, for all commercial pipe sizes.†

J-M Zerolite* Sheets and Pipe Insulation

Zerolite is a newly developed, resin bonded, mineral wool insulation for temperatures to *minus* 400 F. In addition to possessing the same basic characteristics as Rock Cork, Zerolite is highly fire-retardant, resists petroleum and organic solvents, and has the added advantage of 6 to 10 per cent lower conductivity. Furnished in the same types and sizes as Rock Cork.

Details on Request

Write for complete information on any Johns-Manville insulating material.

† Also available in sections to fit straight runs of copper pipe or tubing with nominal diameters of ⅜ in. and larger.

Kimberly-Clark Corporation

Neenah, Wisconsin

KIMSUL
INSULATION

New York 17, N. Y.
122 East 42nd Street
Atlanta 3, Georgia
22 Marietta St., N. W.
Chicago 3, Illinois
8 S. Michigan Avenue
San Francisco 4, Calif.
155 Sansome Street

A PRODUCT OF Kimberly Clark RESEARCH

Many-Layer, Stitched—Each ply is continuous and separate; all are held together at the density of maximum efficiency by strong stitching. No heat-leaking thin spots, no money-wasting thick spots in the Kimsul blanket.

KIMSUL is unique among building insulations and acoustical materials because it is made of many individual plies—each one a continuous separate layer of soft, clean, creped, asphalt-treated cellulose fibers. Each ply is controlled carefully in manufacture both as to thickness and crepe structure. The result is a flexible blanket, inherently uniform in thickness—an important factor in a thermal or sound insulating material. The Kimsul plies and the creped Pyrogard* cover are held together with rows of strong stitching.

KIMSUL is reduced to ⅕ its installed volume for easier shipment, handling and storage. On the job, the Kimsul blanket is expanded in installation. The stitching controls the expansion to the density of maximum efficiency.

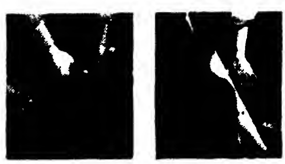
Flexible—fits into corners, tucks behind pipes, electrical wiring and other "tight spots." No areas unprotected.

Clean—no sharp particles to irritate, nothing to sift; stitched ply construction prevents settling or sagging.

Caulkable—one ply or many plies may be compressed to high density in narrow or wide joints, sealing out cold air and sound. Kimsul asphalt-treated wood fiber does not break up during caulking or tamping operation.

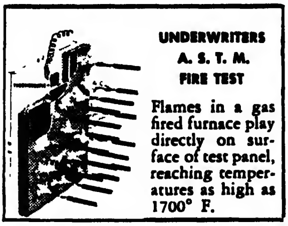
Insulated Fastening Edge—Kimsul blankets are extra wide to provide fully insulated fastening edges, and to insure completely filling spaces where framing may be slightly off center.

Over-Framing Compressibility—Kimsul insulation, in Standard and Commercial thicknesses, may be easily compressed over framing members. Especially valuable for 48 in. wide Kimsul—suitable for mass or prefabricated construction.



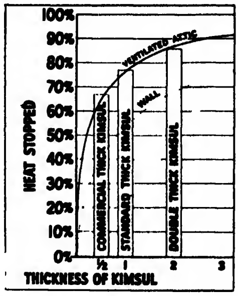
Any Width, Any Length—It's easy to cut exact lengths or narrow widths. Avoids muss and fuss. Workmen do a fast, neat job—with Kimsul.

* T. M. Reg. U. S. Pat. Off

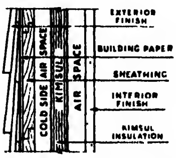


Fire-Resistant — Special permanent chemical treatment makes Kimsul resist fire. **PYROGARD fire-resistant cover**—(a Kimsul feature) resists flame-spread.

Small Storage Space—Easy Handling—Kimsul package is small, tough. Compressed to ¼ installed volume. Compact; easy to carry; convenient to store; not easily damaged.

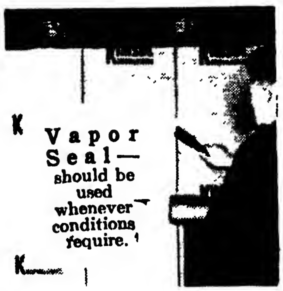


Fuel and Power Saving—first increment of thickness gives greatest value.



Air Space—is a prime requisite. Use a vapor-permeable building paper under exterior finish. Ventilation in Attics and Floors should never be omitted. Use approximately one sq ft of louver area for 1000 sq ft of ceiling area.

Light Weight—1.4 lb per cubic foot. Standard Thick Kimsul weighs only 115 lbs per 1000 sq ft.
Moisture-Resistant—Asphalt treatment of each ply sheds water.
Resists Mold, Rot, Vermin—The materials of which Kimsul is made offer no subsistence to vermin or insects. Special chemical treatment resists mold and fungus.
"k" Factor—0.27 Btu/sq ft/hr./°F.



Vapor Seal—should be used whenever conditions require.

SOUND CONTROL →

Sound Deadening (*one room to another*). Kimsul flexible blanket used in staggered stud construction.

1) Absorbs sound from diaphragmatic action of wall panels.
2) Absorbs sound which leaks through joints, thus maintaining original sound resistance of partition.
3) Cushions wall surface.
4) Prevents accidental bridging.

Sound Absorption (*within a room*). →

KIMSUL'S blanket design makes it inexpensive as a sound absorbing element. See coefficients below.

KIMSUL BUILDING INSULATION SPECIFICATIONS

Thicknesses	Thermal Resistance "R"	Sound Absorption (Av.)	Square Feet Per Roll Standard Widths 16, 20, 24 inches	Square Feet Per Roll Standard Widths 48 inches
Commercial Thick (Approx. 0.5 In.)	1.85	.40	200	500
Standard Thick (Approx. 1.0)	3.70	.55	200	500
Double Thick (Approx. 2.0)	7.40	.70	100	250

For further information, write to
THE KIMBERLY-CLARK CORP., NEENAH, WIS.

Insul-Mastic Corporation of America

General Offices:

1150 Oliver Bldg., Pittsburgh 22, Pa.

GILSONITE INSUL-MASTIC TYPE "D" INSULATION INSULATES, DEADENS SOUND, RUSTPROOFS, WATERPROOFS Steel, Copper, Tin, Galvanized Sheet Metal Ducts, Plates, Partitions, Buildings, etc.; Pipes, Pipe Lines; Oil Tanks, Water Tanks, etc.

An application of Gilsonite Insul-Mastic Type "D" Insulation reduces the acoustic properties of metal plates, metal partitions, metal ducts, etc. It insulates against cold, reduces heat losses, is impervious to humidity conditions, and protects all surfaces over which it is applied against corrosion. It is chemically inert, and not subject to electrolysis.

Type "D" is waterproof throughout its entire mass, and is especially effective in preventing mildew and rot caused by condensation of moisture vapor on encased metal panels. The total moisture absorption when kept continually submerged in water, is only 3.2 per cent—too small to exert any practical effect.

A unique feature of Type "D" is that, without any form of mechanical attachment, it bonds tenaciously with surfaces at any angle from vertical to horizontal, including ceilings. Its bond is not broken by extremes of heat or cold, and the material adheres tenaciously throughout its many years' life of full efficiency.

Type "D" is a highly viscous, semi-plastic material composed of from 65 per cent to 75 per cent granulated cork held in a matrix of genuine Gilsonite asphalt which surrounds each individual cork granule like a jacket. The total asphaltic base of Type "D" contains approximately 50 per cent washed select fine Gilsonite. Inert fillers of flake mica, asbestos fibre, etc., provide for prolonged efficient functioning.

Application of Type "D" is easily and rapidly made by spray under high air pressure, thereby assuring uniform thickness of covering over entire surfaces sprayed, regardless of "high" or "low" areas, angles, bolt or nail heads, or other surface irregularities. There are no joints or seams in a sprayed-on application. (Type "D" can also be troweled.) The material dries in from 3 to 9 days depending upon weather, and reaches constant weight in about 18 days.

A density as low as 16 lb per cu ft can be obtained with Gilsonite Insul-Mastic Type "D" when applied by spray and dried to constant weight. When applied by trowel, density varies according to the manner in which troweling is done; a fair average is 30 lb per cu ft when troweled by hand. The extremely low density (or high insulating value) of Type "D" obtained by spray application, is because that only by spraying can a large amount of desirable air cells be incorporated and retained in the mastic.

Quantities of Gilsonite Insul-Mastic Type "D" Insulation required for different thicknesses, per 100 sq ft are approximately: ⅛ in.–12 gal; ¼ in.–20 gal; ⅜ in.–30 gal. When applied ¼ in. thick at the rate of 20 gal per 100 sq ft, the approximate weight of the applied material is 1½ lbs per sq ft. Shrinkage is negligible.

The thermal conductivity ("K" factor) of Gilsonite Insul-Mastic Type "D" Insulation, per sq ft per 1 in. thickness, is approximately 0.36.

An application of Type "D" ¼ in. thick reduces heat flow through metal plates from 60 to 65 per cent.

Gilsonite Insul-Mastic Type "D" retains its full functions of insulation and resistance to acids and alkalies throughout temperature ranges from 300 F to −40 F, but above 230 F, the coating becomes progressively more susceptible to injury by mechanical means. Down to −40 F, the material does not check, crack, scale, or lose its bond.

Type "D" has a black finish and does not require any additional painting or other exterior surface treatment. If a light or bright surface is preferred, however, Insul-Mastic manufactures a gilsonite base aluminum coating which is completely compatible when sprayed over Type "D." The resulting combination has the added advantage of extremely high reflective properties. If a colored finish is desired for beautifying purposes, Insul-Mastic Corporation supplies special Vermont slate granules in a variety of colors for surface application over Type "D."

Type "D" is used as it comes from the drums. No pre-heating or other preparation of the material is necessary.

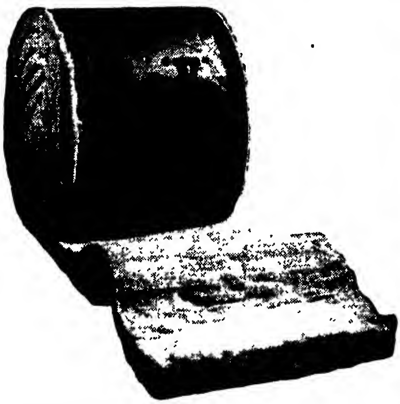
Catalog on request.

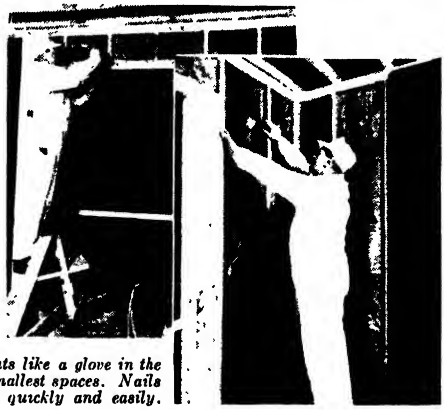
Lockport Cotton Batting Co.

Lockport, New York

FLAMEPROOFED **"Lo-K"** ***COTTON INSULATION***

Fire proofed and manufactured under Department of Agriculture Specifications





Fits like a glove in the smallest spaces. Nails in quickly and easily.

Comes in four featured types to meet every insulation need: (1) *Standard*, open, blanket roll, backed by tough, waterproof, asphalt-coated kraft paper to form an effective vapor barrier. (2) *Enclosed Blanket.* Insulation is completely enclosed in envelope made of asphalt-coated paper on one side and a porous or "breather" type paper on other side. (3) *Open Aluminum Foil*—providing all the features of open Type 1 plus the extra value of aluminum foil backing. Forms an effective vapor barrier—stops 90 per cent of radiant heat. (4) *Enclosed Aluminum Foil.* Superior in insulation plus values and thermal efficiency. Provides greater convenience, comfort, economy and performance.

Thermal Conductivity—The "k" value for cotton is 0.24 Btu/hr/sq ft/degree F/inch. (*See table.*)

Light Weight—Weight of 1 cu ft is ⅞ lb. (*See table.*)

Flame-Proofed—To comply with Department of Agriculture specifications.

Moisture-Resistant—Chemical treatment, combined with natural protective coating on cotton fibres, enables cotton to effectively resist moisture. Prevents rot and mildew.

Smooth Texture—Cotton contains none of the sharp particles that irritate the skin.

Flexible—Cotton batt may be expanded or contracted to fit any enclosure.

Easy to Warehouse and Handle. Offers far more "compressibility." Requires one third the trucking and warehouse space of ordinary insulation.

Simple and Economical to Install. Saves from 25 to 40 per cent in costs.

Designed to Maintain Maximum Utility. Resists all types of deterioration. Won't sag or settle. Packaged in Rolled Form.

Thicknesses—inches: 1, 1½, 2, 3, 3⅝. Width—16, 20, 24 in. centers. Lengths: Standard from 12 ft up.

INSULATING VALUE OF VARIOUS INSULATORS[a]

The coefficients of conductivity (k value) are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. of thickness

Type of Insulation	Wgt. per Cu Ft	k^b Value
Cotton: Insulating Batt	.875	0.24
Rock Wool: Fibrous material made from rock	10.00	0.27
Mineral Wool: Fibrous material made from mineral slag	...	0.27
Glass Wool: Fibrous material made from glass slag	1.50	0.27
Rigid Insulation made from sugar cane fibre	13.50	0.33
Chemically treated wood fibre between layers of paper	3.62	0.25
Eel grass between layers of paper	3.40	0.25
Stitched and creped expanding fibrous blanket	1.50	0.27
Shavings: Various from planer	8.80	0.41
Corkboard: No binder added	7.00	0.27
Rigid insulation made from wood fibre	15.90	0.33
Rigid fibre board made from shredded wool and cement	24.20	0.46

[a]Compiled from Chapter 6.
[b]"k" indicates temperature conductivity.

The Pacific Lumber Company

100 Bush Street, **San Francisco 4, Calif.**

35 East Wacker Drive, **Chicago 1, Ill.** 5225 Wilshire Blvd., **Los Angeles 36, Calif.**

PALCO INSULATION WOOL

PROVIDES EFFICIENT INSULATION

FLAME PROOF SAFERIZED PROCESS

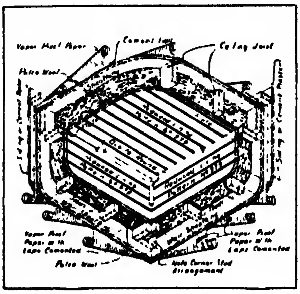
Here are some of the outstanding qualities that make Palco Wool an ideal insulation for either private home or low-temperature refrigeration—Low thermal conductivity (0.255 Btu) makes it one of the most efficient insulations in use today. Palco Wool is entirely self-supporting, and will not settle or compact inside a wall. Permanence is further insured by its ability to withstand moisture—due to its non-hygroscopic characteristics, it will not decay or deteriorate. Palco Wool repels insects and vermin. It is odorless, and will not absorb odors. Specially processed Palco Wool is Saferized under an exclusive process which makes it flame-proof.

The low cost and high efficiency of Palco Wool make it your choice for home or cold storage insulation. Constant low temperature for refrigeration—year 'round comfort for homes—these are the benefits offered by Palco Wool.

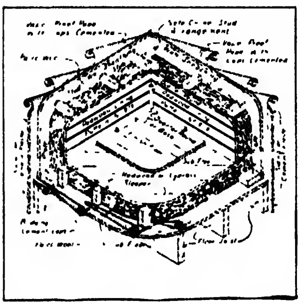
Write now for your free copy of these booklets—The Cold Storage Manual or the Home Owners' Manual.

Check These
8 Proved Qualities:

1. High thermal efficiency
2. Economy
3. Durability
4. Non-Settlement
5. Odor-Proof
6. Vermin-Repellent
7. Moisture-Resistant
8. Fire-Resistant

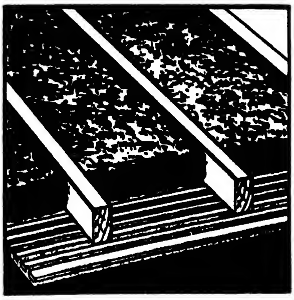


Typical method of applying insulation for ceiling of Cold Storage building, showing vapor-proofing.



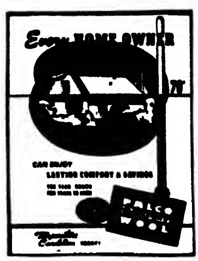
Normal application of insulation and vapor-proofing for floor of Cold Storage building.

Cold Storage Manual



Installation of a full four-inch thickness of Palco Wool in ceiling of private home.

Home Installation in bathroom, using Palco Wool as a noise muffler and an insulator.



Home Owners' Manual

United States Gypsum Company

General Offices: 300 W. Adams Street, **Chicago, Ill.**

INSULATION PRODUCTS

Batt **Decorative** **Structural**

RED TOP* INSULATION WOOL

Red Top Batt

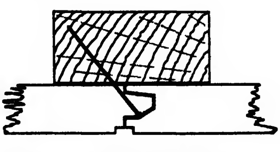
RED TOP MINERAL WOOL BATTS
"k" Valve .27 Btu's

Made for standard (16 in. or 24 in. O.C.) framing in two thicknesses, Medium (Approx. 2 in.) and Thick (Approx. 3 in.). Lengths are either 24 in. or 48 in. Batts are provided with an efficient vapor barrier paper providing nailing flanges for easy application. Adequately packed to insure material will arrive at the job in good condition.

Decorative

Decorative

WEATHERWOOD* PLANK—Manufactured in widths of 8, 10, 12 and 16 inches and in lengths 6, 8, 10 and 12 feet—½ inch thick. The fitted edges (see cut) conceal nails and seal against dust and air infiltration. WEATHERWOOD Plank is made in Blendtex (gray and tan blends) and Hilite (ivory) colors. When combined in variations of shades and width, Weatherwood Plank produces maximum values in both insulation and decoration.



Appearance of edge after installation

*Reg. U. S. Pat. Off.

WEATHERWOOD TILE—Available in 12 x 12, 12 x 24, 16 x 16 and 16 x 32 inches in ½ thickness. Colors are Blendtex (gray and tan blends) and Hilite (ivory).

WEATHERWOOD PANELTILE—Hilite color available in 12 x 24 and 16 x 32 inches in ¾ inch thickness. Blendtex colors available in 12 x 24 and 16 x 32 inches. Tile sizes 12 x 24 and 16 x 32 inches can be mill cross scored to represent 12 x 12 and 16 x 16 sizes. All tile have fitted edges.

Structural

WEATHERWOOD SHEATHING—Asphalt coated. 2 feet x 8 feet x ²⁵⁄₃₂ inches thick, with tongue and grooved long edges for horizontal application. Also available in 4 x 8, 4 x 9, 4 x 10 and 4 x 12 feet in either ½ inch or ²⁵⁄₃₂ inches thickness with square edges for vertical application.

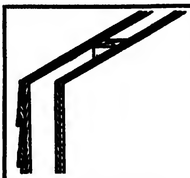
WEATHERWOOD BUILDING BOARD—4 feet wide, made in lengths 6, 7, 8, 9, 10 and 12 feet, ½ inch thick in ivory color. Effectively insulates, strengthens and decorates.

WEATHERWOOD INSULATING LATH—18 x 48 inches x ½ inch thick with V joint on long edges. Gives an excellent plaster bond and also acts as a cushion for plaster with sound deadening qualities.

Heat Loss Factors

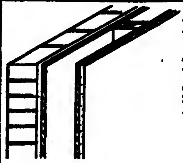
The heat loss factors shown on the opposite page indicate the comparative insulation value of various insulating treatments included in common construction systems.

NOTE: These figures apply to 1 story buildings. To get figures for 2 story homes add 20 per cent to the values below for the wall constructions and divide by one-half for floor and ceiling constructions. It is important to use correct factor due to variations in the ratio of wall and window areas.



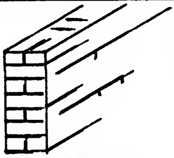
WALLS
Basic Construction—Frame
Wood Siding
Wood Sheathing
2 x 4's
Rocklath & Plaster

	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.248	.120	.083	.064
Substituting in Above Basic Const.				
a. 25/32" WW Sheathing	.186	.105	.076	.059
b. ½" WW Plaster Base	.183	.103	.076	.059
c. ¾" WW Plaster Base	.157	.094	.069	.055
d. 1" WW Plaster Base	.143	.088	.067	.054
e. 25/32" WW Sheathing and ½" WW Plaster Base	.147	.090	.068	.054
f. ½" WW Bldg. Bd., Tile or Plank	.187	.104	.077	.059
g. ¾" Bldg. Bd., or Tile	.160	.095	.070	.056
h. Gyplap Sheathing	.310	.135	.090	.068
i. ½" Sheetrock	.260	.123	.085	.064
j. Gyplap Sheathing and ½" Sheetrock	.330	.136	.091	.069



WALLS
Basic Construction—Brick Veneer
4" Brick
Wood Sheathing
2 x 4's
Rocklath & Plaster

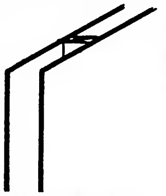
	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.270	.125	.085	.065
Substituting in Above Basic Const.				
a. 25/32" WW Sheathing	.202	.111	.078	.061
b. ½" WW Plaster Base	.200	.107	.077	.060
c. ¾" WW Plaster Base	.178	.102	.073	.058
d. 1" WW Plaster Base	.157	.094	.070	.055
e. 25/32" WW Sheathing and ½" WW Plaster Base	.162	.095	.070	.055
f. ½" WW Bldg. Bd., Tile or Plank	.215	.112	.079	.061
g. ¾" WW Bldg. Bd. or Tile	.187	.104	.075	.059
h. Gyplap Sheathing	.350	.128	.087	.066
i. ½" Sheetrock	.288	.130	.088	.066
j. Gyplap Sheathing and ½" Sheetrock	.368	.142	.093	.069



WALLS
Basic Construction
8" Brick Wall—4" Face Brick and 4" Common Brick—No interior finish

	No Wool Between Stud*	Adding Red Top Wool** 1"	2"	3"
Basic Construction	.500			
Adding to Above Basic Const.				
a. ½" Plaster	.480			
b. Rocklath and Plaster (Furred)	.300	.143	.093	.069
c. ½" WW Plaster Base and Pl. (Furred)	.220	.121	.084	.064
d. ¾" WW Plaster Base and Pl. (Furred)	.190	.112	.079	.061
e. 1" WW Plaster Base and Pl. (Furred)	.160	.101	.073	.058
f. ½" WW Bldg. Bd., Tile or Plank	.230	.124	.085	.065
g. ¾" WW Bldg. Bd., Tile or Plank	.200	.115	.081	.062
h. 1" WW Bd., Plank or Tile	.170	.104	.075	.059
i. ½" Sheetrock Furred	.320	.146	.095	.070

*Based on ⅝" Furring Strip
**Based on Full Dimension

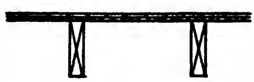


WALLS
Basic Construction—Plywood
Plywood on Wood Studs
⅜" Outside—¼" Inside with one Air Space Over ¾"

	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.431	.151	.095	.074
Substituting in Above Basic Const.				
a. ½" WW Bldg. Bd., Tile or Plank	.275	.126	.087	.065
b. ¾" WW Bldg. Bd. or Tile	.230	.115	.081	.062
c. 1" WW Bldg. Bd. or Tile	.196	.106	.076	.060
d. ⅜" Sheetrock	.430	.151	.095	.074
e. ½" Sheetrock	.413	.148	.095	.074
f. Adding to basic construction 25/32" WW Sheathing	.218	.113	.080	.061

CEILINGS
Basic Construction
⅜" Rocklath and ½" Plaster

	No Wool Between Joists	Adding Red Top Wool 1"	2"	3"
Basic Construction	.610	.169	.116	.080
Substituting in Above Basic Const.				
a. ½" WW Plaster Base & Plaster	.329	.136	.091	.068
b. ¾" WW Plaster Base & Plaster	.290	.128	.087	.066
c. 1" WW Plaster Base & Plaster	.213	.110	.079	.061
d. ½" WW Bldg. Bd., Tile or Plank. No Plaster	.356	.139	.092	.067
e. ¾" WW Bldg. Bd. or Tile	.268	.124	.086	.065
f. ½" WW Bldg. Bd. or Tile	.220	.113	.080	.062
g. ⅜" Sheetrock	.670	.174	.118	.089
h. ½" Sheetrock	.635	.170	.115	.079



FLOORS
Basic Construction
Maple or Oak Flooring on Yellow Pine Sub-Flooring

	No Wool Between Joists	Adding Red Top Wool 1"	2"	3"
Basic Construction	.340	.138	.091	.068
Adding to Above Basic Const.				
a. ½" WW Bd. on bottom of joists	.180	.102	.075	.059
b. ¾" WW Bd. on bottom of joists	.158	.094	.070	.055
c 1" WW Bd. on bottom of joists	.141	.088	.066	.053

Above calculations based on data from A.S.H.V.E. GUIDE—1942.

Wood Conversion Company

Dept. 220-9 First National Bank Building
St. Paul 1, Minnesota
NEW YORK CHICAGO DENVER

EFFICIENT INSULATION FOR EVERY NEED

For many years a leader in the insulation field, Wood Conversion Company manufactures a complete line of flexible, fiber and rigid insulation for all industrial and domestic purposes. This insulation is the product of scientific research, and is especially designed to embody the most desirable qualities for every use. Backed by the name of Weyerhaeuser, Wood Conversion Company insulation assures high efficiency and long, satisfactory service.

Balsam-Wood Sealed Blanket Insulation—A building insulation that is completely armored against the three major insulation enemies—wind, vibration and moisture. The Balsam-Wool mat is completely enclosed in tough vapor resistant liners which surround all four sides, with spacer flanges which assure positive application. Balsam-Wool will not settle or pack down within walls—is fire retardant and vermin resistant.

K-25—A clean, new wood fiber insulation with an amazingly low K factor. Shipped in compact bales, K-25 is easily processed on the job to create a lastingly effective insulation and is ideal for use in walk-in coolers, cold storage plants, locker plants, etc.

K-25 Fiber Pneumatic System—The modern, high-speed, automatic way to insulate domestic refrigerator cabinets and doors. Fluffed to proper low density in the manufacturer's plant, K-25 is blown into cabinets and doors under high pressure, forming a tightly felted insulating mat without joints, laminations or voids.

Tufflex—A soft, felted blanket material made from fleecy wood fiber, combines high insulating efficiency with toughness and exceptional resistance to heavy impact blows. Tufflex is light in weight and nonabrasive—will not tear or pull apart even when cut into narrow strips. Tufflex is available in rolls or sheets of various thicknesses and widths.

NU-WOOD STRUCTURAL INSULATION

Nu-Wood Interior Finish—A multiple-purpose wood fiber material available in tile, plank and board. Nu-Wood insulates, decorates and quiets noise. Nu-Wood's colors are soft and harmonious—will not fade. The Nu-Wood interior finish line includes Sta-lite, an insulating interior finish with more than 70 per cent light reflection.

Nu-Wood Insulating Lath—Used as a plaster base, assures strong, smooth, true walls and ceilings, free from cracks.

Nu-Wood Roof Insulation—Will fill all requirements of Federal specification LLL-F-321b. Furnished in any practical thickness.

American Flange & Manufacturing Co. Inc.

30 Rockefeller Plaza, New York 20, N. Y.
Plaza 7-2200

Ferro-Therm

Reg. U. S. Pat. Off.

STEEL INSULATION

FULLY PROTECTED BY U. S. AND FOREIGN PATENTS
ISSUED AND PENDING

Ferro-Therm installed in a cold storage room

Ferro-Therm Steel Insulation, made from rigid steel sheets with a special alloy coating, reflects 90 per cent to 95 per cent of all radiant heat. This high reflectivity, combined with extremely low heat storage capacity, provides maximum insulating efficiency in a minimum overall thickness.

Saves Pay Space and Weight

In cold storage construction, the number of sheets of Ferro-Therm depends on the temperature to be maintained and the U value required. The k value of Ferro-Therm, based on tests, is listed in the Data Book of the *American Society of Refrigerating Engineers* as 0.226 Btu per (hr) (sq ft) (°F temperature difference). Laboratory tests and thousands of applications have demonstrated that a wall of Ferro-Therm will provide insulating efficiency equivalent to a wall of mass insulation approximately twice as thick.

Assures Rapid Pull Down of Temperature

The low heat storage capacity of Ferro-Therm is extremely important in achieving rapid pull down of temperature, and in saving refrigeration costs for the initial and each subsequent cooling of space. Specifically, the heat storage capacity of a single sheet of No. 38 gauge is 0.029 Btu per (hr) (sq ft) (°F temperature difference). This is approximately 1/6 of the heat storage capacity of 1 sq ft of 1 in. thickness corkboard.

Permanent, Fire-Proof Insulation

Ferro-Therm construction eliminates trapping of moisture condensate, with subsequent deterioration of the construction. As it is all-metal, Ferro-Therm cannot be penetrated by rodents, vermin or termites, and is absolutely non-combustible. The value of Ferro-Therm for fire protection is apparent.

125° Below Zero Maintained in Altitude Test Chambers

Ferro-Therm has proved its superiority in buildings, cold storage rooms, refrigerated cabinets, locker rooms, dry ice containers, refrigerated railway car construction, ovens, high-temperature storage tanks—in fact, practically every type of application where high insulating efficiency with economies in space and weight are a requisite. The most notable demonstration of Ferro-Therm performance has been its selection for the insulation of altitude chambers for the testing of Army and Navy aviation equipment and personnel. In these chambers, temperatures as low as −125°F were maintained, with a temperature drop of +70°F to −70°F in 10 to 12 min.

Our catalog, giving data and installation details on Ferro-Therm, will be sent upon request.

Infra Insulation, Inc.

10 Murray Street, New York 7, N. Y.

Telephone: COrtlandt 7-3833

Thermal Factors of Infra Insulation

C.052	Heat Flow Down	- Equals	6″ Rockwool :	U - .045
C.083	Heat Flow Up	- Equals	3.97″ Rockwool :	U - .066
C.10	Lateral Heat	- Equals	3.33″ Rockwool :	U - .068

Infra Insulation, Type 4, is a tough, multiple sheet, aluminum insulation with 4 heat ray reflective surfaces: two outer reflective spaces and 2 rows of inner, triangular reflective spaces. It is easily and quickly installed between joists and studs, steel girders, etc. The 2¼ cu ft carton contains 1000 sq ft and weighs but 55 lbs.

Heat flow through wall spaces without insulation is as follows: by Conduction, 5 per cent to 7 per cent; by Convection, 15 per cent to 25 per cent; by Radiation, 65 per cent to 80 per cent.

RADIATION:—The aluminum surfaces of Infra Insulation turn back, REJECT, 97 per cent of the heat rays which strike them. They transmit, or emit, or radiate only 3 per cent of any heat actually absorbed by all the three heat flow methods. The surfaces of ordinary insulation ABSORB and EMIT more than 90 per cent.

CONDUCTION:—One sq ft of Infra Insulation weighs but 1 oz, has only 1 cu in. of mass. The ratio is *1 of Infra mass to 431* of low-conductive air. Ordinary insulation has a ratio of *1 of mass to 23* of air.

CONVECTION AND VAPOR:—Each of the 2 tough aluminum sheets of Infra Insulation has ZERO permeability to all gases, including heated air, cold air, and water vapor. Together with the inner, separating ACCORDION partition, they prevent the flow of heat by Convection, and the flow of vapor when properly installed.

In their book, "Insulation," Dalzell & McKinney of the *American Technical Society* state on Page 29: *"Thermal insulation with a metal is made possible by taking advantage of the low thermal emissivity of aluminum foil and the low thermal conductivity of air. It is possible with this type of insulation practically to eliminate heat transfer by radiation and convection."*

Conveniently packaged, Infra is easily installed.

Infra Insulation uses 99.5 per cent pure Aluminum, made in accordance with special Infra Emissivity Specifications. It has 500 per cent to 1700 per cent greater bursting strength than ordinary foils. The special fiber used is permanently flame-proof, mold-proof and vermin-proof. By nature of its structure, Infra Insulation can not form condensation, nor absorb nor store moisture.

Since Infra emits practically no heat rays, and since aluminum's melting point is 1250F, Infra Insulation is an efficient FIRE-STOP. Infra is sanitary, inhospitable to rodents and vermin and DOES NOT RETAIN ODORS. Mechanics like to work with Infra. It is CLEAN, free of DUST or lint, with permanent freedom from floating particles.

Ask for a copy of the new edition of our authoritative manual, "Simplified Physics of Thermal Insulation," address Dept. V.-G.

ALFOL INSULATION

Manufactured by

Reflectal Corporation

155 East 44th Street
New York 17, N. Y.

HEAT INSULATION for ALL PURPOSES

Easy to Apply

Handy Package—250 Sq Ft of Insulati

Alfol Building Blanket—for all Types of Building Structures.

Alfol Building Blankets consist of spaced layers of Alfol Aluminum Foil Insulation attached along the edges to a liner sheet of heavy vapor proof paper.

Packaged in handy rolls of 250 sq ft each for use on 16 in., 20 in. or 24 in. centers. Weighs less than $^1/_{10}$ lb per sq ft.

- **High Insulating Efficiency**
- **Positive Vapor Barrier.**
- **Low Heat Storage Capacity.**
- **Negligible in Weight.**
- **Moisture Proof.**
- **Durable.**
- **Odorless and Clean.**
- **Easily Applied.**
- **Low Cost.**

Specifications

Description	Widths	Net Area per Roll	Net Weight per Roll
Type I.— 1 Layer ALFOL	16"-24"	250 sq. ft.	17 lbs.
Type II.— 2 Layers ALFOL	16"-20"-24"	250 sq. ft.	19 lbs.
Type III.— 2 Layers ALFOL	16"-24"	250 sq. ft.	20 lbs.
Type IV.— 3 Layers ALFOL	16"-20"-24"	250 sq. ft.	23 lbs.

A L S O

● ALFOL PREFABRICATED INSULATION PANELS

For Tanks, Towers, And All Types of Heated Equipment.

● ALFOL ASBESTOS

Alfol Aluminum Foil Insulation Laminated To Asbestos For Ovens, Ranges, Boiler Jackets, Hot Water Heaters, And All High Temperature Insulation Purposes.

● ALFOL JACKETING

Heavy Reinforced Paper Combined With Alfol Aluminum Foil. Provides Insulation And Vapor Barrier In One Convenient Form. For Refrigerator Cars, Trucks, Buses, Trailers, Etc.

Write for complete information, catalogs and prices.

Silvercote Products, Inc.

161 East Erie Street, Chicago, Ill.

SILVERCOTE Reflective INSULATION

Silvercote Heat Reflective Surfaces—The silver-like surface of Silvercote reflective insulation consists of a polished, heat reflective coating applied to a special kraft paper. The importance of using a Silvercote radiant heat reflective surface in modern building construction is obvious when it is realized that from 50 to 80 per cent of the heat transferred across a normal air space is in the form of radiation.

REFLECTIVE SHEET INSULATIONS

Silvercote Duplex—A thin flexible vapor barrier and insulation consisting of two sheets of Silvercote paper bonded together with asphalt. This material, containing two exposed Silvercote surfaces, weighs approximately 65 lbs per thousand sq ft and is manufactured in 500 sq ft rolls in widths of 36 in. or 52 in. to span two 16 in. or 24 in. standard framing spaces. These widths permit bow-in of the Silvercote over the room side of the framing members to form an air space between the insulation and the interior finish. The water vapor permeability of Silvercote Duplex, when tested in accordance with the dry method *ASTM* tentative standard C214-47 and subsequent modifications, is 0.23 grains per sq ft per hour per inch of mercury vapor pressure difference.

Silvercote Simplex—An economical vapor permeable reflective insulation designed for use where a vapor barrier is not required. Silvercote Simplex is a single sheet of special kraft paper coated on both sides with the Silvercote surface. It weighs approximately 30 lbs per thousand sq ft and is manufactured in 500 sq ft rolls in 36 in. or 52 in. widths to span two 16 in. or 24 in. standard framing spaces. These widths permit bow-in of the Silvercote over the exterior side of the framing members to form an air space between the insulation and the exterior sheathing. The water vapor permeability of Silvercote Simplex, when tested in accordance with the dry method *ASTM* tentative standard C214-47 and subsequent modifications, is 99.2 grains per sq ft per hour per inch of mercury vapor pressure difference.

REFLECTIVE BLANKET INSULATIONS

Blanket Insulation, Silvercote on Vapor Barrier Side—A popular building insulation available in various thicknesses and faced on the vapor barrier side of the blanket with Silvercote paper. Manufacturers of this type of reflective blanket apply an asphalt coating to the back of the Silvercote paper for the twofold purpose of bonding the insulation material to the Silvercote paper and to lend vapor resistant properties to the blanket at the room side of the insulation. The opposite side of this type of reflective blanket is a plain kraft liner perforated where necessary to allow compression packaging and also to assure ample water vapor permeability at the so-called breather side of the blanket.

Reflective Blanket, Silvercote on Breather Side—This product was developed for application in structures where only one air space adjacent to the cold side of the blanket is available. An air space faced on one side with a Silvercote surface will have approximately equal effectiveness whether formed on the vapor barrier or the breather side of the blanket. Since Silvercote paper is in itself not a vapor resistant material, it can readily be used on either side of blanket insulation. The required vapor barrier on the room side of blanket insulation is provided by the asphalt bonding agent.

Reflective Blanket, Silvercote on Both Sides—A de luxe insulation material utilizing the maximum insulation value of blanket insulation, air spaces and heat reflective surfaces. The vapor barrier side of this reflective blanket is Silvercote paper bonded to the insulation material with an asphalt vapor barrier adhesive. The breather side of this efficient insulation is therefore plain vapor permeable Silvercote paper.

AVAILABLE UPON REQUEST

Silvercote's Handbook of "U" Values

108 page illustrated booklet listing 12,852 "U" values of various walls, floors and ceilings

Silvercote's Handbook of "U" Values is unique in that it provides summer as well as winter "U" values. The Handbook's special listing of ceiling "U" values to indicate heat flow down characteristics will be of interest to those who are concerned with summer comfort as well as winter fuel savings.

Specimens of Page Style and Content Arrangement

CEILINGS
1/2" PLASTER ON
3/8" GYPSUM LATH

Ventilated Space Between Ceiling Section and Roof

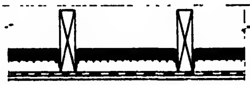
INSULATION: FLEXIBLE BLANKET, SILVERCOTE ON BOTH SIDES

INSULATION THICKNESS	COVERING OVER JOISTS	DIRECTION OF HEAT FLOW: DOWN — SUMMER	UP — WINTER, TEMPERATURE ZONES: -30°	-20°	-10°	0°	+10°	+20°	+30°
1" Blanket	None	.10	.14	.14	.14	.14	.14	.14	.14

CEILINGS
1/2" PLASTER ON
3/8" GYPSUM LATH

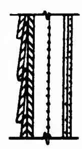
Ventilated Space Between Ceiling Section and Roof

INSULATION: FLEXIBLE BLANKET, SILVERCOTE ON VAPOR BARRIER SIDE



INSULATION THICKNESS	COVERING OVER JOISTS	DIRECTION OF HEAT FLOW: DOWN — SUMMER	UP — WINTER, TEMPERATURE ZONES: -30°	-20°	-10°	0°	+10°	+20°	+30°
1" Blanket	None	.11	.15	.15	.15	.15	.15	.15	.15
2" Blanket	None	.08	.10	.10	.10	.09	.09	.09	.09

WOOD FRAME WALLS
YP LAP SIDING



INSULATION: SILVERCOTE DUPLEX

SHEATHING	INTERIOR FINISH	TEMPERATURE ZONES: -30°	-20°	-10°	0°	+10°	+20°	+30°
Fir or Y.P. 25/32" and Building Paper	1/2" plaster on 3/8" gypsum lath	.15	.15	.15	.14	.14	.14	.14
	3/4" plaster on metal lath	.15	.15	.15	.15	.15	.14	.14
	3/8" gypsum wallboard	.15	.15	.15	.15	.15	.14	.14
	1/2" insulating board	.12	.12	.12	.12	.12	.12	.12
	1/4" plywood	.15	.15	.15	.15	.14	.14	.14
Plywood 5/16" and Building Paper	1/2" plaster on 3/8" gypsum lath	.16	.16	.16	.16	.16	.15	.15
	3/4" [illegible] lath	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

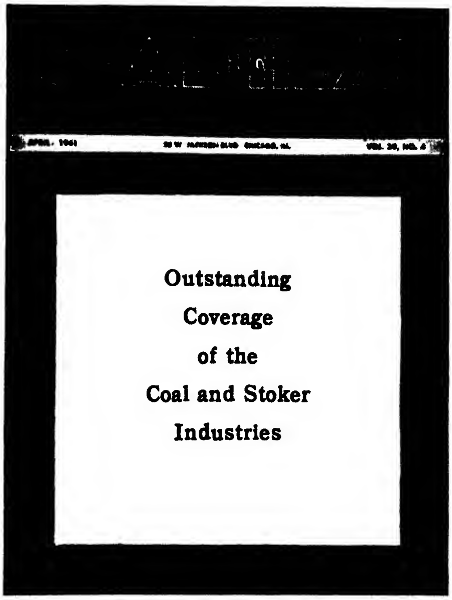
Write for Silvercote's Handbook of "U" Values.

Coal-Heat

Published at

20 W. Jackson Blvd., Chicago 4, Illinois

Phone Wabash 9464



FOR information on the use and sale of stokers, coal and coal heating equipment, you can turn to COAL-HEAT with complete confidence.

Here is a magazine that appeals to every man concerned with the market, use and sale of solid fuel and modern heating equipment. Having long since recognized the importance of properly designed efficiently operated, properly maintained equipment to the successful use of coal, and therefore to the welfare of the coal industry, COAL-HEAT constantly emphasizes the significance of the "equipment factor" in heating merchandising.

It is only natural that COAL-HEAT was the first trade magazine to recognize and promote the small stoker; to introduce many new developments in coal-burning equipment to further fuel conservation; to support the use of dustless treatment; and to urge the sale of equipment by coal men.

COAL-HEAT has at its disposal an almost unlimited number of sources of authentic information on the topics it covers; its articles are written by the best informed men in the coal, stoker and heating industries. It enjoys quite a following, not only among the most progressive merchants in these industries, but among the industry's leading combustion and heating engineers. For years it has championed the importance of the fuel engineer to the coal and stoker industries, and each year prints many articles *for* and *by* fuel engineers.

COAL-HEAT's fundamental editorial policy is "to further the more satisfactory use and sale of coal and modern coal-burning equipment." It actively supports the application of scientific and engineering knowledge to the use of coal and coal-burning equipment. It covers both the merchandising and utilization of the coal, stokers and modern heating equipment.

With over a million stokers in use today, the importance of COAL-HEAT's field is clearly evident. It has been and is COAL-HEAT's job to supply coal and stoker men with the information they need to insure satisfaction for stoker users. The same is true with hand-fired heating plants and all kinds of household and commercial coal heating equipment.

In addition to providing its readers with a basic and diversified editorial program, COAL-HEAT also publishes a number of books and booklets, manuals and reprints covering a wide range of subjects of interest to coal, stoker and heating men. Its series of heating guides for the consumer have proved particularly popular. These are available at small cost.

Subscription rates—$2.00 a year; $3.00 for two years in United States. Canadian and foreign rates $3.00 a year.

Advertising rates and other information will be furnished upon request.

Domestic Engineering Publications
1801 Prairie Avenue, **Chicago 16, Illinois**

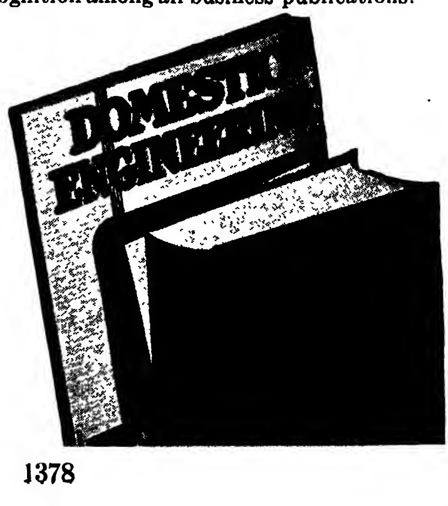
9 TIMES A WINNER! During the past seven years DOMESTIC ENGINEERING editorial programs have been signally honored by our nation's top-flight judges of editorial excellence.

Why do we call attention to these achievements? First, because they are of definite significance to every manufacturer in this field. For, behind all of the editorial projects of these publications there is organization, teamwork, balance and attention to long range objectives. Not only are these the ingredients of prize winning editorial programs, they are prime requisites which must be provided by a publication if the advertising carried in it is to be of greatest possible benefit to readers and advertisers alike.

These awards signify a consistent, year-after-year effort plus a constantly broadening base of editorial service. The prize winning achievements of past years have become the basis for greater and more noteworthy accomplishments of succeeding years. This has been demonstrated by the growing intensity of the competition for these prizes. Over 600 entries were registered in the contest.

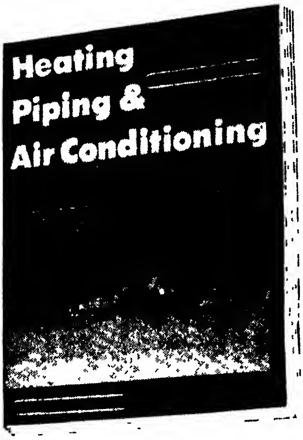
These editorial efforts, launched as routine service for our industry, have been given outstanding recognition among all business publications.

3 Awards in 1947!



KEENEY PUBLISHING COMPANY
6 North Michigan Avenue, **Chicago 2, Ill.**

Heating, Piping and Air Conditioning



This is the publication which carries the Journal of the A.S.H.V.E. in addition to its own regular editorial section. Its field is that of industry and large buildings. It is devoted to the design, installation, operation, and maintenance of heating, piping and air conditioning systems in such plants and buildings.

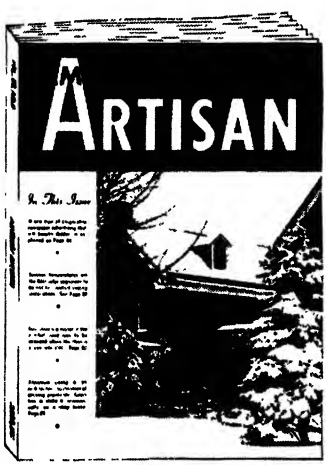
Each January issue includes a complete directory of commercial and industrial heating, piping, and air conditioning equipment, which lists all products, their trade names, and the manufacturers' addresses. It is the established buying and specifying guide of the industry.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping and air conditioning in industrial plants, and other large buildings, federal, state, and city governments, school boards, and public utilities. All A.S.H.V.E. members are subscribers.

Such coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in its selection during the preparation of plans and specifications; in its actual purchase for installation; in its year-'round buying for operating and maintenance requirements. Without waste, the manufacturer of air conditioning products and equipment can reach through H. P. & A. C. those from whom he is seeking the necessary engineering acceptance. Write for our new booklet "A Quick Picture."

Member—A.B.P.—A.B.C. Subscription Prices—U.S. $3 per year.
Canada, Central and So. America—$4 per year. Elsewhere $6 per year.

American Artisan



AMERICAN ARTISAN covers the field of warm air heating, residential air conditioning, and sheet metal contracting. Its readers are warm air heating and sheet metal contractors, dealers, jobbers, manufacturers, and public utility companies.

A special section of each issue has been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central forced warm air heating system. As a result of the ready adaptability of this type of heating system to all air conditioning factors, hundreds of thousands of homes today have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever year-'round air conditioning is desired.

Each January issue includes a complete directory of warm air heating, air conditioning, and sheet metal products and equipment, which lists all products, their trade names, and the manufacturers' addresses.

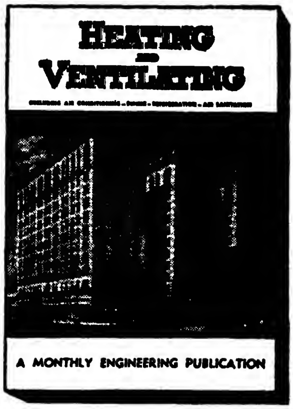
The key man in the residential air conditioning picture is the warm air heating and sheet metal contractor—the one man experienced in "treating air" at a central place and getting it properly distributed. And the key publication—because it reaches these key men with an information service that has made it the recognized authority on residential air conditioning practice—is AMERICAN ARTISAN. For full information on this field, write for a copy of "A Quick Picture."

Member—A.B.P.—A.B.C. Subscription Prices—U.S. $3 per year.
Canada, Central and So. America—$4 per year. Elsewhere $6 per year.

HEATING AND VENTILATING

AIR CONDITIONING - PIPING - HEATING - VENTILATION - REFRIGERATION

HEATING AND VENTILATING is a monthly publication read by engineers and contractors. These are the men who specify, install and maintain the mechanical equipment used in heating, ventilating, air conditioning and refrigeration systems of industrial, commercial and institutional buildings and residential projects. Its readers include also engineers and designers of the firms which manufacture this mechanical equipment.



The editorial content is edited to be of practical use to these engineers, and is prepared under the direction of field-experienced professional engineers.

Generally speaking, the emphasis is on practical rather than on theoretical considerations.

Each month an original Reference Data sheet is included for permanent use in a standard binder.

Special sections are published from time to time. These sections are devoted to subjects of timely interest, such as Heat Pump, Radiant Heating, Piping, Cooling Coils, Air Sanitation, Exhaust Hoods, Air Conditioning, etc. A comprehensive Buyers Directory is published the first of each year.

CIRCULATION

HEATING AND VENTILATING total distribution (A.B.C. Report, May, 1948)—classified as follows:

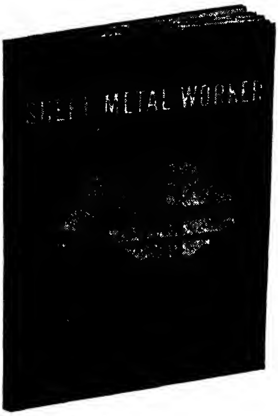
Consulting Engineers (449) and Architects (138) Engineers Employed by them (434)	1,021
Contractors (2,487) and Engineers Employed by Contractors (397)	2,884
Governments and School Boards and their Engineers	881
Public Utility Group	555
Industrial Firms, their Executives, Engineers and other Employees	1,699
Buildings, Real Estate Management Companies, their Engineers	627
Manufacturers of Air Conditioning, Heating, Piping and Ventilating Equipment, Their Officials and Employees (796) and Designing Engineers (239)	1,035
Manufacturers' Agents and Sales Engineering Firms (238) Sales Engineers and Salesmen (892)	1,130
Wholesalers (177) and Dealers (532)	709
Educational Institutions, Libraries, Technical Associations	402
Miscellaneous and Unclassified	396
	11,339
Field Staff, Correspondents, Exchanges, Advertisers, Advertising Agencies and Samples	712
TOTAL	12,051

Subscriptions to HEATING AND VENTILATING, 148 Lafayette St., New York 13, are $3.00 a year.

Sheet Metal Worker

Published by Edwin A. Scott Publishing Company

45 West 45th Street, **New York 19; N. Y.**



Subscription rates—$3.00 per year, U. S., Canada and Pan Amer. Foreign $4.00. Advertising rates on request.

THE January 1949 issue of SHEET METAL WORKER was its 75th Anniversay and Directory Number. It is the oldest publication in its field and is of vital importance to men interested in sheet metal work—air conditioning—warm-air heating and ventilation. Founded in 1874 and published to 1909 by David Williams Company; 1909 to 1920 by United Publishers Corp.; since 1920 by, the Edwin A. Scott Publishing Co.

Subscribers are mainly merchandising contractors purchasing practically all products and equipment which they fabricate, erect or install. Manufacturers, jobbers and distributors also subscribe.

The market has three main divisions:

(1) Equipment for resale in connection with erection or installation work.
(2) Materials for fabrication.
(3) Shop equipment and supplies.

Circulation: SHEET METAL WORKER is a member of the Audit Bureau of Circulations and the Associated Business Papers.

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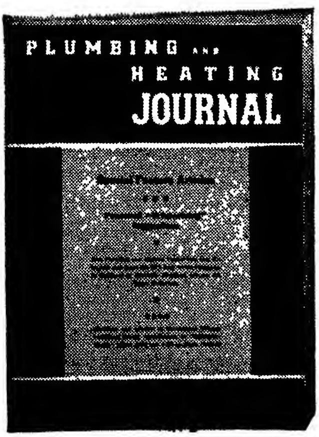

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PLUMBING and Heating Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. It covers both the technical and business phases of their work.

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ENGINEERS OF HUMAN COMFORT

The Heating, Ventilating and Air Conditioning Engineers through their work and research bring to our homes, schools, offices, factories, theaters, hospitals and other public buildings in both summer and winter, that climate best suited to our comfort and health. These men realize the basic importance of heating and ventilating as a primary element in the well-being of civilized mankind, living and working mostly indoors. They are truly *Engineers of Human Comfort.*

Started in 1894, by a small but progressive group, The American Society of Heating and Ventilating Engineers now numbers over 6800 members, whose express purpose is to improve the Art through the interchange of ideas and the stimulation of scientific research and invention.

The Society membership now includes engineers, educators, scientists, physicians, architects, contractors, and leaders of industry. Membership consists of Honorary, Life, Presidential, Member, Associate, Junior and Student grades.

The management of the Society is entrusted to 4 Officers and a Council of 13 elected members. Continuity of policy is insured by electing 4 men annually for a 3-year term and retaining the retiring president on the Council for 1 year. Research work is in charge of the Committee on Research consisting of 15 members, 5 being elected annually for a period of 3 years.

Two national meetings are held each year—the Annual Meeting during January or February, and the Semi-Annual Meeting usually in June. Regional meetings are held at the direction of the Council and 41 local chapter organizations hold monthly meetings.

The three major activities of the Society are: Membership service, Publication, and Research, the record of its accomplishments being permanently recorded in the annual TRANSACTIONS.

Headquarters of the Society are maintained in The New York Life Insurance Building, 51 Madison Avenue, New York, N. Y., and its research laboratory, devoted to the study of fundamental principles of heating, ventilating and air conditioning, is located at 7218 Euclid Avenue, Cleveland, O.

In September, 1894, a little group of nationally known engineers, educators and manufacturers gathered in New York and agreed that the great art of heating and ventilating deserved and required recognition as an essential, distinctive and highly specialized division of modern engineering.

These keen, alert, progressive men knew that the methods and equipment of their day could be improved, even beyond their own vision, if all the personalities striving for such improvement could be welded into one organized cooperative group imbued with the same ideals and aiming for the same goal. They therefore formed themselves into the nucleus of such an organization and called it The American Society of Heating and Ventilating Engineers.

Foreseeing the need for research they made it one of their first acts to establish a Committee on Standards. That the Charter Members had great faith in their enterprise is evident, although little did they dream that progress would be so rapid in their profession.

During the intervening years, since that little group of 75 pioneers unfurled the banner of THE AMERICAN SOCIETY OF HEATING AND VENTILAT-

ING ENGINEERS, thousands of the real leaders of thought and action in heating, ventilating, and air conditioning have gathered about that standard and carried it proudly before them far along the way of outstanding accomplishment. They may be identified among engineering groups by the distinctive emblem which was adopted by the Charter Members.

The first Annual Meeting was held in New York, N. Y., January 22–24, 1895, and the organization was incorporated that same year, under the laws of the State.

A. S. H. V. E. RESEARCH LABORATORY

Since 1919 THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has maintained a permanent research staff and facilities devoted solely to the study of fundamental problems in the field of heating, ventilating and air conditioning. During the past quarter century much has been done to advance the art by establishing scientifically sound data which the engineer can apply in the design, operation, and maintenance of heating, ventilating, and air conditioning systems and equipment.

For twenty-five years the Society's Research Laboratory was located at the U. S. Bureau of Mines Building, Pittsburgh, Pa. The Laboratory was moved to Cleveland in 1944, and early in 1946 the Society purchased premises at 7218 Euclid Avenue. In addition to work at the Society's Laboratory, a substantial part of the research program has been carried on through the medium of cooperative agreements with leading educational institutions of the United States and Canada.

All research activities are planned and supervised by the Committee on Research of 15 elected members, assisted by various Technical Advisory Committees of the Society. The research activities are financed from Society funds, of which a portion comes from membership dues and from its publications, and these funds are amplified by contributions from friends in the industries engaged in the general field of heating, ventilating, and air conditioning.

THE GUIDE

A distinctly new service was inaugurated by the Society in 1922 when it established THE GUIDE. Now in 1949, as the 27th Edition of the HEATING VENTILATING AIR CONDITIONING GUIDE makes its appearance, it is notable that THE GUIDE has served effectively not only the membership but the entire profession and the allied industries, and has received world wide recognition as a reliable and authoritative compendium of useful heating, ventilating and air conditioning data.

Throughout the 27 years of its service THE GUIDE has become a reference book of unchallenged position in its special field of engineering. The intention of its founders, to provide an instrument of service containing reference material on the design and specification of heating, ventilating and air conditioning systems and containing essential and reliable information concerning modern equipment, has been carefully safeguarded by those responsible for the compilation of each edition.

Steadily through the succeeding years THE GUIDE has grown in size, in service and in importance, so that the current edition contains 1384 pages of authoritative up-to-date information pertaining to the design and specification of heating, ventilating, air conditioning and piping systems, as well as a comprehensive Catalog Data Section on the modern equipment of leading manufacturers.

THE GUIDE exerts today one of the most positive influences tending to elevate, improve and extend the whole Art and Industry of Heating, Ventilating and Air Conditioning. It is universally recognized as the most useful and authoritative work in its field, being used by practicing engineers and manufacturers in all parts of the world, and as a text-book by a growing number of the world's principal engineering institutions.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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